韓國海洋工學會誌 第13卷 第4號, pp.159~168, 1999. 11

◉ 論 文

A Fast Analysis on Floating Wave Energy Conversion System

Sang-moo Lee* · Seoung-gun Kim**
(99년 10월 20일 접수)

부유식 파랑 에너지 변환 시스템의 단순화된 열역학적 해석법

이 상 무* · 김 성 근**

Key Words : Wave Energy Conversion(파랑에너지 변환), Floating Chamber(부유식챔버), Oscillating Water Column(부유식 진동수주), Conversion Efficiency(변환효율). Thermal Energy Equation(열역학적 방법)

초 록

부유식 파랑에너지 변환시스템(Oscillating Water Column)에 대한 해석은 입력파와 챔버, 챔버내 공기의 상호작용으로 인하여 어려움이 많다. 이 논문은 이와 같은 요소를 고려하면서도 쉽고 간편한 해석법을 제시한다. 파랑에너지에 의한 자가발전은 파랑에너지를 기계적 운동으로 변환하고 이를 전기에너지로 변환함으로써 가능하다. 본 논문은 파랑에너지에서 기계적 에너지로 변환하는 과정에 집중하여 그 부분의 성능을 해석한다.

단일 진동수 규칙파가 입력되었을 때에 파에 의하여 챔버의 상하운동이 선형적으로 발생하는 것으로 보며, 이 상하운동에 챔버내의 압력 영향을 고려하였다. 상하운동과 챔버내로 투과한 파, 그리고 챔버내 압력에 의해 발생되는 파에 의해 챔버내의 상대운동을 정하고, 그 상대운동에 의한 공기의 압축 팽창과 온도상승을 근사적 열역학적 방정식으로 해석하였고 오리피스를 통한 유량을 결정하였다. 얻어진 식은 간단하면서도 관련요소의 영향을 전반적으로 표현한다. 결과에 따르면 고정식의 에너지 변환식은 부유식의 특별한 경우로서 파악되었다. 또한 고정식의 시스템을 그대로 부유식으로 바꿨을 때 그 변환효율은 적어지는 것으로 나타났다. 본 해석법은 계산이 간편하므로 설계단계에서 유용하게 활용될 수 있을 것으로 기대된다.

1. Introduction

The ocean wave always prevails on earth's surface. And human beings have long thought of its utilization. Recent industrial advancement requires the use of clean energy instead of the fossil energy, which associated with global warming. With no reliable single source of clean

* Dong Yang University, Yongju, Korea
** Marine Engineering Lab. Cheju National University, Cheju, Korea

energy, diverse energy sources are under study for development. Ocean wave energy is one of these sources. Though the total amount of the wave energy being large, concentration of the energy varies much with time and the efficient and reliable mechanism for conversion is not well developed yet.

Wave Power Devices have been developed over the last three decades[1]. A few European countries, such as Sweden, the UK, Denmark, Norway, and Ireland, have already launched wave power plants[2]. Asian countries, including Japan, China, and India, have installed pilot plants or a few hundreds kilowatt plants[3]. In Korea feasibility studies and experimental projects have been done by various institutes[4], but no pilot plant has yet been launched. However wave energy is scattered all over the water surface of the earth. The amount of potentially extractable energy is quite large, but its utilization needs arduous work for a long time. As an example, the KAIMAI project in Japan[5], which tested a floating-type device for four different kinds of turbines, was performed over ten years. Based on this test a device of the Backward Bent Duct Buoy(BBDB) was suggested and showed low drag and better performance. Depending on the scale of BBDB, its application is expected to navigation aids, open sea buoys for telemetering, as an underwater warning device, for fish gathering and station keeping, or large-scale power suppliers. These applications are common to floating-type wave power devices. Recent applications extend to desalination, which is not affected by power fluctuation.

Floating-type devices have other advantages: They can be installed at any place in the ocean as long as enough wave power persists; They can be utilized as wave dissipation devices connected multitudinously; and they are relatively free from the tidal effect contrast to the onshore devices. Since the offshore water depth is comparatively higher than onshore, wave breaking is less and more energy can be transformed into electricity.

