Design of a flexible chain for winch based point absorbers

Kjell Andersson[1], Anders Hagnestål[2], Ulf Sellgren[3]

[1] Machine Design, KTH Royal Institute of Technology
Stockholm, Sweden
kan@kth.se
[2]Electric Power and Energy Systems, KTH Royal Institute of Technology,
Stockholm, Sweden
E-mail
[3]Machine Design, KTH Royal Institute of Technology
Stockholm, Sweden
ulfse@kth.se

Abstract

Wave power is a renewable energy source with a potential to cover up to 10% of the world´s present energy consumption. It has however been difficult to find a cost effective solution to convert the wave energy into electricity. One significant part of the problem is the harsh marine environment in combination with waves that deliver high forces at low speed. These requirements are challenging for both dimensioning the mechanical structure as well as for an efficient energy conversion. The dimensioning forces strongly depend on the wave power concept, the Wave Energy Converter (WEC) implementation and the actual Power Take-Off (PTO) system. A key issue for making wave power a competitive alternative as a future energy source is to reduce the Levelized Cost Of Energy (LCOE) to a competitive level. Winch Based Point Absorbers (WBPAs) could potentially accomplish this if a key component – a low-cost, durable and efficient winch, can be developed.

A crucial part of a winch system is the force transmitting chain or wire that transmits the hydrodynamic wave energy via a winch drum to the generator. In this article we propose a solution with a flexible chain where an elastomeric bearing is used as a means to achieve the relative motion between the links in the chain. With this solution no sliding is present and the motion is achieved as a deformation in the elastomeric bearing. This paper focuses on the design of the links in the chain while the design of the elastomeric bearings is described in more detail in (Andersson, 2018). The link was designed based on two aspects, primarily to maintain minimum number of joints in the chain and the other one was to have a maximum possible relative angle between the links when rolled up over the drum in order to effectively utilize the flexing property of the elastomeric bearing and reduce bending of the links. A detailed strength analysis of the link has been performed as well as topology optimization to increase the strength to weight ratio.

Keywords: Wave power, point absorbers, winch, power take-off, elastomeric bearing

1 Introduction

Wave power is a promising future source of renewable energy, which potentially could contribute to an energy production corresponding to as much as 10% of the world´s present energy consumption (Stock-Williams, 2012). Point absorbers are compact WECs where a buoy at the surface is moved by the waves, and that motion is used for energy conversion to electricity. The implementation of the Power Take-Off (PTO) unit is in most cases a linear device. Such devices occupy a lot of space and suffer from very large end stop forces at storms due to limitations of the stroke length. If we can implement a PTO using a winch-based system (see figure 1) it can be considerably smaller and would not suffer from the end stop problem. It has, however, proven to be very difficult to implement such a winch that is durable enough to handle the up to 80 million cycles that wave power units experience during their lifetime.

Figure 1 Illustration of a point absorber with a winch-based PTO where the winch is integrated in the buoy (Agge, 2016)

A winch-based system must also be able to handle the large hydrodynamic forces as well as the control forces required for efficient energy conversion. Properly balanced control forces can significantly increase the amount of harvested energy and they are, thus, important means to significantly reduce the LCOE. Standard winches have 1-2 orders of magnitude too short lifetime to be economically viable options. An ordinary wire-based winch system is not able to take more than in the order of 100 000 to 1 000 000 cycles due to bending fatigue and wear when the wire is rolled on and off the drum, which gives far too short service intervals and consequently a very low system availability (Hagnestål A., 2017). Also note that maintenance at sea is very expensive. Thereby, new mechanical transmission solutions are required if cost-efficient Winch Based Point Absorbers (WBPAs) should be implemented. This paper presents one such proposal where a flexible chain has been developed. Several WBPA concepts have been presented during the last 15 years or so. In Norway, Ingvald Straume developed WBPA with his company Straumekraft AS (Straume, 2010), and later Purenco. Fred Olsen has developed several WBPA during the last fifteen years and a prototype of their current concept Lifesaver, which has several winches working in parallel, has successfully operated outside Hawaii ("Fred Olsen," 2018). Nemos is a relatively new concept which also employs winches for the Power Take-Off function. The Nemo concept is adapted for sites with existing offshore wind power, where they can share the same power connections and mechanical

structures to reduce cost ("NEMOS," 2018). This system is however not a point absorber, but rather an elongated floating body that transmits wave energy to a generator shaft by a belt system.

As far as we know, there is no WBPA system existing today that can deal with the large forces required for efficient phase control. At KTH Royal Institute of Technology, we have established a multidisciplinary cooperation platform for ocean energy, and we are currently developing such a winch system within that platform. The chain solution with flexible elastomeric bearings is part of a development project for such a winch system where a prototype will be built by the end of 2018. If the project is successful, the intension is to develop a WBPA and to demonstrate that wave power can be a competitive future source of renewable energy. If the chain solution with elastomeric bearings can be used in a winch solution for WBPA this can be one important contribution to the project. What we aim to find out has been formulated as a research question is; *"Can a chain concept with elastomeric bearings be a potential solution in a winch for a WBPA?"*

This paper is organised as follows; chapter 2 presents some initial requirements for a winch based solution. Chapter 3 presents a design process for designing a durable chain with elastomeric bearins. In chapter 4 the proposed concept is experimentally tested. Chapter 5 discusses the results and finally conclusions and future work are given in chapter 6.

