Hydroelastic Motion and Drift Force of Twin-hull Very Large Mobile Offshore Structure

by
Ken Takagi
Department of Naval Architecture and Ocean Engineering, Osaka University
2-1 Yamadaoka, Suita, Osaka 565-0871, Japan
Email takagi@naoe.eng.osaka-u.ac.jp

Introduction

We proposed a concept of floating wind power plant, which is composed of slender beams and lower-hulls so that the structure is constructed in lightweight possible for its propulsive performance. The structure is advancing with sails and thrusters under the lower-hull, and it is navigated so that wind turbines are in service at beam wind. Many vertical struts are equipped on lower-hulls, which induce a lateral lift force to counter the wind drag force. We called this structure VLMOS (Very Large Mobile Offshore Structure).

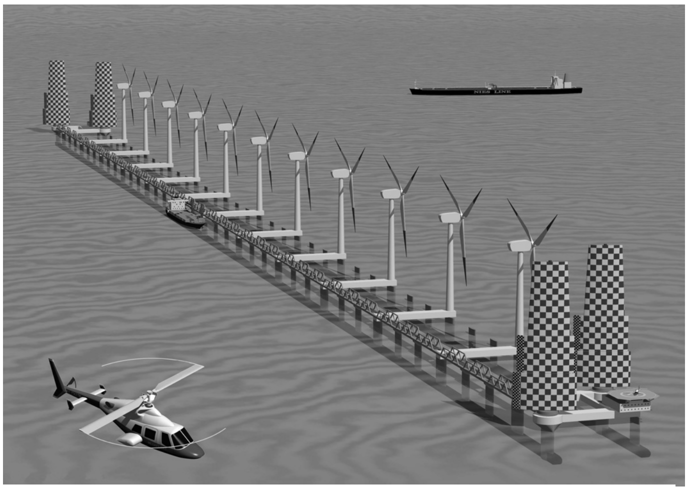

Fig.1 Image of the twin-hull VLMOS drawn by National Institute of Environmental Studies

We have been working on the feasibility study about this concept in recent three years. I showed an initial design of the structure and its hydroelastic analysis at the last workshop. The initial design of the structure has four lower hulls and wind turbines are set in two lines. However, we found that the generated power by the wind turbine in the second line is 50%-60% of the first line, when the distance between two lines is quadruple of diameter of the turbine. Although multi-lines arrangement of the wind turbine is an appealing design for a good maneuverability of the structure, we gave up this design because of bad efficiency of the wind turbine.

This year, we are investigating a single-line arrangement of the wind turbines.Fig.1 shows an image of the new design. The overall length of the structure is 2,160m, the width is 64m and the draft is 20m. The wind turbines are supported by two lower-hulls. The lower hulls are connected with vertical struts and transverse beams. The strut induces the lateral force to counter the wind drag force as stated above, and it also plays an important roll on the stability of the structure.

It is supposed that the hydroelastic behavior of the new design in waves is different form that of the multi-hull design. The hydroelastic motion is affected by many factors such as the static stability, elasticity of the lower-hull, shape of the lower-hull and so on. So, we decided to perform a simple analysis to know the basic hydroelastic behavior of the structure. Another concern about this design is the drift force in waves. Area of the strut is decided so that the induced lift force can exceed the wind drag force. In order to keep a sufficient area, the chord length of the strut becomes longer than the multi-hull design, and it is expected that the drift force becomes larger. So, it is important to know the relation between the chord length and the drift force. These two topics are discussed in this paper.

Hydroelastic Motion

In order to know the basic hydroelastic behavior of the structure, it is important that the numerical method is not time consuming to cover the wide range of frequencies and directions of the incident wave, while the numerical results should be accurate. Although, we have a three-dimensional computer code based on the pFFT method as presented at the last workshop (Takagi and Noguchi [2005]), we decided to use a simple method. The following assumptions are employed.