From a modelling viewpoint, considerable progress has been made so that reasonable estimates of applicability can be provided before a pilot launch. This has been done by extending the existing concepts into more realistic models, so that the degree of approximation is reduced[6]. Special features of the individual device are emphasized. However, Oscillating Water Column(OWC) devices are mainly focussed in wave power applications.

Analysis and modelling of OWC devices can be divided into two stages. The first is the system modelling of energy transformation from the wave to the air flow through the orifice of the wells-turbine; and the second is the energy transformation from the air flow of the wells-turbine to electrical power. The latter needs a turbine model, an induction generator model, and a circuit design[7][8]. The former is related to the marine hydrodynamics and gas dynamics. Though marine hydrodynamics contributes theory and experiments in modeling, the interaction between water elevation and air in the chamber is not well modeled. For a fixed device, Takahashi[9] suggested conversion efficiency from the wave to air flow of the orifice. He used the approximation of the gas equations in the chamber. The results is nonlinear to wave amplitudes. For a floating device, the reference[4] assumed that air flow at the orifice be linear to the pressure change. Assuming the chamber motion is independent from the internal pressure, Kim and Park[10] considered the relative motion of the OWC ceil and the internal wave elevation.

The present paper presents a quick analysis technique that provides overall efficiency of the conversion mechanism. Introducing the linear motion dynamics of the floating chamber to the approximated nonlinear thermal gas equation in the chamber, an explicit form of the governing equation is derived and is fedback to the simplified wave dynamics. The final equation is nonlinear with respect to the incident wave amplitude. Though much simplification has been done, the technique takes into account the important features of the mechanism, such as interaction of the incident wave, diffraction, compression of the air in the chamber, and the dynamics of the chamber and orifice.

2. Theoretical Derivation of the Governing Equation

The analysis procedure is described below. The governing equation for the gas in the chamber is derived from mass continuity and the thermal equations. These equations are approximated for small variation of chamber volume. The chamber motion is modelled as a linear dynamic system, excited by the incident wave and the internal pressure change. The equations are expressed in complex form and solved to give conversion efficiency.

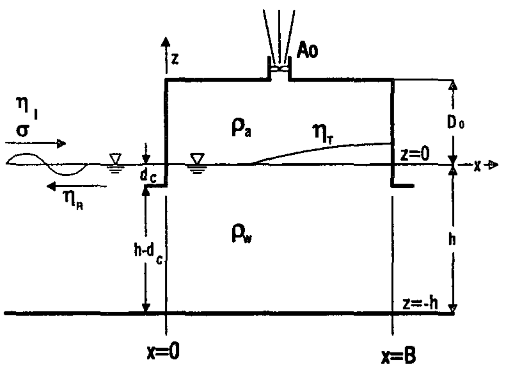

Fig. 1 Definition sketch

2.1 Thermal Equation of the Air Chamber

The continuity equation for the air mass in the chamber is

$$\frac{d(\rho_a V)}{dt} + \dot{m} = 0 \tag{1}$$

where ρ_a is the density of the air in the chamber, V is the volume, and $\dot{m}$ is the rate of the air mass flux through the nozzle. The air mass flux is linearly proportional to the air velocity w_p at the nozzle as

$$\dot{m} = \rho_a c_d \varepsilon A_w w_p \tag{2}$$

where c_d the contraction coefficient, ε the orifice contraction ratio (A_0/A_w), A_w the free surface area of the water column, and A_0 the area of the orifice. The air velocity, w_p , is determined in terms of the temperature difference[9] as

$$w_p = \beta\phi\sqrt{2c_p|T_a - T_o|} \tag{3}$$

where $\beta = 1$ for outflow and -1 for inflow, T_0 is the absolute temperature of the open air, c_p is specific heat at constant pressure (=1005 J/kgK), and φ is the velocity coefficient of air nozzle. Substitution of the gas equation $\rho_a = p_a/RT_a$ to Eq. 1 yields the flow rate equation as

$$\frac{d}{dt}\left(\frac{p_a V}{RT_a}\right) + \dot{m} = 0 \tag{4}$$

where R is the gas constant (=287.2 J/kgK for air). The conservation of energy in the air chamber is described as

$$p_a \frac{dV}{dt} + c_v \frac{d}{dt}\left(\frac{p_a V}{R}\right) + \dot{m} c_p T_a = 0 \tag{5}$$

where c_v is the specific heat at constant volume (=717.1 J/kgK).