2   Requirements

For the development of a winch based solution we have formulated some initial preliminary requirements for the targeted WEC size and operation conditions (Hagnestål A., 2017). In table 1, requirements for two different sizes are listed where the 1/10 force scale is aimed for seas with smaller waves, such as the Baltic Sea, and the full scale is for North Atlantic sea conditions. It should however be noted that the main focus of this paper is on one main important component of such a winch system, i.e. the power transmission chain.

Table 1. Preliminary winch requirements

| Winch unit	1/10 scale	Full scale
Maximum stroke	25 m	37 m
Peak vertical speed	7 m/s	8 m/s
Typical speed (peak)	0.5-2 m/s	3-4 m/s
Maximum force	200 kN	2000 kN
Winch efficiency	> 97 %	> 97 %
Requirements from operational environment	Resistance to corrosion, UV radiation, biofouling	Resistance to corrosion, UV radiation, biofouling
Environmental impact	No leakage of non-biodegrad-able fluids	No leakage of non-biodegradable fluids
Design life	20 years, 80 million cycles	20 years, 60 million cycles
Service intervals	> 5 years	> 5 years
Winch width	< 2 m	< 3 m

In addition to these initial requirements we have made some assumptions about drum diameter and the existence of a pretension system for securing that the chain is always having a minimum required tension. We also assume that a safety system handling overvoltage and other power related problems, e.g. overheating in the PTO is present. These types of problems typically occur during the largest waves (25m) which causes high power and speed to handle.

3 Concept development

The design of a winch solution as a PTO for a Wave Energy Converter is facing many challenges and contradictory requirements. In an earlier paper (Hagnestål A., 2017) we have identified and discussed some of these challenges e.g. the diameter of the drum need to be chosen to balance the requirement of a high speed/low torque, i.e. a small drum radius, with a preferred large radius for the chain or wire being winded around to increase fatigue life.

3.1 Chain concepts with flexible pins

Chain-like solutions require handling of rolling and sliding contact between the links. A suggested solution approach to this problem was introduced in (Hagnestål A., 2017), where the rolling and sliding contact was avoided by replacing this movement with elements having an elastic deformation. The main idea here is to place elastic components between load carrying links in the transmission. In figure 2 below, two possible arrangements suggested in (Hagnestål A., 2017), are shown; the use of a rubber-like elastic part (left) and the use of metallic fins in spring steel (right).
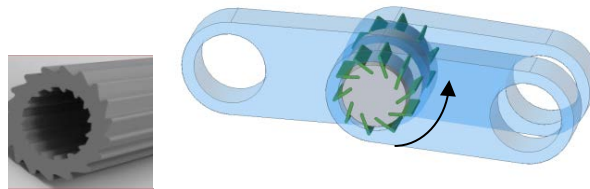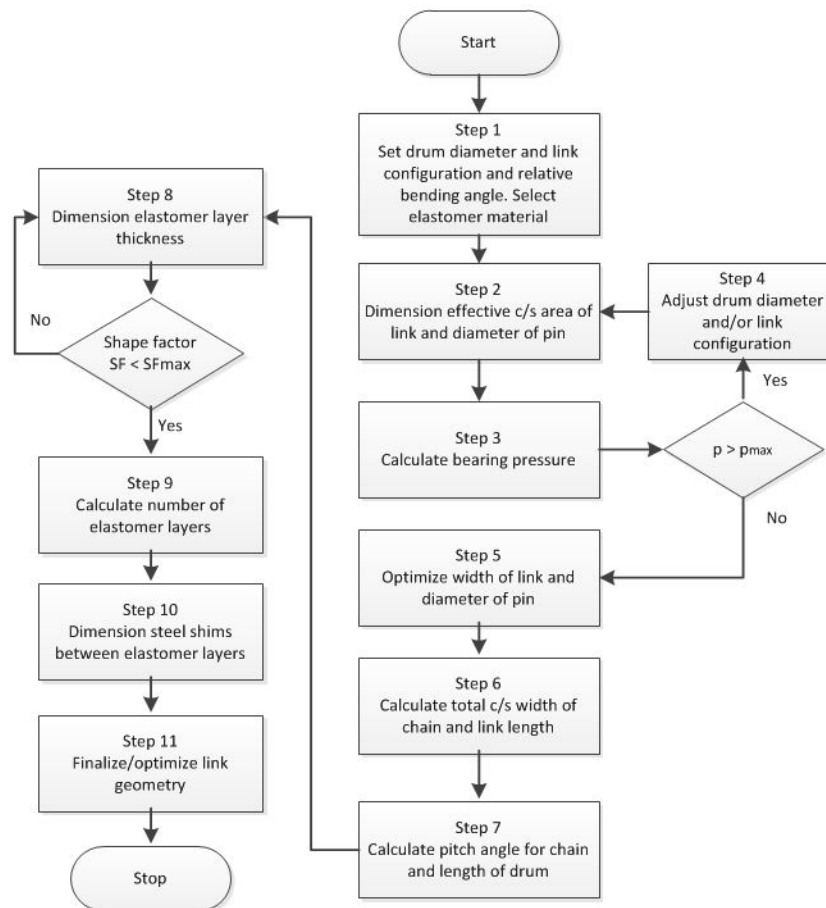