- □ The length of the structure is infinite, and the effect of the strut for the boundary value problem is neglected. Thus, the problem is periodic in the longitudinal direction, and all properties vary sinusoidally.
- □ Static stability is estimated from the water line area of the strut.
- □ The strut and the transverse beam are assumed to be rigid.

Cartesian coordinate system is defined so that the z axis is vertically upward and the $x-y$ plane coincides with the calm water. The x axis is parallel to the lower-hull. The direction of the incident wave has an angle χ with respect to the x-axis, and when $\chi = 0°$ the wave direction is .the following sea. The problem is assumed to be harmonic in time with the circular frequency ω. Since the water depth is infinite, the wave number of the incident wave is $k = \omega^2 / g$ where g is the gravitational acceleration.

Since the problem is sinusoidal in x-direction, the velocity potential can be written as

$$\Phi_L(x,y,z,t) = \Re\left[\phi_L(y,z)e^{i(\alpha_0 x + \omega t)}\right] \tag{1}$$

where $\alpha_0 = k\cos\chi$. $\phi_L(y,z)$ satisfies

$$\frac{\partial^2\phi_L}{\partial y^2} + \frac{\partial^2\phi_L}{\partial z^2} - k^2\phi_L = 0, \quad \text{in the fluid,} \tag{2}$$

$$k\phi_L + \frac{\partial\phi_L}{\partial z} = 0, \text{ at } z = 0 \tag{3}$$

$$\frac{\partial\phi_L}{\partial\nu} = \sum_{j=1}^{3} i\omega\xi_j\nu_j, \text{ on a lower-hull,} \tag{4}$$

where ξ_j is the amplitude of $j-$th mode, (ν_1,ν_2) is the normal vector inward to the fluid and $\nu_3 = y\nu_2 - z\nu_3$. $j=1$ is sway, $j=2$ is heave and $j=3$ is roll.

The solution can be obtained by solving a boundary integral equation with a Green function, which satisfies (2) and (3). The numerical calculation of the Green function is performed with an accurate and quick method proposed by Kashiwagi et al [1994]. Since the distance between two lower-hull is wide, the wave interaction between them is evaluated by employing the wide spacing approximation.

The effect of elasticity is simple, since a sinusoidal deformation is assumed. Equations of motion are obtained as

$$\left\{-\omega^2(m + A_{11}) + i\omega B_{11} + \alpha_0^4 EI_y\right\}\xi_1 + \left\{-\omega^2(m + A_{13}) + i\omega B_{13}\right\}\xi_3 = e_1 \tag{5}$$

$$\left\{-\omega^2(m + A_{22}) + i\omega B_{22} + C_{22} + \alpha_0^4 EI_z\right\}\xi_2 = e_2 \tag{6}$$

$$\left\{-\omega^2 A_{31} + i\omega B_{31}\right\}\xi_1 + \left\{-\omega^2(I_{xx} + A_{33}) + i\omega B_{33} + C_{33} + \alpha_0^2\Gamma\right\}\xi_3 = e_3 \tag{7}$$

where A_{ij} is the added mass, B_{ij} is the damping coefficient, C_{ij} is the restoring force due to the static pressure, ξ_i is the amplitude of motion, e_i is the wave exciting force, m is the mass of the structure, I_{xx} is the moment of inertia, EI_x and EI_y are the rigidity of the structure in y direction and z direction respectively, Γ is the torsional rigidity.

