SESSION ON OCEAN ENGINEERING

Chairman: Dr. M. W. C. Oosterveld

Ocean Engineering Committee Memberships: G. van Oortmerssen (Chairman) – S. J. Rowe (Secretary) – R. A. Barr – E. Huse – A. Ivanov – K. Kokkinowrachos – J. Lundgren – J. U. Römeling – M. Takagi

Discussion of the Report and the Draft Recommendations of the Ocean Engineering Committee. (Cf Proceedings, Volume 1, p.469 – 526)

I. DISCUSSIONS

OE–1

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In the report (3.5.2 Current–wave Interaction) it is written that the drag coefficient of the cylinder reduces with increasing ratio of current to wave orbital velocity. I had investigated on wave forces acting on submerged circular cylinders moving forward with a constant velocity in regular waves and hydrodynamic forces acting on the cylinder forced to surge in a steady flow experimentally (Ref. [1]). The experimental results showed that the drag coefficients depend mainly on the reduced velocity Ur and the current velocity does not always reduce the drag coefficient. I want to introduce the results here. The drag force F acting on the surging cylinder in a steady flow or the cylinder moving forward with a constant velocity in regular waves is written by the Morison equation extended to the relative velocity concept as follows;

$$F_D(t)=-1/2\rho AC_D|U+\bar{u}\ \sin\ \omega t|\times(U+\bar{u}\ \sin\ \omega t) \qquad (1)$$

where ρ is the water density, A = DL, D is the diameter, L is the length of the cylinder, C_D is the drag coefficient, U is the forward velocity or steady flow velocity, $\bar{u}$ is the maximum value of surging velocity of cylinder or orbital velocity of fluid particle, ω is the circular frequency of surging motion or the incident wave.
The drag force $F_D(t)$ is expanded as a Fourier series and the first term of series is written as follows;

$$F_D(t)=-1/2\rho AC_D\,\bar{u}^2[(a^2+1/2)+2a\ \sin\ \omega t], \qquad (\text{for } a\geq 1)$$

$$F_D(t)=-\frac{1}{2\pi}\rho AC_D\,\bar{u}^2\{2(a^2+1/2)\sin^{-1}a+3a\sqrt{1-a^2}+4[1/3(a^2+2)\sqrt{1-a^2}+a\ \sin^{-1}\ a]\times\sin\ \omega t\},\ (\text{for } a<1) \qquad (2)$$

Where $a = U/\bar{u}$.
The measured horisontal in–line force F is expanded as

$$F=F_0+\sum_{n=1}^{3}(A_n\ \cos\ n\omega t+B_n\ \sin\ n\omega t) \qquad (3)$$

Upon comparing Equations (2)–(3), we have drag coefficients as follows;

$$\left.\begin{aligned} C_{D0} &= \frac{-Fo}{1/2\rho A\ \bar{u}^2(a^2+1/2)} \\ C_{D1} &= \frac{-B_1}{\rho A \bar{u}\ U} \end{aligned}\right\} \quad (\text{for } a \geq 1)$$

$$\left.\begin{aligned} C_{D0} &= \frac{-2\pi Fo}{\rho A\ \bar{u}^2[2(a^2+1/2)\sin^{-1}a+3a\sqrt{1-a^2}]} \\ C_{D1} &= \frac{-\pi B_1}{2\rho A\ \bar{u}^2[1/3(a^2+2)\sqrt{1-a^2}+a\ \sin^{-1}\ a]} \end{aligned}\right\}$$

$$(\text{for } a < 1) \qquad (4)$$

Where C_{D0} is the steady drag coefficient obtained from the steady force F_0 in eq. (3) and C_{D1} is the oscillating drag coefficient obtained from the oscillating force. The physical grounds of the two drag coefficients are on the time dependent drag coefficient as shown in Ref.[1].

I show the results in Fig. 1 and Fig. 2. These figures show that the drag coefficients depend on reduced velocity Ur and the value of C_D is not necessarily smaller than the drag coefficient of the cylinder in a steady flow or those of the cylinder oscillating in a still water.

