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A flexible chain proposal for winch based point absorbers

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ABSTRACT

Ocean wave power is a promising renewable energy source for future energy production. It has however been difficult to find a cost-effective solution to convert the wave energy into electricity. The harsh marine environment and the fact that wave power is delivered with high forces at low speeds makes design of durable mechanical structures and efficient energy conversion challenging. 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 WEC using a winch as a Power Take-Off system, i.e. a Winch Based Point Absorber (WBPA), could potentially accomplish a low Levelized Cost Of Energy (LCOE) if a key component – a low-cost, durable and efficient winch that can deal with high loads – can be developed.

A key problem for achieving a durable winch is to find a force transmitting solution that can deal with these high loads and handle up to 80 million cycles. In this article we propose a design solution for a force transmitting chain in a WBPA system where elastomeric bearings are used as a means to achieve the relative motion between the links in the chain. With this solution no sliding is present and the angular motion is achieved as a deformation in the elastomeric bearing when the chain is winded on a drum.

The link was designed primarily to minimize the number of joints in the chain: Thereby the maximum allowed relative angle between the links when rolled up over the drum should be as large as possible within practical limits. The angle is to be handled by the elastomeric bearing. A detailed strength analysis of the link has been performed as well as topology optimization to increase the strength to weight ratio.

A test rig for a first proof of concept testing has been developed and the first preliminary test results indicate that this concept with using elastomeric bearings can be a potential solution for a durable chain and should be analyzed further for fatigue conditions and under water operations.

Keywords: Chain transmission, Power take-off system, Elastomeric bearing, Wave power.

1. INTRODUCTION

Ocean 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 [1]. It has however been difficult to implement the Wave Energy Converters (WECs) due to a few key problems that are difficult to address with standard technology. Thereby, development of new design solutions that address these key problems is needed.

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 require a lot of space and suffer from very large end stop forces at storms due to a limited stroke length. A winch-based PTO would become 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. 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 the reduce the Levelized Cost Of Energy (LCOE) to sufficiently low values.

Standard winches have 1-2 orders of magnitude too short lifetime to be economically viable options. An ordinary wire-based winch system will not be able to take more than in the order of 100 000 to 1 000 000 cycles (compare with the required 80 million 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 very low system availability [2].

Several WBPA concepts have been presented during the last 15 years or so. In Norway, Ingvald Straume developed WBPA with his company Straumekraft AS [3], and later Purenco. It seems, however, that they are no longer active. In Sweden, Ocean Harvesting Technologies developed a WBPA concept for several years, where a counter weight was used as a pretension system [4]. They, however, abandoned the idea since they could not find a durable winch solution. 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 [5]. The relatively new concept from Nemos [6] also employs winches for the Power Take-Off function. Their concept is adapted for sites with offshore wind power, where they can share the same power connections and mechanical structures to reduce cost. 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.

It is obvious that new mechanical transmission solutions are required if Winch Based Point Absorbers (WBPA) should be possible to implement with a low Levelized Cost Of Energy (LCOE). One possibility is to base the transmission solution on a chain concept since chains are good at taking high forces and commonly used in heavy duty transmissions. However, chain based solutions require handling of the problem with a rolling and sliding contact between the links. A suggested solution approach to this problem was introduced in [2], where this rolling and sliding contact was avoided by replacing this movement with elements having an elastic deformation. The main idea with this conceptual solution was to place elastic components between the load carrying links in the transmission. In figure 1 below, two possible arrangements suggested in [2], are shown; the use of a rubber-like elastic part (left) and the use of metallic fins in spring steel (right).

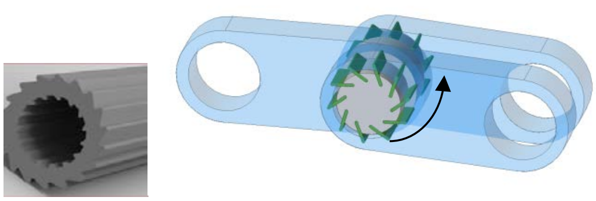

Figure 1 Two possible arrangements using a flexible pin; the use of a rubber-like elastic part (left) and metallic fins in spring steel (right).

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.