The equations presented above are the fundamental equations of the thermodynamics without any assumption of adiabatic change. However, it is not easy to find the general solution to the equations. For simplicity we assume that the incident harmonic wave has a single frequency, and that the motion response of the ocean water column is also linear to the wave. Further simplification is applied to the volume change, which is a small variation with respect to the total volume so that we can approximate the thermal gas equation.

2.2 Approximation of the Equations

Since Eqs. 4 and 5 are non-linear, it is not easy to find a full solution. To get an approximate solution we assume that the incident wave has a single frequency, the variations of V, T_a and p_a with respect to time t are quite small, and they can be expressed by the following sinusoidal equations as

$$V^* = V - V_0 = -a_2 A_w \sin(\sigma t + \phi_2) \quad (6)$$

where V^* the oscillatory fraction of V, σ the angular velocity of the incident wave, and a_2 and ϕ_2 the relative motion amplitude and the phase of the chamber to the internal water surface elevation. This relative motion of the floating structure, which is newly introduced in this paper, adds more difficulties in the analysis than the one of fixed case. It is assumed that a_2 is much less than the depth of the air chamber D_0. The consequent variations of temperature and pressure in the chamber are presented with phase shift ϕ_p as

$$T^* = T_a - T_O = \alpha T_O \sin(\sigma t + \phi_2 + \phi_p) \quad (7)$$

$$p^* = p_a - p_o = \nu p_o \sin(\sigma t + \phi_2 + \phi_p) \quad (8)$$

where α and ν are small compared to 1.0. To get an approximation to α and ν, Eqs. 6, 7, and 8 are substituted into Eqs. 4 and 5. The values of α, ν, and ϕ_p are determined by putting $\sigma t + \phi_2 = 0$ or $\pi/2$ in terms of the relative amplitude a_2 and the frequency σ as

$$\alpha = (\gamma - 1)\frac{a_2}{D_0}\cos\phi_p \quad (9)$$

$$\nu = \gamma\frac{a_2}{D_0}\cos\phi_p \quad (10)$$

$$\cos\phi_p = \pm\sqrt{1+K^2} - K \quad (11)$$

where

$$K = (c_d \phi \varepsilon)^2)(\gamma - 1)c_p \frac{T_a}{(\sigma D_o)^2}\frac{D_o}{a_2} \quad (12)$$

$$\gamma = c_p / c_v \quad (13)$$

It is noted that the above equations are nonlinear with respect to the relative amplitude.

2.3 Incident Wave and Air Response in the Chamber

Assuming the incident waves η_I, reflected waves η_R in the seaward side, and standing waves η_T in the chamber are expressed as

$$\eta_I = a_I \sin(\sigma t - kx - \phi_I) \quad (14)$$

$$\eta_R = a_R \sin(\sigma t + kx - \phi_R) \quad (15)$$

$$\eta_T = a_T \cos k(x - B)\sin(\sigma t) \quad (16)$$

where a_I, a_R, and a_T are the amplitudes, ϕ_I and ϕ_R are the phase differences, and k is the wave number. The coordinate system is shown at Fig. 1, where x is the incoming wave direction to the water column, and B is the breadth of the column.

The continuity of the total volume of the water flow through the wave entrance at x=0 leads to

$$a_I \sin(\sigma t - \phi_I) + a_R \sin(\sigma t - \phi_R) = a_T \sin kB \cos \sigma t \qquad (17)$$

The continuity of the total pressure across x=0 gives

$$a_I \sin(\sigma t - \phi_I) + a_R \sin(\sigma t - \phi_R) = \frac{1}{\rho_w g} f(kh, kd_o) p^* + a_T \cos kB \cos \sigma t \qquad (18)$$

where h is the water depth at the site of the water column, d_0 is the draft of water column, and

$$f(kh, kd_o) = \frac{2 \cosh kh \sinh k(h - d_o)}{\cosh k(h - d_o) \sinh k(h - d_o) + k(h - d_o)} \qquad (19)$$