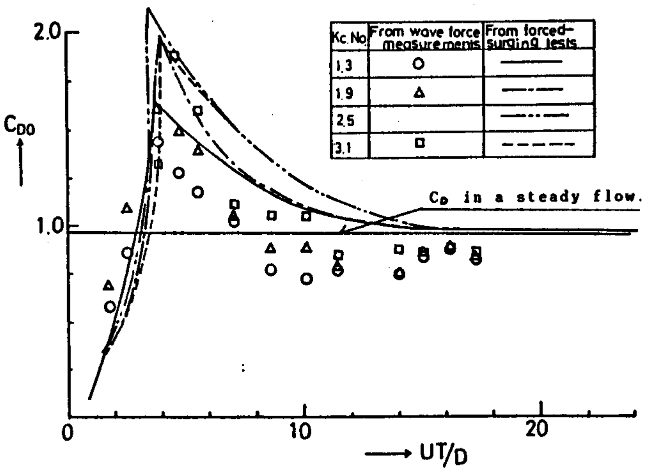

Fig. 1 Drag Coefficient C_{D0} of Cylinder Obtained from Steady Force F_0

In order to research on the low frequency damping force, we had carried out similar experiments on TLP section model (Ref. [2]) and got interesting results. Two kind of experiments were carried out to research on the slow drift damping, the first one is the two harmoic forced oscillation tests which is composed of the low frequency large amplitude surging motion and high frequency small amplitude surging motion. The second one is the low frequency large amplitude forced surging tests in regular waves. We intended to simulate the slow drift oscillation accompanied with the wave frequency motion in waves by these experiments.

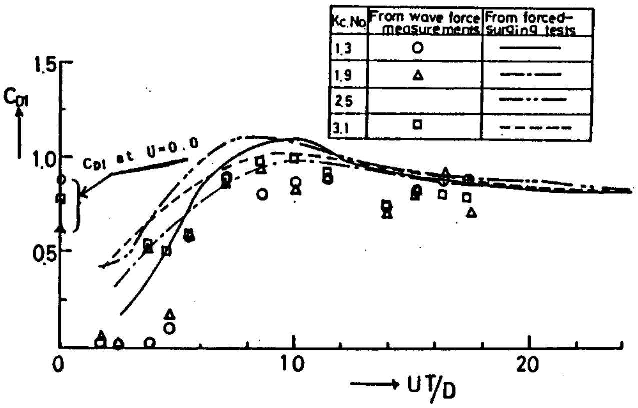

Fig. 2 Drag Coefficient C_{D1} of Cylinder Obtained from Oscillating Force

The drag force F_D in two harmonic motions or low frequency motion in waves written as follows;

$$F_D = -\frac{1}{2}\rho A_T C_D[\bar{U}_L\ \cos\ \omega_L t + \bar{u}\ \cos(\omega_S t+\varepsilon)]$$

$$\times|\bar{U}_L\ \cos\ \omega_L t + \bar{u}\ \cos(\omega_S t+\varepsilon)| \qquad (5)$$

where A_T is the projected area of TLP section model, $\bar{U}_L$ is the maximum velocity of cylinder due to the low frequency motion and $\bar{u}$ is the maximum velocity due to the wave frequency motion or the maximum orbital velocity of fluid particle, ω_L is the circular frequency of the low frequency motion, ω_S is the circular frequency of the wave frequency motion or waves and ε is the phase difference between the low frequency motion and wave frequency motion or incident wave.

Upon comparing eq. (5) with measured forces, we obtain the drag coefficients. From the ω_L−term of the Fourier−series obtained by expanding the measured force, we obtain the drag coefficient C_{DT} for the slow oscillation and from the ω_S−term, C_{D1} is obtained. In Fig. 3 and Fig. 4, the drag coefficient C_{DT} of TLP MODEL for the slow oscillation obtained from the two harmonic oscillation (Fig. 3) and low frequency oscillation in regular waves (Fig. 4) are shown. These results show that the drag coefficient for the low frequency oscillation depend on the reduced velocity U_{rc} defined by the maximum velocity $\bar{U}_L$ of the low frequency motion, period T_S of wave frequency motion or wave and diameter D of column. In the range of $U_{rc} < 2.0$, The value of drag coefficient C_{DT} for low frequency motion is smaller than that of the drag coefficient C_{DTO} obtained from the simple harmonic oscillation tests in a still water. In the range of $2.0 < U_{rc} < 4.0$, The value of C_{DT} is greater than that of C_{DTO}. In the range of $U_{rc} > 4.0$, the value of C_{DT} is approximately equal to that of C_{DTO}. These tendency are very similar to those obtained from forced oscillation tests in a steady flow or the wave force measurements on the cylinder with a constant forward velocity in regular wavs on which I already talk before and as shown by Moe and Verley (Ref. [3]).