This paper presents a force transmission solution for a WBPA which is a further development of the chain based solution in [2] and in this solution the elasticity between the links is implemented using elastomeric bearings. The focus of this paper is on design and dimensioning of the elastomeric bearings while the design of the links in the chain is described in more detail in [7].

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. We currently have 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. What we aim to find out has been formulated as a research question; *"Can elastomeric bearings be designed such that they can both take the large tensile loads and allow for enough angular deflection to be used 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 theoretical introduction to elastomers and a concept for a flexible bearing solution. 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 [1]. In Table I requirements for two different sizes are listed where the 1/10 force scale winch 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.

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. The type of problems typically occur during the largest waves (25m) which causes high power and speed to handle.

TABLE I
PRELIMINARY WINCH REQUIREMENTS

Winch unit	1/10 force 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

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 [2] we have discussed a number of challenges that we have and identified some of these challenges. One example is that the diameter of the drum needs to be chosen to balance the requirement of a high speed/low torque with a preferred large radius for the chain or wire being winded around due to fatigue life. A principle layout of a WBPA winch with coordinate system and main components, i.e. a generator, wire/chain winded around the drum and a pretension system is shown in figure 2.

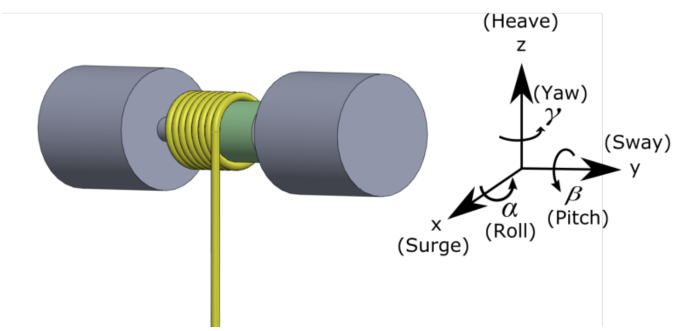

Figure 2 Coordinate system for a WBPA winch where corresponding coordinate axis for the special case when the winch is aligned with the waves are given within parenthesis.

3.1 Chain concept with elastomer bearings

A further development of the flexible pin concepts presented in [2], resulted in a concept solution with elastomeric bearings as the flexible part allowing the angular movement between the links.

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. It is a chemical process for converting natural rubber or related polymers into more durable materials by adding sulfur or other equivalent curatives or accelerators.

3.2.1 Design of elastomers

One important factor to keep in mind, when designing with elastomers is that these in general are very dependent on the shape factor of the body. The shape factor is defined as the ratio of loaded area to the total surface area that is free to bulge. Consider e.g. the rectangular elastomeric block in figure 3.

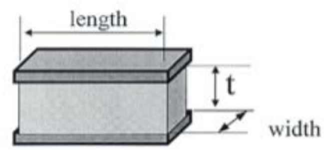

Figure 3 A simple rectangular elastomeric block.

If a normal vertical load is applied to the component above, then the shape factor S for such a component is given as:

$$S = \frac{A_L}{A_B} \qquad (1)$$

Where:

A_L= *Loaded area* $(length \cdot width)$ *and*

A_B= *Bulge area* $(2 \cdot (length \cdot width)/t)$

We can get a large increase in compressive modulus of an elastomer by increasing its shape factor making them capable to bear huge amount of compressive load. At the same time it is capable of flexing in the shear direction within certain limits. This means that to dimension the elastomeric bearing we need to consider both compressive and shear stiffness (K_c and K_s), how these are interdependent and how these are depending on loading conditions etc.

The compressive modulus is varying due to the variation of shape factor. When the shape factor increases, this cause a nonlinear increase of the compressive modulus of the elastomer, see figure 4.

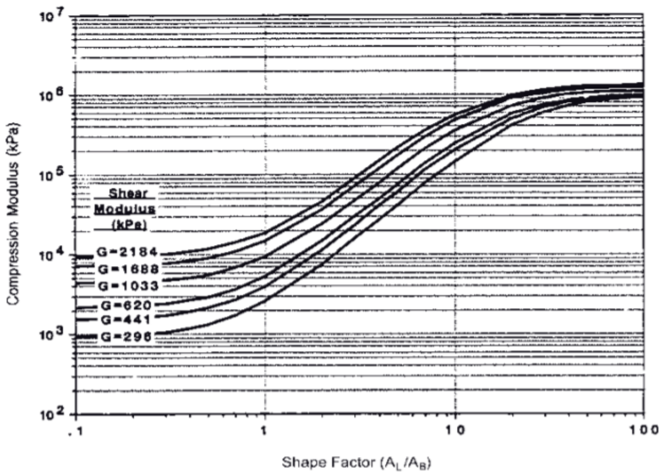

Figure 4 Influence from shape factor on compressive modulus E_c.[8].