Neglecting the radiation wave due to the column motion, the relation of the air response in the chamber to the incident waves is obtained by adding Eq. 17 and 18 as

$$2a_I \sin(\sigma t - \phi_I) = \frac{1}{\rho_w g} f(kh, kd_o) p^* + a_T(\sin kB \cos \sigma t + \cos kB \sin \sigma t) \qquad (20)$$

The effective amplitude a_0 of the wave surface oscillation in the chamber is given by a_T as

$$a_o = \frac{\sin kB}{kB} a_T \qquad (21)$$

2.4 Motion of the Air Chamber

The floating water chamber, positioned by mooring cables, is subject to the incident waves. The six degrees of device motion is affected by the internal air pressure as well as the incident wave. For simplification, only the heave motion is considered as the interaction and the motion response is linear as

$$m_H \ddot{z} + c_H \dot{z} + k_H z = a_I k_H r_f \sin(\sigma t - \phi_I + kB/2) + p^* A_w \qquad (22)$$

where m_H is the mass of the chamber including the hydrodynamic mass effect, z is the heave motion of the chamber, c_H is hydrodynamic damping coefficient for heave, $k_H = \rho_w g A_H$ is the heave static restoring coefficient, g is gravitational acceleration, A_H is the cross sectional area of the chamber at the waterplane, and r_f is the factor of the wave shape to the exciting force. The heave exciting force due to the incident wave is evaluated with respect to the wave elevation at the middle, which is represented by the phase difference $kB/2$. The second term in the right hand side in Eq. 22 represents the effect of the internal air pressure. To a wave with frequency σ, the response is given as the particular solution to Eq. 22. Using the relation Eq. 8 and Eq. 10, the solution to Eq. 22 is rewritten as

$$z = \Im\{a_I \xi_o r_f \exp(i(\sigma t - \phi_I + kB/2 - \phi_H))\} + \Im\left\{\xi_o \Lambda a_2 \frac{A_w}{A_H} \cos\phi_p \exp(i(\sigma t + \phi_2 + \phi_p - \phi_H))\right\} \qquad (23)$$

where

$$\xi_o = \frac{\rho_w g A_w}{\sqrt{m_H^2(\sigma_0^2 - \sigma^2)^2 + \sigma^2 c_H^2}} \qquad (24)$$

$$\phi_H = \tan^{-1}\left(\frac{\sigma c_H}{m_H(\sigma_0^2-\sigma^2)}\right) \tag{25}$$

$$\sigma_o = \sqrt{\frac{k_H}{m_H}} \tag{26}$$

$$\Lambda = \frac{\gamma p_o}{\rho_w g D_o} \tag{27}$$

and $\Im$ denotes the imaginary part of the complex quantity and i denotes $\sqrt{-1}$. The motion response is expressed in terms of the amplitude and phase of the relative motion, which is to be determined by solving the thermal equation, the wave motion and the motion equation of the chamber, simultaneously.

2.5 Solving the Governing Equations

Now we come back to the kinematic relation of the air volume, which is defined by the motion difference of the chamber and internal wave. The relation is written from Eq. 6 in complex form as

$$\Im\{-a_2 A_w \exp(i(\sigma t+\phi_2))\} = \Im\{-A_w a_o \exp(i\sigma t)\} + zA_w \tag{28}$$

Since Eq. 28 holds for arbitrary time, the equality holds for the real part, too. We can drop out the imaginary notation without loosing generality. Using Eqs. 23 and 28, the relative displacement a_2 affected by the device motion is obtained in terms of the incident wave and internal surface elevation as

$$a_2\exp(i(\sigma t+\phi_2)) = \Theta^{-1}[a_o\exp(i(\sigma t)) - a_I\xi_o r_f\exp(i(\sigma t-\phi_I+kB/2-\phi_H))] \tag{29}$$

where

$$\Theta = 1+\xi_o\Lambda\frac{A_w}{A_H}\cos\phi_p\exp(i(\phi_p-\phi_H)) \tag{30}$$

The effect of the chamber motion appears in two ways. The effect due to incident wave is presented in Eq. 29, while the one due to the interaction is presented in Eq. 30. When the water chamber is fixed, Eq. 30 is unitary, and when not fixed, it is not unit anymore.