Fig. 2 shows an example of RAOs in

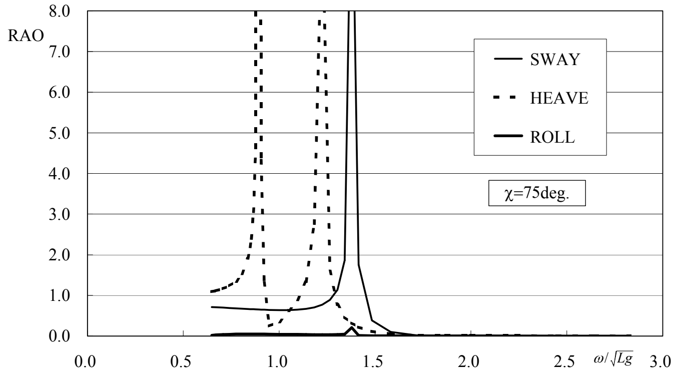

Fig.2 RAOs of the twin-hull VLMOS

oblique wave. Since the wave number k is proportional to square of the circular frequency ω, the real part of the coefficient of the sway motion is simplified as $-\omega^2 M + \omega^8 E \cos^4 \chi$, where M denotes the mass term and E denotes the elastic restoring force. This means the sway motion has a resonant frequency, and it varies quickly as the direction of the incident wave varies. Similarly the real part of the heave coefficient is simplified as $-\omega^2 M + \omega^8 E \cos^4 \chi + C$, where C denotes the restoring force due to the static pressure. It is apparent that the heaving motion has two resonant frequencies. The resonance due to the buoyancy occurs at low frequency (long wavelength) and the other resonance due to the elasticity occurs at high frequency. The resonant frequency due to the elastic restoring force varies quickly as the direction of the incident wave varies, but the resonant frequency due to the buoyancy does not vary appreciably with the change of wave direction. Since, the real part of the roll coefficient can be written in the same form as that of the heaving motion, the rolling motion has two resonant frequencies.

We are performing the tree-dimensional analysis, and the results will be shown at the workshop together with other results by the simple analysis.

Drift Force

It is supposed that the wave drift force is mainly induced with the wave scattering by the strut. We assume that the wave scattering by the lower-hull is negligible for the shake of simplicity. Other simplifications are that the strut is infinitely deep, the thickness of the strut is negligible, the motion of the strut is ignorable and the structure is infinitely long.

Suppose a periodic in line array of struts whose chord length is $2c$ and the drought is infinity with gaps $2a$ and distance d between their centers. Cartesian coordinates are chosen with the struts occupying $y=0$, $-\infty < z < 0$, where $x-y$ plane coincides with the calm water surface. A wave is incident upon the array from $y<0$ with the angle χ. The wave number of the incident wave is k and we write

$$\alpha_0 = k\cos\chi; \quad \beta_0 = k\sin\chi. \tag{8}$$

Since the draft of the strut is infinite, the depth dependence can be extracted out and the linearized velocity potential can be written as

$$\Phi_S(x,y,z,t) = \Re\left[\phi_S(x,y)e^{kz+i\omega t}\right]. \tag{9}$$

$\phi_S(x,y)$ satisfies

$$\frac{\partial^2\phi_S}{\partial x^2} + \frac{\partial^2\phi_S}{\partial y^2} + k^2\phi_S = 0, \quad \text{in the fluid,} \tag{10}$$

$$\frac{\partial\phi_S}{\partial y} = 0, \quad \text{on a strut.} \tag{11}$$

Because of the periodicity of the struts the field at a point $x+d$ differs from that at x by a factor $e^{i\beta_0 x}$, being the change in phase of the incident wave. Thus we may write

$$\phi_S(x+md,y) = e^{im\alpha_0 d}\phi_S(x,y), \quad m = 0,\pm1,\pm2,\cdots \tag{12}$$

and we need only consider a single strip such as $x\in\left[-\tfrac{1}{2}d,\tfrac{1}{2}d\right], -\infty<y<\infty$ with (8) providing the extension of the solution to the whole plane. It follows from (12) that we may write

$$\phi_S(x,y) = e^{i\alpha_0 x}\psi(x,y) \tag{13}$$

where $\psi(x,y)$ is periodic in x with period d. Therefore, the most general form for $\phi(x,y)$ which ensures that (13) is satisfied is

$$\phi_S(x,y) = \sum_{n=-\infty}^{\infty} A_n e^{-i(\alpha_n x - \beta_n y)}, \tag{14}$$