Figure 2 Two possible arrangements using a flexible pin; the use of a rubber-like elastic part (left) and metallic fins in spring steel (right).

In this type of arrangement only elastic deformation will occur when winding the chain-like transmission on the drum and no sliding will occur between the components of the chain. But there will of course be some sliding in the the link-drum contact.

3.2 Chain concept with elastomeric bearings

A further development of the flexible pin concepts resulted in a concept solution with elastomeric bearings as the flexible part allowing the relative angular movement between the links.

Start

Step 1
Set drum diameter and link configuration and relative bending angle. Select elastomer material

Step 2
Dimension effective c/s area of link and diameter of pin

Step 4
Adjust drum diameter and/or link configuration

Step 3
Calculate bearing pressure

p > pmax

Yes

No

Step 5
Optimize width of link and diameter of pin

Step 6
Calculate total c/s width of chain and link length

Step 7
Calculate pitch angle for chain and length of drum

Step 8
Dimension elastomer layer thickness

Shape factor
SF < SFmax

No

Yes

Step 9
Calculate number of elastomer layers

Step 10
Dimension steel shims between elastomer layers

Step 11
Finalize/optimize link geometry

Stop

Figure 3 Illustration of the design process for developing a chain with elastomeric bearings.

A design challenge is to make the transmission chain stiff in tension and flexible in bending, i.e. to make the elastomeric bearings stiff in compression and flexible in shear. For that reason, the elastomeric bearings are composed of a number of elastomer layers and steel shims bonded together by a vulcanization process, which is a chemical process for converting natural rubber or related polymers into more durable materials by the adding sulphur or other equivalent curatives or accelerators. The design process for developing a chain with elastomeric bearings is illustrated in figure 3.

The initiation of the process is the set of requirements for the winch chain to fulfill, as listed in table 1. Some initial assumptions and choices must be taken as the first step of this process. The start of the process (step 1) is to set winch drum diameter and relative bending angle between each pair of the links in the chain. The material for the elastomeric bearing also needs to be selected as basis for the design. Next (step 2) the effective cross section area necessary to take the peak shear load (100kN) is calculated as well as the pin diameter. Thereafter, the bearing pressure is calculated and compared (step 3) with the maximum allowed pressure for the bearing. The maximum allowed pressure is based on a demand to keep the strain in the elastomeric material low to avoid a hardening effect on the shear modulus G. If the pressure is too high the design needs to be modified (step 4), either by changing link configuration (e.g. to 2+3 from 1+2) or changing drum diameter. Once an acceptable pressure is obtained the size of width of the link and the diameter of the pin is varied until an optimal set of values is obtained (step 5). Thereafter, based on the obtained values the total width of chain and the link length are calculated (step 6). Step 6 results in an overall width of the chain of 200 mm including the gaps between the links which allow a

sideway angle between links for rolling the chain over the drum in a spiral way. The angles of interest here are the helix angle α and the side angle β, see figure 4, where α is the helix angle required to achieve the required pitch and β is the relative side angle between two consecutive link set in a plane tangential to the drum surface. This requires a clearance beneath the pin of every link set to achieve that degree of freedom, see figure 5.
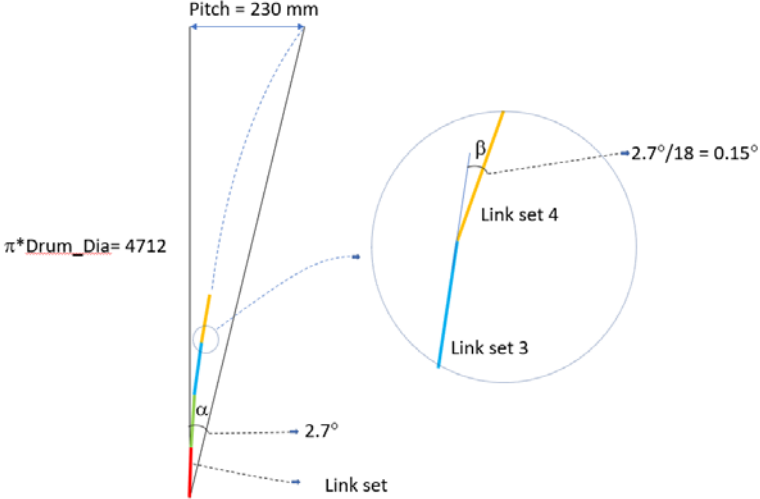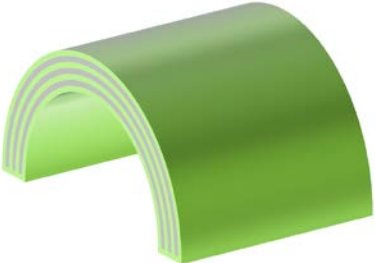