From Eqs. 8, 10, 29, 30, 20, and 21 the equation for the internal wave elevation is obtained as

$$\frac{a_I}{a_o}\exp(-i\phi_I) = \left[2+\Theta^{-1}\xi_o r_f\Lambda f\cos\phi_p\exp(i(\phi_p+kB/2-\phi_H))\right]^{-1} \left[\Theta^{-1}\Lambda f\cos\phi_p\exp(i\phi_p)\frac{kB}{\sin kB}\exp(ikB)\right] \tag{31}$$

Eq. 31 gives a nonlinear relationship between the incident and internal wave amplitudes. When the amplitude a_I and the frequency σ of the incident wave is given, the phase shift ϕ_I and the effective amplitude a_0 are numerically evaluated using Eqs. 13 and 31 by iteration. And then the relative motion from Eq. 29 and the chamber motion from Eq. 23 are calculated.

2.6 Air Power and Efficiency

The work that is done to the air mass in the chamber for small time dt is given by

$$W_a\,dt = p^* dV^* \tag{32}$$

The extracted air power averaged over one wave period $\overline{W_a}$ is

$$\overline{W_a} = -\frac{1}{T}\int_o^T p^* dV^* \tag{33}$$

where T is the wave period. From Eqs. 6, 8, and 11, the extracted power is

$$\overline{W_a} = \frac{1}{2}\frac{\gamma p_o\sigma A_w}{D_o}a_2^2\sin\phi_p\cos\phi_p \tag{34}$$

It is noted that the extracted power is dependent upon the relative motion between the chamber and the internal wave elevation, specifically the relative amplitude and phase. The design of a turbine requires tuning so that it maximizes Eq. 34.

For periodic waves, the input wave power $\overline{W_i}$, or the rate of energy transport, per unit wave crest width is

$$\overline{W_I} = \frac{1}{2}\rho_w g a_I^2 C_g \tag{35}$$

where

$$C_g = \frac{1}{2}\frac{\sigma}{k}\left(\frac{1+2kh}{\sinh kh}\right) \tag{36}$$

is the group velocity.

The efficiency of power conversion from a wave to air is calculated as the ratio of the averaged extracted power to the wave power, which is

$$Eff_o = \overline{W_a}/\overline{W_I}l_B = \frac{\gamma p_o}{\rho_w g D_o}\frac{B\sigma}{C_g}\frac{a_2^2}{a_I^2}\sin\phi_p\cos\phi_p \tag{37}$$

where l_B is the crestwise length of the chamber. The conversion efficiency, Eq. 37, includes the effect of the chamber motion and adiabatic change of the air, while the radiation wave due to the chamber motion is not included. However, one can easily extend the present procedure to include six degrees of freedom motion of the chamber.

3. Theoretical Discussion and Numerical Results

Based on the theoretical results, Eq. 31, a few factors are discussed. Then the effect of the dynamics of the device is discussed, which is the main concern of this paper.

3.1 Theoretical Discussion

By studying the extreme cases, we can find the meanings of the results. To find if it confirms the fixed case that is presented by many authors, we can put $\lim_{\xi_0\to 0}$ as if the device is held forcedly. The limiting case of Eq. 31 becomes

$$\lim_{\xi_o\to 0}\frac{a_I}{a_o}\exp(-i\phi_I) = \frac{1}{2}\left[\Lambda f\cos\phi_p\exp(i\phi_p) + \frac{kB}{\sin kB}\exp(ikB)\right] \tag{38}$$

which exactly reproduces the main result of the reference[10]. In this case the conversion efficiency is obtained by putting $a_2 = a_0$ according to Eq. 28.

3.2 Numerical Results

Since this paper extends the conversion analysis to floating-type, main concern of the numerical study is the effect of the dynamics of the device. The numerical computation is carried out on a chamber of which dimensions[11] are shown in Table 1. The system efficiency is computed when it is fixed and floated. The variation of the heave damping coefficient of the system is investigated. The nonlinear effect of the wave amplitude is showed.