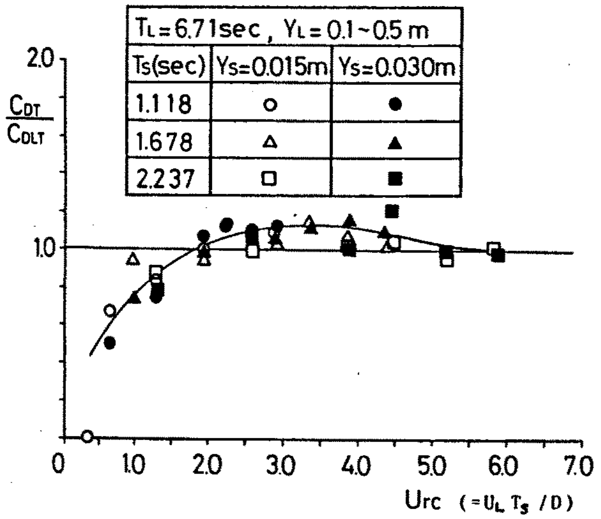

Fig. 3 Ratio of Drag Coefficient C_{DT} of TLP Section Model for Slow Drift Motion Obtained from Two Harmonic Surging Tests to Drag Coefficienct C_{DLT} Obtained from Simple Harmonic Surging Tests

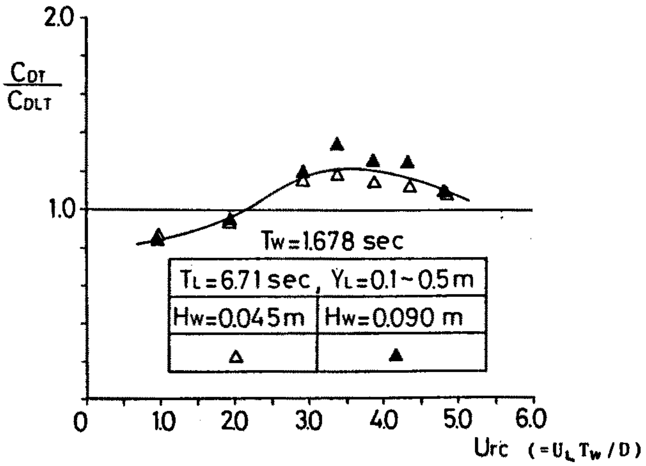

Fig. 4 Ratio of Drag Coefficient C_{DT} of TLP Section Model for Slow Drift Motion Obtained from Forced Surging Tests in Regular Waves to Drag Coefficient C_{DLT} Obtained from Simple Harmonic Surging Tests

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OE−2

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I am agreeable with the description of the report on the mooring line damping. The mooring line damping is very important at the resonant frequency of the moored ocean structure. I had proposed a practical calculation method of the dynamic tension of mooring line in 1978 (Ref. [1]). In the method, it is assumed that the catenary line keeps its profile during the motion and the amplitude of the motion of mooring line is obtained from the catenary theory. Observations on motions of the mooring chain excited at its upper end indicate that the assumption is valid in the water but it is not valid in air. The velocity and accelaration of the mooring chain are obtained from the amplitude and frequency of the motion. By using the appropriate drag– and added mass–coefficient, we can calculate the drag force and inertia force acting on the mooring chain. The dynamic tension can be calculated by taking account of the drag force, inertia force and linear restoring force calculated with the catenary theory. The mooring line damping can be obtained directly from the drag force acting on the line. A remarkable feature of this calculation method is its short computation time and its accuracy of the calculation result, and we can get the frequency domain solution of the dynamic tension of mooring lines.