Figure 4 Definition of helix angle α and side angle β.
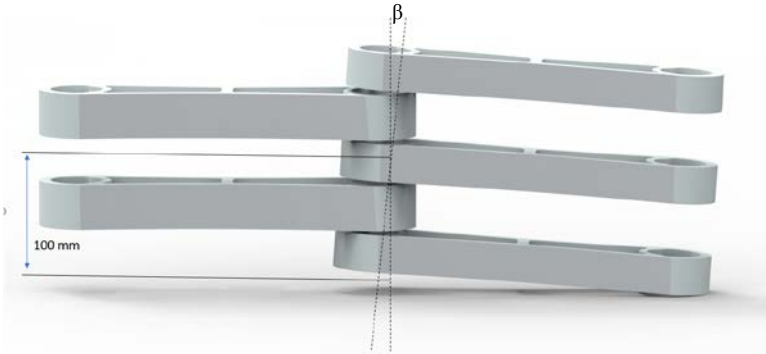
Figure 5 Illustration of side angle β.

Next step in the process is to design and dimension the elastomeric bearings which are the active elements in the chain allowing the relative angular movement between the links. A principle layout of the elastomeric bearing for this chain is illustated in figure 6.


Figure 6 Principle layout of a laminate elastomeric bearing (Lingaiah, 2017)

The thickness and number of the elastomer layers can be determined by considering both compressive and shear stiffness. The required shear stiffness is determined by the elastomer thickness while the number of elastomer layers is determined by the required compressive stiffness. For the elastomeric layers we have chosen a nitrile rubber, NBR 80 Shore A with the tradename Nipol NX775 (Zeon_Chemicals), which has a relatively high stiffness and

good resistance to the actual marine environment. A more detailed description of the design and dimensioning of the elastomeric bearings (step 7-10) is given in (Andersson, 2018) and is not discussed in any detail here.

The link was designed based on two aspects, primarily to get a minimum number of joints in the chain and the other one was to have a maximum possible relative angle between the links when rolled up over the drum in order to effectively utilize the flexing property of the elastomeric bearing. A more detailed strength analysis and final geometry modifications of the link (step 11) has been made using ANSYS Workbench 18.1 ("ANSYS Workbench 18.1,") finite element modeling and analysis tool.

In order to minimise weight and increase the strength to weight ratio we have used topology optimization of the link, see figure 7. One objective was set to minimize the compliance for a load case of 100 kN with two response constraints; to reduce the volume by 50 percent and to maintain a global von Mises stress below 300 MPa.
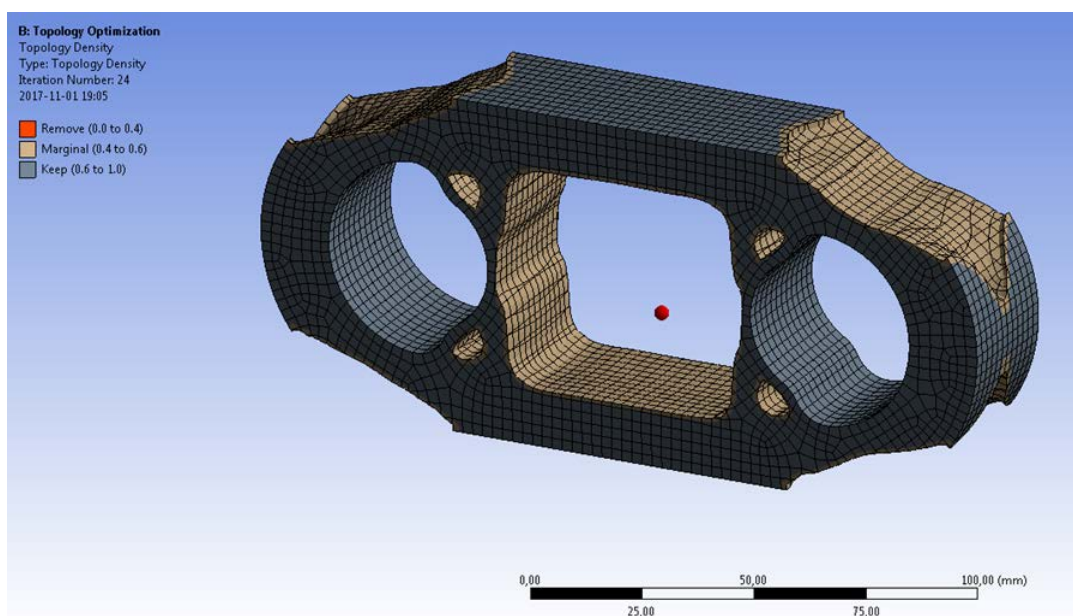
Figure 7 Resulting geometry after topology optimization (Lingaiah, 2017)

