International Journal of Engineering Sciences & Research Technology

(A Peer Reviewed Online Journal)
Impact Factor: 5.164

IJESRT

IJESRT
INTERNATIONAL JOURNAL OF ENGINEERING SCIENCES & RESEARCH TECHNOLOGY

THEORY AND SIMULATIONS OF AN END STOP SOLUTION IN A LINEAR WAVE POWER GENERATOR

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DOI: 10.5281/zenodo.3828902

ABSTRACT

The aim of this paper is to present a theoretical derivation of an end stop design in linear synchronous permanent magnet generator, utilized in a wave energy conversion system. To ensure a stable and proper design of the mechanical system and avoid a great impulse force in the hull, the upper end stop shall smoothly decrease the velocity of the translator to zero if the wave amplitude is larger than the stroke length. As the upper end stop, the spring, compresses, the downward force in the system is no longer only represented by the translator's and buoy's masses, but also the reaction force of the spring. The buoy exposes a greater downward force and submerges further down in the water. A relative motion between the wave and buoy occurs, an important parameter when the upper end stop shall be designed

KEYWORDS: Wave power generator, mechanical damping, hydrodynamics, linear generator.

1. INTRODUCTION

A number of different research groups around the world are currently investigating the possibilities to convert the energy in the ocean waves to electric energy [1-3]. Each project and technique have to handle different questions and challenges. The concept presented in this paper has been developed within Uppsala University, Sweden. The wave energy converter itself consists of a linear generator placed on the ocean floor connected to a buoy at the ocean surface.

The project has an experimental site on the Swedish west coast outside the town Lysekil. The first full-scale wave energy converter was installed in March 2006 and since then, the site has continuously been updated where the latest wave energy converter, L12, was deployed in July 2013. The main milestones in the project are further presented in [4-6].

In a linear electric machine, the moving magnetic part is known as the translator whereas the winding, is inserted in the stationary stator. Fig. 1 presents the projects latest prototype, the linear generator L12. For further reading of the linear generator L12, see [7].

ISSN: 2277-9655
Impact Factor: 5.164
CODEN: IJESS7

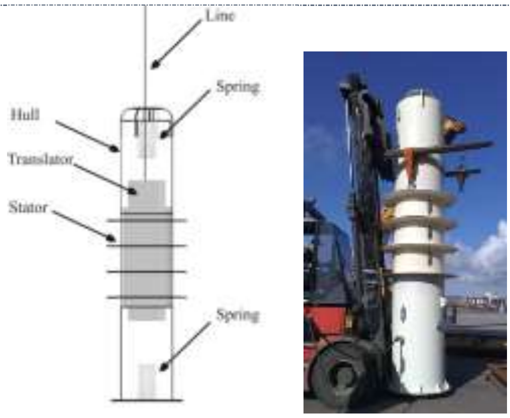

Fig. 1. The linear generator L12

A wave energy converter has to withstand extreme forces, placing high demands on the mechanical design. A solution adopted to dampen the adverse forces is illustrated in Fig. 2, and as shown, it consists in the combination of a spring and two rubber dampers. This study focuses on the dimension of the spring.

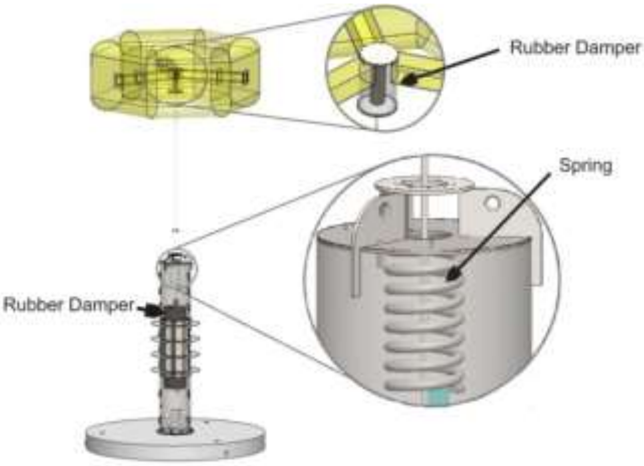

Fig. 2 Design of the damping system

For a correct dimension of the damping system, the forces, the stresses and the reaction time of the different components in the damping system shall be calculated. Ref [8] presents a detailed study of the reaction and delay time of the rubber damper and spring. However, to be able to perform a correct design the mechanical system, those parameters are not enough, the designer requires the amplitude of the maximum force given by the mass and maximum velocity of the moving part, i.e. the translator. Without that parameter, the end stop solution will either be over-dimension, resulting in a higher cost, or under-dimensioned, reducing the life time of the device significant. The authors have not been able to find a similar study for a similar case, so therefore, we here present the theory and the study behind the design of the end stop solution in the linear generator L12.