Recently we applied the method on the estimation of the motion of a moored semi–submerged structure which is designed for the OTEC plant (Ref. [2]). I show results of forced surging tests and calculations of the damping coefficient of the platform with mooring chain in Fig. 1. The figure shows that the great part of the damping force is due to mooring lines in low frequency range.

We measured motions of the platform in various conditions. I show the results of surging motion without the cold water pipe and with mooring chain in irregular waves in Fig. 2. Experiments show that mooring chains reduce the slow drift oscillation, and calculation results taking account of the damping force acting on mooring chains show good agreement with the experimental result.

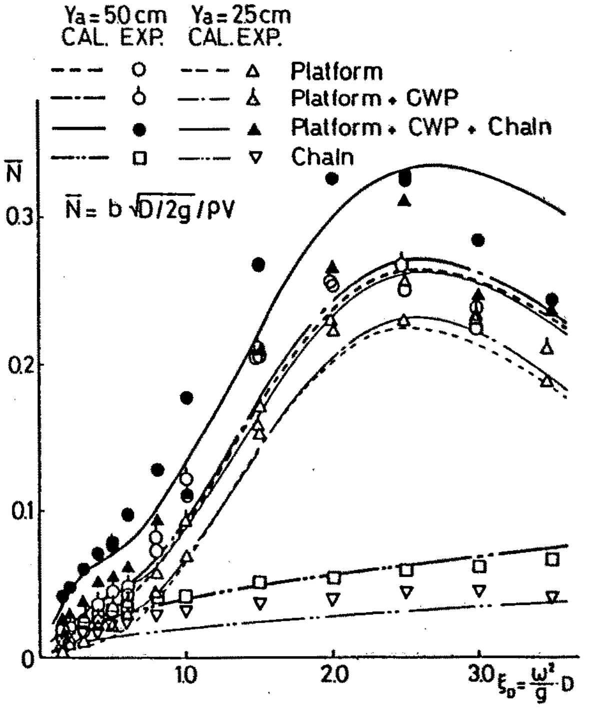

Fig. 1 Damping Coefficient of OTEC Platform with CWP and Mooring Chain

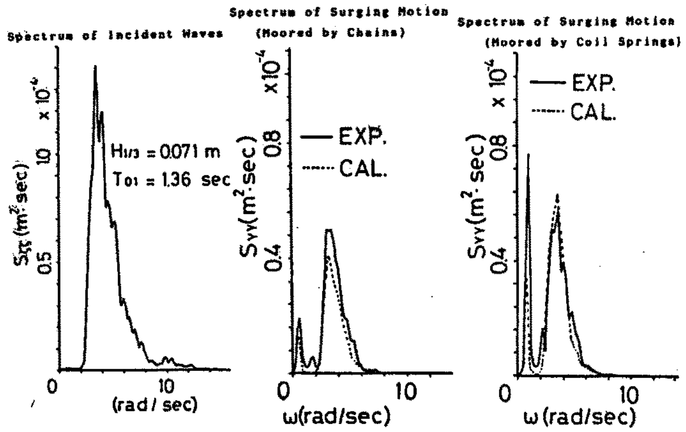

Fig. 2 Surging Motion of Platform without CWP Moored by Chain or Coil Spring

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OE−3

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INVESTIGATION OF LOW FREQUENCY DAMPING, CHARACTERISTICS OF MOTIONS AND DRIFT FORCES ON SEMI−SUBMERSIBLES

1. Introduction

The aim of the present contribution is to give some results of the recent BSHC studies supporting and/or discussing some aspects of the very nice Committee Report, such as low frequency damping, characteristics of motions, and mean wave drift forces. The comments are connected with related items of the Report as is shown below.