Based on the results from the topology optimization the shape of the link was adjusted. A specific cut out was given to both ends ofthe link. The reasons for the cut out is locate the elastomeric bearing in place and partially prevent the possible sliding motion of the outer layer if there is any breakage of bond with the link. The cut out includes the geometry of the elastomeric bearing in the outmost radius and pin geometry in the inner radius, see figure 8. Also an axial clearance of 3 mm between the links mounted on the connecting pin is added. This is to enable the chain to be wound over the drum and allows the chain to have relative angle between the links in a plane tangential to drum surface. The top and bottom surface of the link were given a radius of curvature to maintain a conformal contact with the drum surface, see figure 8.

Elastomeric bearing outer radius
Pin radius
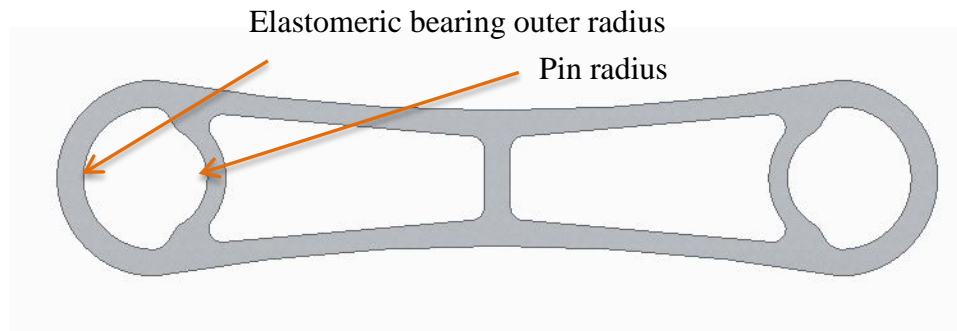
Figure 8 Link geometry after step 11.

Next we will make a more detailed strength analyses using ANSYS to verify the chosen dimesnions of the link geometry after these modifications. First, we need to understand the type of loading that the chain is subjected to while being rolled in and out from the winch as the buoy moves up and down with the heaving motion of the waves. Basically, the chain links are subjected to tensile loads during both the roll out and roll in phases of operation. However the magnitude of the forces differ. The chain is expected to withstand forces up to 200 kN which is the worst case scenario and hence the design is analysed for the worst loading case to meet this requirement. Due to space limitations we only show the load case when the chain is vertical and axially loaded. The conditions for the analysis are as follows;

- Max loading case (1/8 Model)
- Material: Duplex steel EN 1.4462
- Material Yield strength: 500 MPa
- Max load applied : 25kN
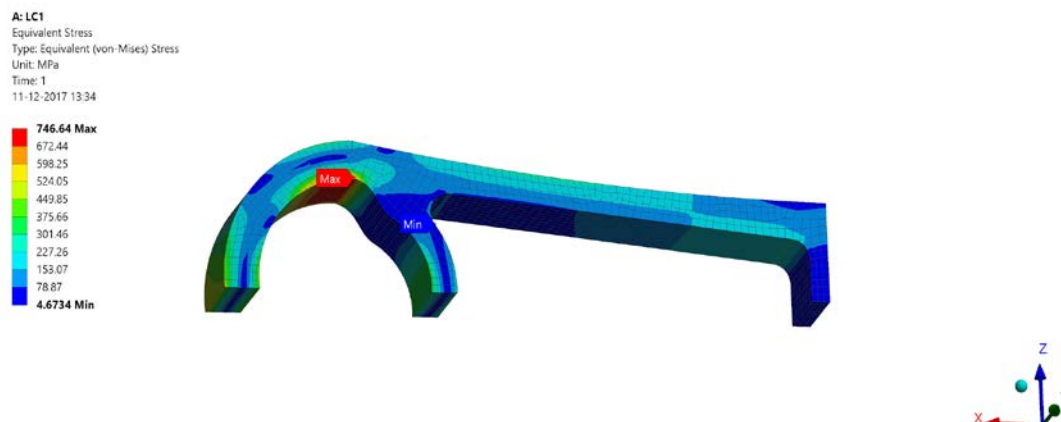
Figure 9 Effective von Mises stresses for max tension load.