Consider a flat sandwich block having bidirectional strain. Then an approximate expression for the relation between shape factor and altered compressive young's modulus is given in equation Eq.(2)[8].

$$E_C = E_0\left(1+\phi \cdot S^2\right) \qquad (2)$$

Where

E_c= Altered compressive modulus [N/mm^2]

E_0= Original compressive modulus in [N/mm^2

S= Shape factor

Ø=Elastomer compression coefficient which is an empirically determined coefficient.

The shear modulus is also affected by the amount of the compressive strain that exist in the elastomer due to the compressive load (see figure 5).

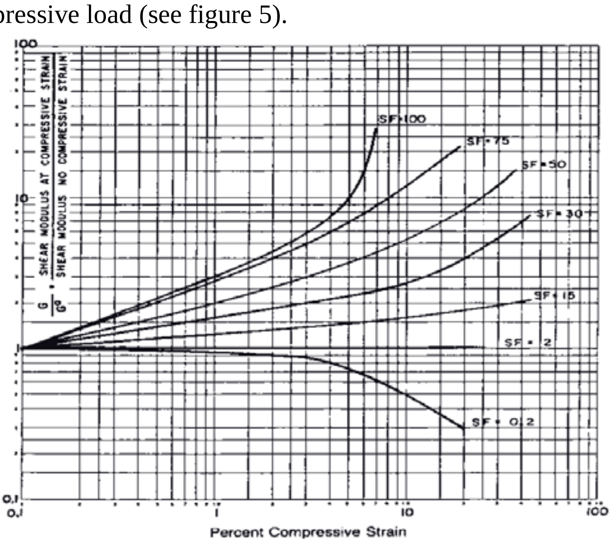

Figure 5 Influence of compressive strain on the shear modulus G [8], where SF in the figure is the shape factor S.

3.2.2 Design of elastomer bearings

Next we want to dimension a laminate bearing. Here we have an elastomer bearing consisting of a number of thin layers of steel sheets with elastomers in between, bonded to the steel surface using a vulcanization process. The principle layout of the actual bearing concept is shown in figure 6, where the green and grey layers correspond to the elastomer and steel correspondingly. For dimensioning of this bearing concept we use Eqs.(1) - (4).

Figure 6 Principle layout of a laminate elastomeric bearing [9].

The thickness and the number of elastomer layers can be determined by considering both compressive and shear stiffness. The required upper limit of shear stiffness is determined by the total elastomer thickness while the number of elastomer layers is determined by the required compressive load capacity. In the case of an elastomeric bearing with a rectangular shape similar to figure 3, and where the load is applied perpendicular to the elastomer, the compressive stiffness for an elastomeric bearing in a such configuration with *N* layers of elastomers can be expressed as [8];

$$K_c = \frac{F_c}{d_c} = \frac{A_L \cdot E_C}{t \cdot N} \qquad (3)$$

Where:

K_c = Compression spring rate

F_c = Applied compressive force

d_c = Compressive displacement

N = Number of identical elastomer layers

t = Individual layer thickness

A_L = Load area

E_c = Individual layer compression modulus

The shear stiffness for an elastomeric bearing in such configuration with a total elastomer thickness t_{tot} can be expressed as [8];

$$K_s = \frac{F_s}{d_s} = \frac{A_L \cdot G}{t_{tot}} \qquad (4)$$

Where:

K_s = Shear spring rate

F_s = Applied force in the shear direction

d_s = Shear displacement

A_L = Load area

G = Shear modulus

t_{tot} = Total thickness of elastomer

The steel shims also need to be dimensioned considering both fatigue loading and peak force loading. The procedure for dimensioning the steel shims is given in [10].