The direct driven magnetic part of the generator, the translator, follows the motion of the heaving buoy in ocean waves. When the generator operates at no-load, the total force F_{bt} on the translator-buoy-body can be written as a sum of the spring force, F_{spring}, wave excitation force, F_e, hydrodynamic resistance force, F_r, hydrostatic buoyancy force, F_h, and the gravity force, F_{mg}: [9,10,11]

$$F_{bt} = F_e + F_r + F_h + F_{spring} + F_{mg} = a((m_t + m_b)C_a + (m_t + m_b)) \qquad (1)$$

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Where:

$$F_{spring} = kd \quad (2)$$

$$F_h = \rho g A_\omega y \quad (3)$$

$$F_e = F_{Krilov} + F_{Diffraction} \quad (4)$$

$$F_{mg} = g(m_t + m_b) \quad (5)$$

C_a presents the added mass constant, k represents the spring constant, d is the compressed length of the spring inside the top of the generator, A_ω presents the water plane area of the buoy, y is the vertical position of the buoy, the sea water density is presented as ρ, g is the acceleration of gravity whereas m_t and m_b represent the mass of the translator and buoy itself. The mechanical, magnetic and electrical losses are small [12,13] and here neglected.

The directions of the forces are defined in Fig 3.

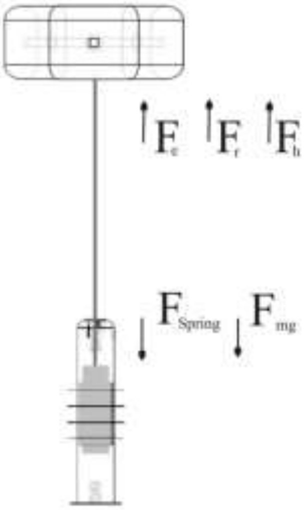

Fig. 3 The introduced directions

The wire connecting the buoy with the translator is considered non-elastic and only heave motion is considered, i.e. buoy and the translator are seen as a combined body with the same vertical velocity.

The excitation force on the buoy can be derived and summed from the normal vector on the wet surface and the water pressure at the specific point, illustrated in Fig 4.

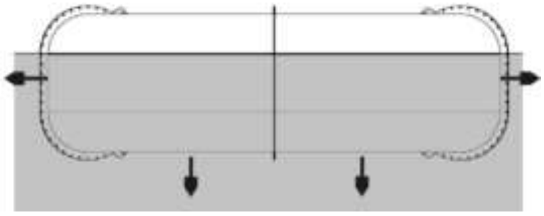

Fig. 4. The excitation force

ISSN: 2277-9655
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As the disturbance, i.e. diffraction of the wave by the partly submerged body is neglected in long waves; the excitation force reduces to the Krilov force, depending on the draft of the buoy. As the dynamic force varies, the buoy either rises or submerges further into the wave, i.e. the Krilov force decreases respective increases with the displacement:

$$F_e = F_{Krilov} = \rho g A_\omega \Delta y(t) \tag{6}$$

where $\Delta y(t)$ is the varying buoy-displacement. The added mass coefficient is taken into consideration and is for a buoy with this geometric property estimated to 0.8. [14]

2. MATERIALS AND METHODS

The direct driven magnetic part of the generator, the translator, follows the motion of the heaving ocean waves. To ensure a stable and proper design of the mechanical system and avoid a great impulse force on the hull, the upper end stop shall be designed to smoothly decrease the velocity of the translator.