The highest equivalent von-Mises stress in the link is 746 Mpa (figure 9). This is a local stress concentration due to an abrupt geometry change which causes a singularity in that mesh region, i.e. it is a synthetic stress peak that can be handled by rounding the edges. However, the max nominal stress in the link is around 300-350 MPa. This stress value is well below the yield strength of the material which is 500 MPa. The resulting chain configuration (2+3) with elastomeric bearings and its preliminary parameter values are shown in figure 10 and table 2 respectively.
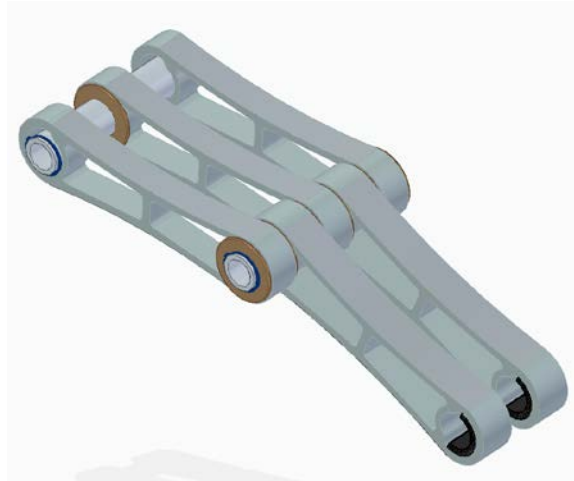
Figure 10 The dimensioned 2+3 link combination with elastomeric bearings for the 1/10 force scale application

Table 2. Preliminary chain parameter values for 1/10 force scale

| *Chain parameter*	*Value*
Pin outer diameter	40 [mm]
Pin inner diameter	33 [mm]
Link1 width	40 [mm]
Link2 width	33 [mm]
Chain width	195 [mm]
Link length (cc)	260 [mm]
Drum diameter	1500 [mm]
No of elastomer layers	4
Thickness of elastomer layers	1 [mm]
No of steel shims	3
Thickness of steel shims	1 [mm]

4   Experimental testing

To test the complete chain, many steps need to be performed. Due to the high loads on the chain in real applications a scale protope is necessary for testing both the strength and fatigue properties of the proposed chain design. However, before making the scale of the assembled chain, component testing is considered to be the first step. A crucial component to start testing is the elastomeric bearing. Here we want to examine if the basic assumption that shear stiffness remains constant or almost constant independent of axial tensile load is true.

For this purpose we have developed a prototype version of the chain and the elastomeric bearing. We have chosen to use one link with the full length (see table 3) but with a smaller hole and pin radius. The elastomeric bearing is adapted and simplified due to manufacturing considerations. The tested bearing is composed of two elastomer layers using the rubber NBR 75 Shore A, with one steel shim in between. Material data and dimensions for the prototype bearing used in the test setup are given in table 3.

Table 3.Material data and dimensions for the prototype bearing

Material property/Bearing data	Value
Shear modulus	1.14 [MPa]
Young's modulus	3.37 [MPa]
Material compressibility coeff., Ø	0.64
Pin outer dia	40 [mm]
Link hole dia	54 [mm]
Link width	40 [mm]
No of elastomer layers	2
Thickness of elastomer layers	2.5 [mm]
No of steel shims	1
Thickness of steel layer	3 [mm]

The test rig comprises of only one link manufactured to full scale, see figure 11. Two rail guide beams form the base of the test rig setup which is positioned vertically with necessary support. Bearings and bearing housings are mounted at the top end of the rail guide beams.
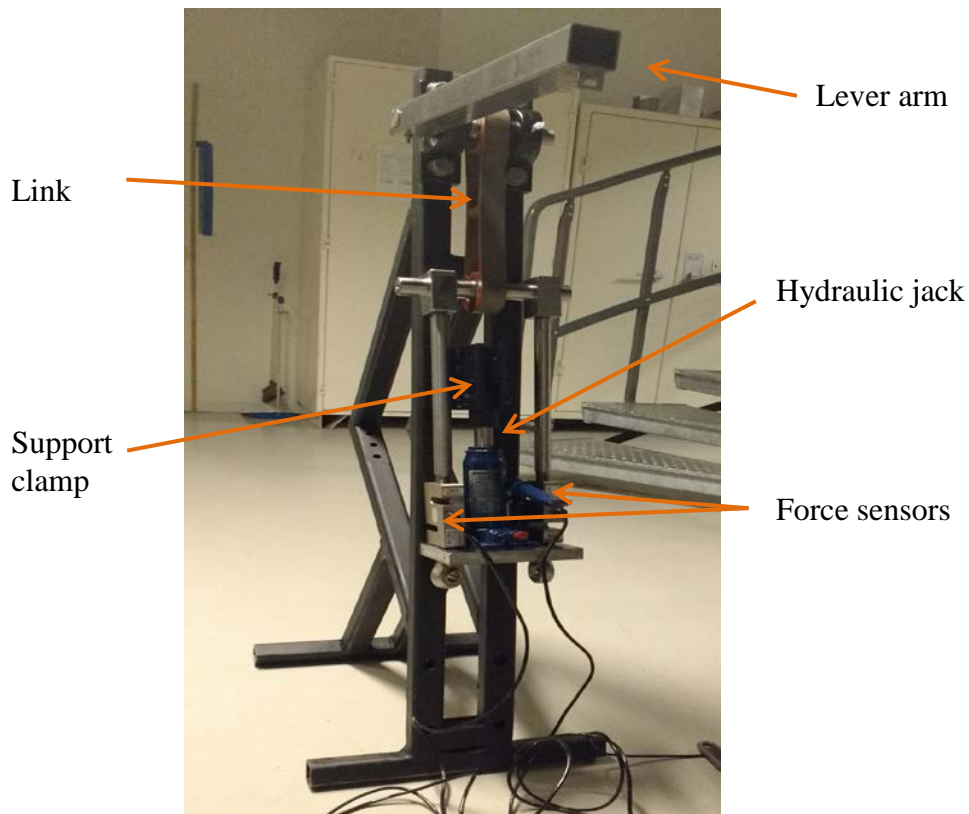
Figure 11 Test rig with hydraulic jack for tensile loading and a lever arm for torsion loading.