3.2.3 Dimensioning of a chain with elastomeric bearings

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 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 448 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 compression strain not higher than about 10% to reduce its influence on the shear modulus G, see figure 5. Also how to combine the links in a 1+2 or 2+3 link fashion need to be taken into account. The proposed chain configuration for the 1/10 force scale is based on using a 2+3 link combination with elastomeric bearings in the connections between the pin and the links. It is illustrated in figure 7.

For the elastomeric layers we have chosen a nitrile rubber, NBR 80 Shore A with the tradename Nipol NX775 [11], which has high stiffness and good resistance to the actual marine environment. In this initial stage of dimensioning the bearing we use the material properties for the chosen rubber given in [8] ,see Table II.

TABLE II
MATERIAL PROPERTIES FOR THE CHOSEN RUBBER(FROM [8])

Material property	***Value***
Shear modulus	2.19 [MPa]
Young's modulus	9.24 [MPa]
Material compressibility coeff., Ø	0.52

The design is made in an iterative manner starting with deciding the number of links for one turn around the drum. This gives an initial requirement of the angle that each link need to flex in relation to the connecting pin. The link is then given an initial shape based on strength calculations as well as for the connecting pin.

Then we calculate the number and thickness of the elastomer layers using Eqs.(1) – (4). Also the thickness of the steel shims between are calculated based on the procedure for dimensioning the steel shims as given in [10].

The main dimensions for this chain concept are summarized in Table III.

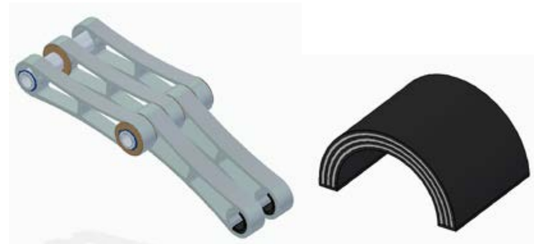

Figure 7 The dimensioned 2+3 link combination and elastomeric bearing for the 1/10 force scale application

TABLE III
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 OF THE ELASTOMERIC BEARING

As the first step to verify our chain concept we check the calculations of compressive and torsional stiffness for the elastomeric bearings. We also have a specific interest of investigating if the shear stiffness remains constant or almost constant independent of axial tensile load.

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 IV) 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 IV.

TABLE IV
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 diameter	40 [mm]
Link hole diameter	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 and a close up of the mounted bearing is shown in figure 8 and 9. As shown in figure 8, the test rig comprises of only one link manufactured to full scale. 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. 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 8 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.

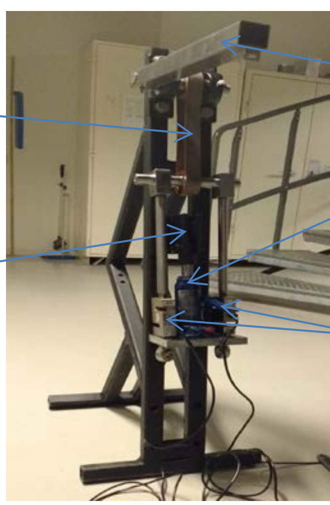

Figure 8 Test rig with hydraulic jack for tensile loading and a lever arm for torsion loading.

Figure 9 Elastomeric bearing mounted between pin and link in the test rig

The measuring setup consists of the following;

- Load cells ANYLOAD 101BH [12] are used to measure the force being applied.
- 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 links deformation is assumed to be negligible.
- 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 Test Procedure

Below is the test procedure for measuring deformations of the elastomer bearing when loading the link:

- Laser pointer and the dial gauge are set to zero reading.
- The hydraulic jack is loaded in steps as shown in Table
- V, which provides the compression load on the elastomer bearing.
- For every compression load value, the lever arm is loaded with dead weights of 1kg and 10kg. This provides the bending moment to shear the elastomer bearing.
- For every load case, radial deformation and position of the laser pointer is recorded.
- This experiment is repeated to confirm the results.

4.2 Experimental results

The table below gives the results of the experiment carried out. As highlighted in the test procedure, the radial deformation and deflection reading of the laser are captured. Shear deformation in the elastomeric bearing is then calculated based on the distance of the laser from the wall.

TABLE IV
EXPERIMENTAL RESULTS, COMBINED COMPRESSION AND TORSION LOAD

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

In the following section, the above results are analyzed and discussed.