An assumption of a sinusoidal motion, *x*, of the wave gives the velocity, $\dot{x}$, of the translator:

$$x = \frac{H_s}{2}\sin(\omega t) \tag{7}$$

$$\dot{x} = \frac{H_s}{2}\omega\cos(\omega t) \tag{8}$$

H_s represents the significant wave height whereas ω states the translator's angular frequency. As the spring compresses, the reaction force, F_{spring}, exposes the buoy for a greater downward force. The buoy submerges further down in the water, presented as $\Delta y(t)$ in Fig 5, and a relative motion between the wave and buoy is achieved, an important parameter when the upper end stop shall be designed. [10]

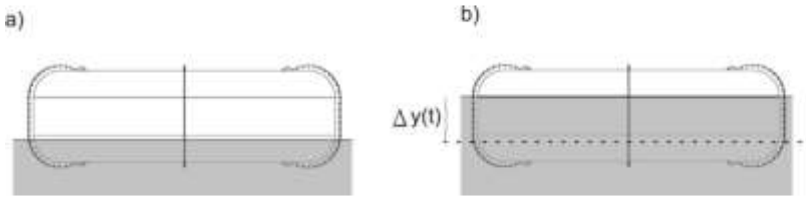

Fig. 5. The partially sunk buoy at two different moments

As the translator reaches the spring, the authors assume a constant velocity of the vertical surface elevation, i.e. the wave's cumulative behaviour is not included in the calculations. Furthermore, the electromagnetic, magnetic and hydrodynamic damping is, as they all increase the damping of the system, ignored.

The geometry of the buoy and the mass of the translator is determined with respect to the energy conversion, i.e. the hydrodynamic input parameters can not be changed. The variables in the system are reduced to the spring constant, the stroke length and the length of the spring. A greater value of the spring constant decreases the length of the spring, presented in Eq 2, giving a longer stroke length of the translator. A high value of the constant gives therefore positive impact on the system, and the authors have designed the system with this in mind. [8]

3. RESULTS

Measured wave data presented in [15] was used as input parameters concluding the amplitude and period of the *wave of the year* during the design process.

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Table 1 presents the system the moment before the translator hits the spring, here presents as p.u.

Table 1. Input parameters

Parameter		Value
F_{spring}	[kN]	0
F_{mg}	[-]	-1 p.u
$F_{hydrostatic}$	[-]	1 p.u
$F_{excitation}$	[kN]	0
$F_{\text{hydrodynamic resistance}}$	[kN]	0
y	[m]	0.76
$V_{translator}$	[-]	1 p.u

As written above, as the translator reaches the spring, the spring compresses and the downward force increases. With the assumption of a constant velocity of the surface elevation after the translator has reached the spring, and the known velocity of the translator, the buoys draft was calculated and utilized to estimate the increasing excitation force, $F_{e,}$ both presented in Fig 6. The results are presented normalized to the initial velocity of the translator.

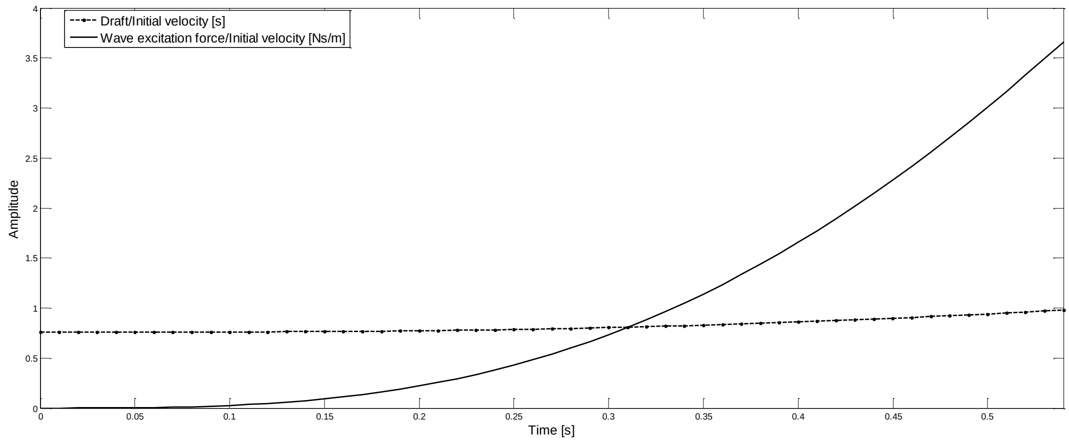

Fig. 6. The increasing draft with the result of an increased excitation force

As expected, the wave excitation force normalized to the initial velocity of the translator increases as the translator reaches the spring. The action time is short, the normalized excitation force reaches a three time higher amplitude within 0,5 seconds.

The behavior of the vertical velocity depends on the acceleration, i.e. it is proportional to the system's total force, $F_{bt,}$ given by Eq (1). With the known velocity of the translator and wave as well as the spring constant, knowledge of the changing excitation force, the stationary gravity- and hydrostatic force, the acceleration and velocity of the translator was calculated and presented in Fig 7, together with the spring compression and the excitation force. The results are presented normalized to the initial velocity of the translator.