The pin is made slightly longer with a slot cut on the end to secure a lever arm. The lever arm is used to apply the bending moment to get a shear deformation in the elastomeric bearing. On the lever arm dead weights will be mounted to produce a given bending moment. A support block is mounted below the bottom end of the link. This block is bolted firmly on the rail beams. A hydraulic jack is used to provide the tension force in the link. In Figure 11 the hydraulic jack and supporting attachments arrangement are shown. When the hydraulic jack is actuated, the piston pushes against the fixed solid support clamp. This in turn will tension the link and the elastomeric bearing.

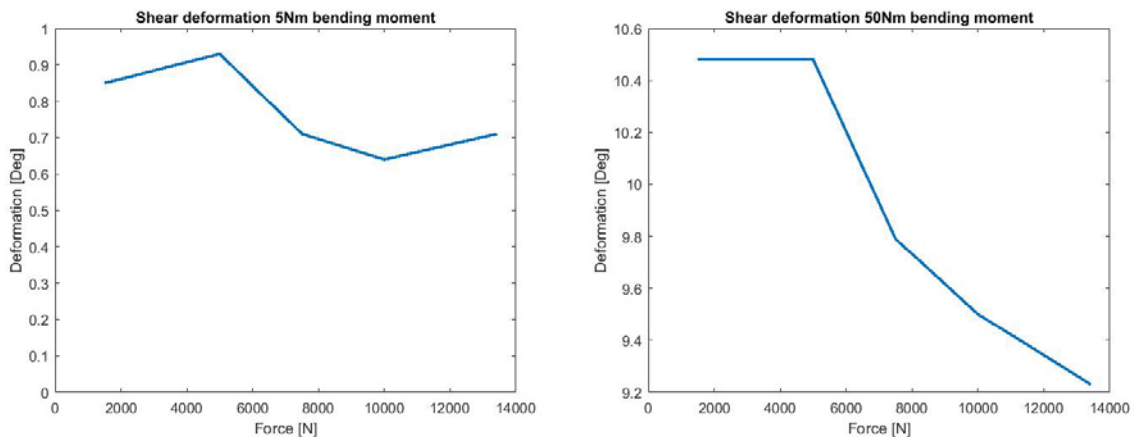The measuring setup consists of the following;
- Load cells ANYLOAD 101BH ("ANYLOAD 101BH," 2017) are used to measure the applied force.
- A dial gauge is used to measure the radial deformation of the elastomeric bearing relative to the link. In this setup the total deformation of two axially loaded bearings is measured. The link deformation is assumed to be negligible compared to the bearing deformation.
- A laser pointer is mounted on the lever arm and is projected on a wall on a given distance from the test rig.

4.1 Experimental results

The table below gives the results of the experiment carried out. During the tests, the radial deformation and deflection reading of the laser are captured. The laser is projected on a wall at a given distance from the test setup. Shear deformation in the elastomeric bearing is then calculated based on the distance of the laser from the wall.

Table 4. Experimental results, combined compression and torsion load (Andersson, 2018)

Compression load (N)	Moment applied (Nm)	Radial deformation (mm)	Shear deformation (Deg)
1500	5	0.12	0.85
	50	0.12	10.48
5000	5	0.40	0.93
	50	0.36	10.48
7500	5	0.52	0.71
	50	0.50	9.79
10000	5	0.87	0.64
	50	0.87	9.5
13420	5	0.88	0.71
	50	0.84	9.23

Figure 12 Shear deformation in the elastomer bearing as function of compressive load for 5 Nm bending moment (left) and 50 Nm (right).

Figure 12 shows the variation of shear deformation in the elastomer bearing as function of compressive load for a beding moment of 5 and 50Nm respecively. The aim was to investigate the deflection angle for the applied bending moment. The results support the assumption that if the strain in the rubber layer is kept low (< 10%) the shear stiffness should be relatively constant. However we can see that the strain caused by the higher loads will result in a slight increase of the shear modulus for the 50Nm bending load.