Figure 10 shows the variation of shear deformation in the elastomer bearing as function of compressive load. The aim was to get a deflection angle for the applied 50 Nm 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.

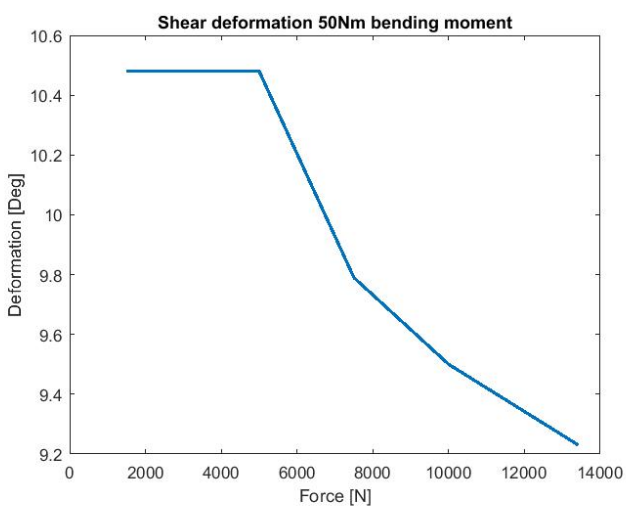

Figure 10 Shear deformation in the elastomer bearing as function of compressive load.

The radial deformation as function of compressive load is shown in figure 11.

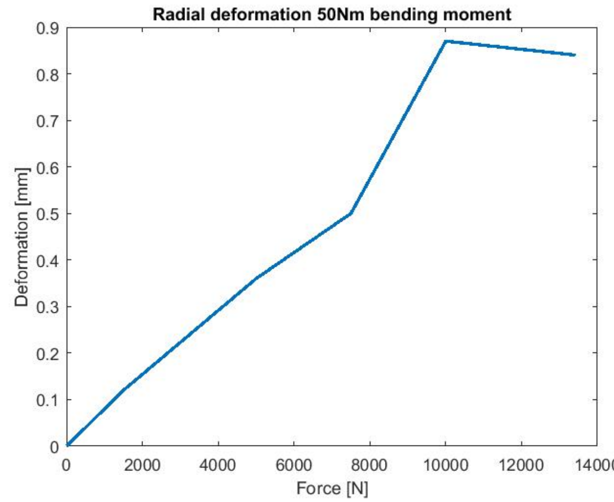

Figure 11 Radial deformation in the elastomer bearing as function of compressive load.

4.3 Verification of results

The experimental results shown in Table V and figures 10-11, have been compared with the theoretical values given by Eqs. (1)-(4). For deriving the compressive modulus from figure 11, we have estimated a linear function for K_c, thus neglecting the deformation value for F=10 kN. A second measurement series has also confirmed this linear dependency between force and deformation. Also, the fact that the measured deformation is acting on two bearings is taken into account. The applied torsion load of 50 Nm resulted in a deformation angle ranging from 9.23 – 10.48 degrees.

Comparison of experimental and theoretical results is summarized as;

- K_c : Experimental 30x106 [N/m] Theoretical 27.6x106 [N/m]
- Deformation angle for 50 Nm load; Experimental 9.23 – 10.48 [degrees] Theoretical 12.5 [degrees]

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. This is mainly since larger forces require thicker force transmitters, and that the winch drum diameter needs to be larger for thicker force transmitters due to space requirements and bending fatigue considerations. This is not necessarily compensated by a higher speed for the larger forces. When designing a WBPA system, the maximum force is a parameter that should be chosen from an economic viewpoint to minimize LCOE. The maximum force is obviously dependent on the buoy size and wave climate, but also strongly on the phase control algorithm that is employed.

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. It should be noted that the dimensioning procedure for elastomeric bearings from [8] is an approximate method which can explain the differences between the theoretical and experimental results.

Also the uncertainty about material properties for the rubber material used in the experiments could explain the deviations. 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 fatigue life. This applies to both the chain solution and the winch system in total. Also, many practical problems with for example force transmitter guiding systems can be hard to predict before offshore tests has 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 the links in a chain. With this solution no sliding is present and the motion 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 elastomeric bearings be designed such that they can both take the large tensile loads and allow for enough angular deflection to be used 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 concept with elastomeric bearings 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.

7. ACKNOWLEDGMENTS

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