where

$$\alpha_n = \alpha_0 + \frac{2n\pi}{d} = k\left(\cos\chi + \frac{\lambda n}{d}\right) \quad \text{and} \quad \beta_n = \pm\left(k^2 - \alpha_n^2\right)^{1/2} \tag{15}$$

to satisfy (10). It is apparent that reflected and transmitted waves exist at $|y| \to \infty$ if $k > |\alpha_n|$, i.e. β_n is a real number. When $d < \lambda/2$, a single reflected and transmitted wave is observed at the infinity. Other terms present evanescent waves, which disappear at the infinity. In general the number of real β_n depends on the value of λ/d, and we can easily estimate the range $-r \le n \le s$ for the real β_n. Since we need only the drift force, coefficients of A_n for $n = -r, \cdots, s$ are of particular interest being amplitudes of the reflected waves.

Porter and Evans [1996] presented an efficient method to solve this problem. We used the same method to obtain the coefficient A_n. Once we get the amplitude of reflected and transmitted waves, the drift force on a strut is estimated from the momentum conservation and the energy conservation as

$$F_y = \frac{\rho g \varsigma_a^2 d\beta_0}{2k^2} \sum_{n=-r}^{s} \beta_n R_n \overline{R}_n, \tag{16}$$

where ς_a is the amplitude of incident wave, R_n is the reflection coefficient and over bar denotes the complex conjugate. Porter and Evans [1996] also showed the application of the wide spacing approximation to the two parallel arrays. We use the same method for the two parallel arrays of struts. The drift force of this case is also obtained from the momentum conservation as

$$F_y = \frac{\rho g \varsigma_a^2 d\beta_0}{2k^2} \sum_{n=-r}^{s} \beta_n R_n^{(1,2)} \overline{R}_n^{(1,2)}, \tag{17}$$

where $R_n^{(1,2)}$ denotes the reflection coefficient for the two parallel arrays of strut.

An anxiety of the twin-hull design is that the interaction among struts induces a blockage effect to the wave transmission of the incident wave. As a result the drift force becomes larger than that of insular strut. Thus we compare the drift force acting on the arrays of strut with that on an insular strut. The numerical procedure for solving the problem of insular strut is found in the handbook by Linton and McIver [2001] Fig.3 shows an example of drift forces versus chord length.

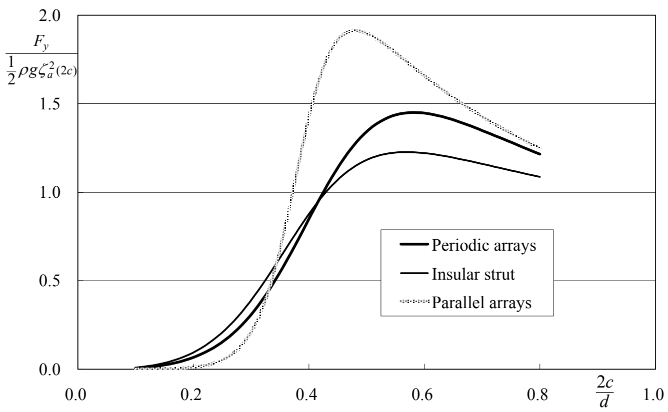

Fig.3 Comparison of the drift force vs. ratio between the gap and the chord length ($kd = 6,\ \chi = 90°$)

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Takagi, K.
'Hydroelastic motion and drift force of a twin-hull very large mobile offshore structure'

Discusser - R.C.T. Rainey

The transverse structure you show in figure 1 is not good. There are two hulls 64m apart, so there will be large 'pinching and prying' forces in the structure in beam seas 128m long. It will have severe fatigue problems.

Reply:

Thank you for your comment. We are planning to investigate a smaller space design to strengthen the transverse structure. Another idea is the mono-hull with outriggers. However, the mono-hull design may be difficult to provide enough space for the hydrogen conversion plant.