The radial deformation as function of compressive load is shown in figure 13. These graphs confirms the assumption that the design critera used to achieve a more or less constant stiffness, independent of compressive load is valid.
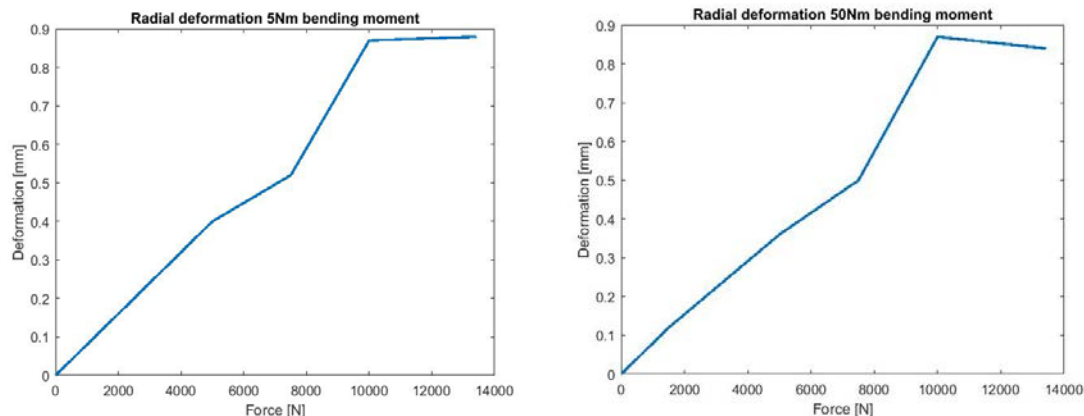
Figure 13 Radial deformation in the elastomer bearing as function of compressive load for 5 Nm bending moment (left) and 50 Nm (right).

5 Discussion

The durable winch challenge is a key problem to solve for WBPA units, and if this problem is solved point absorbers with low LCOE could potentially be constructed. It is important to realize that the problem gets harder for larger forces and larger WECs, due to scaling effects. The problem to accomplish a durable winch is rather hard to solve, but far from impossible. It is evident that the approach presented here could potentially solve the problem with the requirements given in chapter 2 at a sufficiently low cost.

As shown in figure 3, the dimensioning of the links requires a lot of trade-off decisions. On a general level we need a set of links with sufficient cross section to take the high tensile loads. A starting condition is to find a high strength material with a high yield strength that also can deal with the harsh marine environment. Here we have chosen the EN 1.4462 Duplex steel with yield strength of 450 MPa. A thin width of the link would require a thicker layer of elastomer thus leading to many layers of the elastomer to avoid high strain in the elastomer bearing. We want to keep the strain not larger than about 10% to reduce its influence on the shear modulus G. Also how to combine the links in a 1+2 or 2+3 link fashion need to be taken into account.

It should also be noted that the dimensioning procedure for elastomeric bearings from (Gent, 2012),(Andersson, 2018) is an approximate method which can explain the differences between the theoretical and experimental results. The uncertainty about material properties for the rubber material used in the experiments could explain a significant portion of the observed deviation. However, the experimental results indicate that the presented approach is a potential solution to the durable winch problem. It still remains to find out the fatigue properties when testing in real conditions where water absorption, UV-radiation and biofouling will influence the operational life. This applies for both the chain solution as well as for the winch system in total. Also, many practical problems with for example force transmitter guiding systems can be hard to predict before offshore tests have been conducted. Therefore, in practice, only estimates of these properties can be made before sea trials are conducted.

6 Conclusions and future work

In this article we propose a solution with a flexible chain where elastomeric bearings are used to achieve the relative motion between chain links. With this solution no sliding is present in the link-to-link interfaces and the bending of the chain when rolled up on the winch is achieved as a deformation in the elastomeric bearing. The links were designed based on two aspects, primarily to maintain minimum number of joints in the chain and the other one was to have a maximum possible relative angle between the links when rolled up over the drum in order to effectively utilize the flexing property of the elastomeric bearings.

One focus of this paper has been to investigate the research question; *"Can a chain concept with elastomeric bearings be a potential solution in a winch for a WBPA?"*. To answer this question, a full size chain in a 2+3 link configurations with elastomeric bearings have been dimensioned. In addition a prototype version of link with bearings has been designed and experimentally tested in a test rig as a first proof of concept. The first preliminary test results support the hypothesis that this chain concept can both take the large tensile loads and allow for enough angular deflection. This work is ongoing, and the concepts presented here will be carefully evaluated during the remainder of 2018. A prototype of the complete 2+3 chain will also be built and tested in our lab during 2018. Future work also includes a more detailed analysis of the elastomeric bearing concept as well as of the complete winch system with the chain winding up on the drum.

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