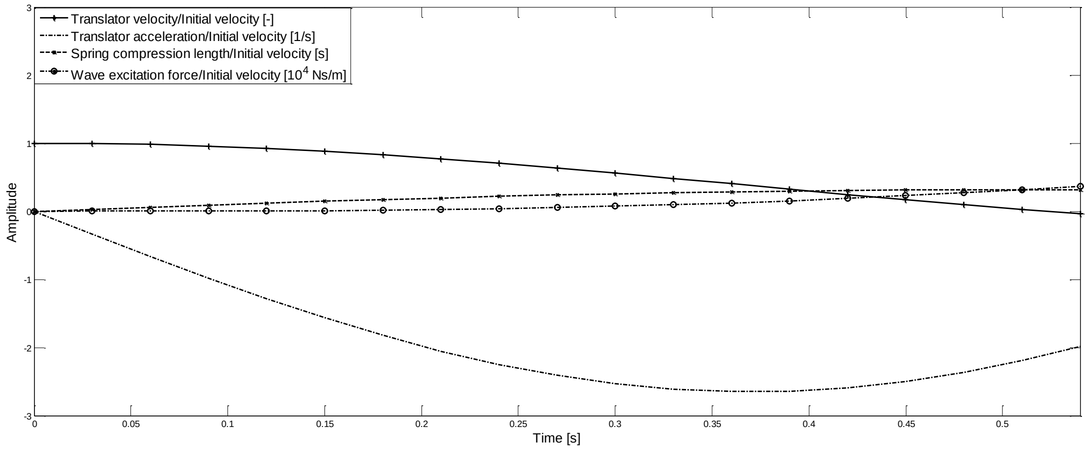

Fig. 7. The resulting scenario

As presented, the chosen available spring constant manage to decrease the translator smoothly to a velocity equal to zero within a compression length of 320 mm, normalized to the initial velocity, when the worst wave's amplitude and period time was utilized as input data. The spring is correct dimensioned for the linear wave energy converter. Again, the action time is short. The translator reaches a velocity equal to zero within 0,55 seconds.

4. DISCUSSION

To ensure a stable and proper design of the mechanical system and avoid a great impulse force in the hull, the upper end stop shall smoothly decrease the velocity of the translator to zero if the wave amplitude is larger than the stroke length. A non-correct design can result in a significant shorter life time of the device, reducing the competitiveness and increasing the uncertainty of the concept. The presented study is therefore of high interest. The input to the model is real wave data and the masses of the translator and buoy. If any of those parameters changes, the designer needs to re-do the study with the possibility to gain other results.

The authors have done a few assumptions. For example, the electromagnetic, magnetic and hydrodynamic damping is ignored, as they both increase the damping of the system, i.e. the study presents a worst case scenario. Further, the added mass of the buoy is not likely to be constant during the deceleration when the translator-buoy system hits the spring in the end of its stroke. The here utilized added mass constant for infinite frequency is a worst case approximation. Experiments of varying added mass during impact of two bodies have been conducted by scientists like Motora et al. [16]

As presented in introduction, a rubber damper is installed on the top of the translator. The original implementation of this component was to avoid steel-to-steel contact as the translator hits the spring, i.e. a noise reduction advice [17], but the implementation shall also act as a damper. However, the damping property of this component is not included in the dimensional calculations.

The rubber damper installed in the buoy is implemented to decrease the so called *snap loads* in the system, i.e. line goes from slack to tense in a short moment creating a large impulse force on the line and its connection points. Its installation is therefore not to damp the translator motion.

5. ACKNOWLEDGMENT

This research was carried out as part of the Statkraft Ocean Energy Research Program, sponsored by Statkraft. (www.statkraft.no) The support is gratefully acknowledged. The authors would also like to thank *The Swedish Research Council*, Grant No **2009-3417,** *Swedish Centre for Renewable Electric Energy Conversion*, *Swedish Governmental Agency for Innovation Systems*, StandUp for Energy strategic government initiative, Swedish Energy Agency, Draka Cable AB, the Göran Gustavsson Research Foundation, Statkraft AS, Fortum, Stiftelsen

Olle Engkvist Byggmästare, Stiftelsen J. Gust Richert, InnoEnergy CIPower, Ångpanneföreningens Forskningsstiftelse, Seabased, Civilingenjörsförbundets Miljöfond, and the Wallenius Foundation for its financial support.

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