Table 1 The Dimension of the Slopped Front Wall Caisson

length	(l_B)	0.28m	$\varepsilon = 0.01$
breadth	(B)	0.08m	
height	(D_0)	0.2m	
front wall depth	(d_c)	0.02m	
wave amplitude =	(a_I)	0.01, 0.015m	
water depth	(h)	0.16	

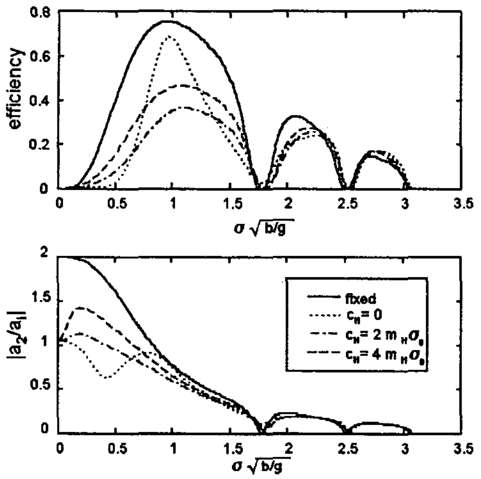

Fig. 2 Conversion efficiency and the relative motion

The computational results are shown in Fig. 2, Fig. 3, and Fig. 4. The overall efficiency is compared for floating and fixed structures with wave amplitude $a_I = 0.01m$ in Fig. 2. The heave damping coefficient varies from $c_h = 0$, critical damping $2m_H\sigma_0$, and to twice the critical damping $4m_H\sigma_0$. The efficiency at the nondimensional frequency around 1 is of concern in the design of the wave energy conversion mechanism. It shows that the wave energy conversion efficiency of the floating chamber is less than that of the fixed one in their peak, irrespective to the heave damping. Though the heave damping is zero, the efficiency is still less than that of the fixed. It also shows that when the heave damping is low, the conversion mechanism is efficient at the narrow band around the natural frequency. As the heave damping increases the efficiency decreases first and then increases again, and the bandwidth becomes broad.

It is not necessary to underestimate the benefit of the floating-type conversion mechanism. The present example is only for numerical comparison purposes and validates the present theory, showing the difference between the floating type and fixed one. It is possible that other designs of the chamber may have different characteristics. The computation is performed to cover a wide range of frequency, which is not needed in a real situation. In actual design stage, only the finite range of frequency is concerned in which the wave energy is concentrated.

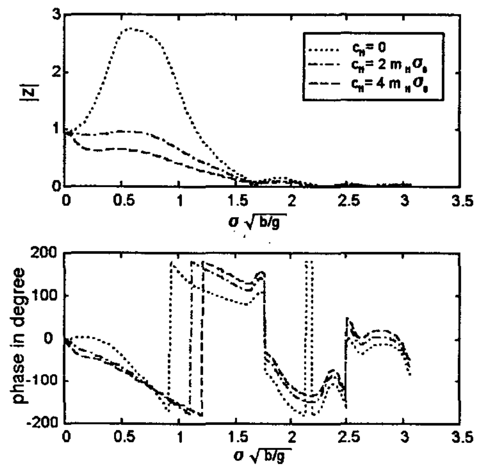

Fig. 3 The motion response of the chamber

As the frequency goes to zero, as shown in Fig. 2, the relative motion of the fixed structure approaches 2, which is twice the incident wave amplitude, due to the reflected wave. The relative motion of the floating chamber approaches 1, because the chamber moves up and down with the internal wave. Though the dynamics system is undamped, the chamber motion does not show sharp peak around the natural frequency, as shown in Fig. 3. This is due to the interaction of the gas dynamics that acts as damping.

To investigate the dependency of the conversion efficiency on the incident wave amplitude, the different amplitudes of $a_I=0.01$ and $a_I=0.015$ are under study. The results in Fig. 4 show that the efficiency and the relative motion are reduced as the amplitude increases. The efficiency difference is about 10% at the peak.

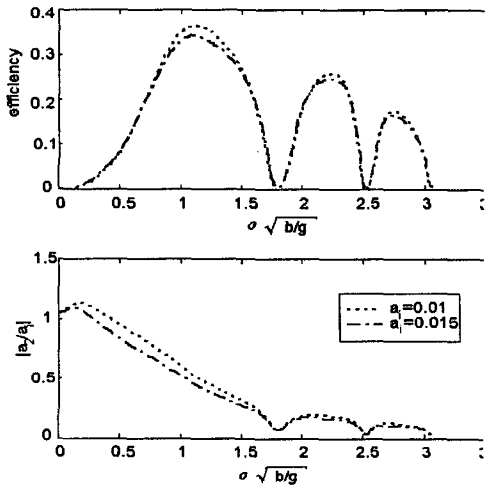

Fig. 4 The effect of the nonlinearity with respect to the wave amplitude

3. Conclusion

This paper proposes a fast and reasonable analysis on the floating wave energy conversion system. The nonlinearity of the gas equation is included in the analysis approximating small variation. Introducing the linear motion dynamics of the floating chamber to the approximated nonlinear thermal gas equation in the chamber, an explicit form of governing equation is derived and is fedback to the simplified wave dynamics. The final equation is nonlinear with respect to the incident wave amplitude. Iterative solutions are obtained and numerically validated. This analysis includes, as a special case, the solution for a fixed structure.

The efficiency of the floating chamber as a wave energy conversion system is generally less than that of the fixed structure. This analysis can be used in design a floating wave conversion system. Further research is recommended to include the wave radiation and diffraction, etc., and the six degrees of freedom motion.

References

1) Duckers, L.,"Water Power-wave, Tidal and Low-head Hydro Technologies", Power Engineering Journal, Vol.9 No.4, pp. 164 -172, Aug., 1995
2) Nielsen, K., "Wave Energy Plants-The Way Forward", IEE Colloquium on Wave Power: An Engineering and Commercial Perspective (Digest No: 1997/098), pp. 7/1 -7/5, 1997
3) Bregman, R., Knapp, R.H. and Takahashi, P.K., "Design Considerations for Ocean Energy Resource Systems", OCEANS '95. MTS/IEEE. Challenges of Our Changing Global Environment. Conference Proceedings, Vol. 2, pp. 1084 -1091, 1995
4) Hong, Do-chun et al., "Development of Electrical Power Plant(60kw) from Ocean Wave Energy(I)", Report of Korea Electric Power Research Institute, KRC-93G-T17, 1994
5) Masuda, Y., Yamazaki, T., Outa, Y. and McCormick, M.E., "The Backward Bend Duct Buoy-An Improved Floating Type Wave Power Device", Proceedings of OCEANS '88. A Partnership of Marine Interests, Vol.3 pp. 1067 -1072, 1988
6) Thomas, G.P., "Mathematical Modelling of Wave Power Devices", IEE Colloquium on Wave Power: An Engineering and Commercial Perspective (Digest No: 1997/098), pp. 2/1 -2/7, 1997

7) Tindall, C.E. and Mingzhou Xu, "Optimising a Wells-turbine-type Wave Energy System", IEEE Transactions on Energy Conversion, Vol. 11, No.3, pp. 631 -635, 1996

8) Huang, F. and Beattie, W.C., "Modeling and Simulation of the Islay Wave Power Conversion System", IEEE International Electric Machines and Drives Conference Record, pp. MB2/2.1 -MB2/2.4, 1997

9) Takahashi, S. "A Study on Design of a Wave Power Extracting Caisson Breakwater", Port and Harbour Institute of Japan, Wave Power Laboratory, 1988

10) Kim, Seoung-gun, and Park, Ro-sik., "A Study for Design Method of Platform for Floating-Type Wave Energy Conversion and Wave Dissipation", Proceedings of the Fourth(1994) International Offshore and Polar Engineering Conference, Osaka, Japan, April, 1994

11) Kim, Seoung-gun., "An Experiment and Numerical Analysis of Wave Energy Conversion Device of Front-sloped Caisson Type", Bulletin of the Marine Research Institute, Vol.21, Cheju National University, 1997