2. Low Frequency Damping (Items 5 and 8.2.5 of the Report)

It is well known that the low frequency damping cannot be determined by the potential theory. Dr. Ikeda's [1] investigations show that in flat plate oscillation in low KC numbers, there is a symmetric vortex shedding. On the other hand, these numbers correspond to 5−6% horizontal displacement restrictions for Semisubmersibles. The pontoons of these platforms are usually slender bodies with an almost rectangular cross sections and that means the points of vortex shedding are known (as in the flat plate). This allows modelling of the hydrodynamic interaction between the pontoon and the fluid, calculation of the low frequency damping coefficient using a discrete vortex method and its insertion, after linearization, in the equation of motions [2].

A comparison between the theoretical and the experimental (using PMM techniques) results is made for Semi−submersibles with a different pontoon height [3], Fig. 1.

Fig. 2 shows sway motion characterstics. The results obtained at BSHC give a slight difference from the rest of the investigations [4] in long waves. According to the author these differences are due to the included low frequency damping. Besides, there is a dependence not only on the frequency, but also on the amplitude of motion, therefore on the wave steepness, which is a premise for a connection with wave drifting damping. These results reflect on sway motions in regular waves in low frequency, as they are presented in Fig. 40(c) and Fig. 43 [5].

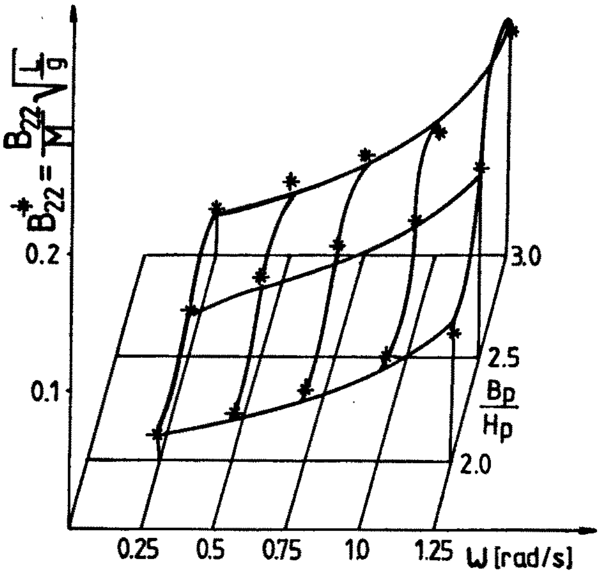

Fig. 1 Comparison of Theoretical Experimental Data for Low Frequency Damping

The differences in the maximum values of the theoretical and the experimental spectral densities for sway motions correspond to regular wave characteristics correlation for SR192 (Fig. 40c [5], Fig. 2).

We think that the low frequency damping investigation should continue by first comparing

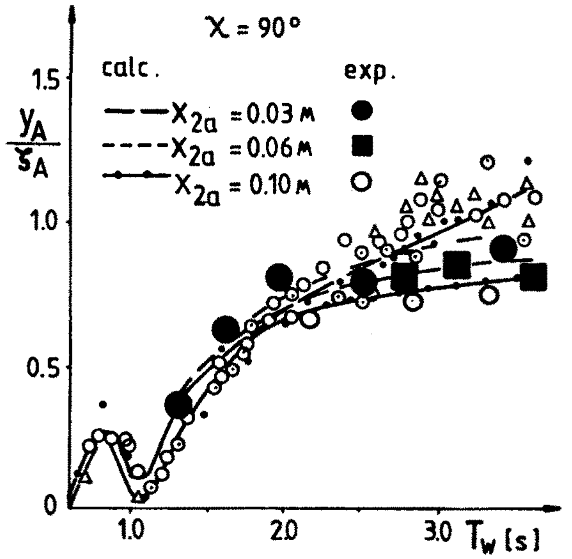

Fig. 2 Sway Motion Characteristics (○, ●, ■Exp. BSHC; — —, – – –, -●●- Calc. BSHC with Low Frequency Damping; Other — ITTC'84 Results)

regular wave results. Special attention must be paid to the modelling of hydrodynamic processes because the eleven causes (5.2 [5]) for the creation of damping have a different contribution. There is a need of different numerical methods having a corresponding limited scope of application. It is necessary to compare model test results for the sake of verifying nonlinear damping, obtained by different experimental techniques.

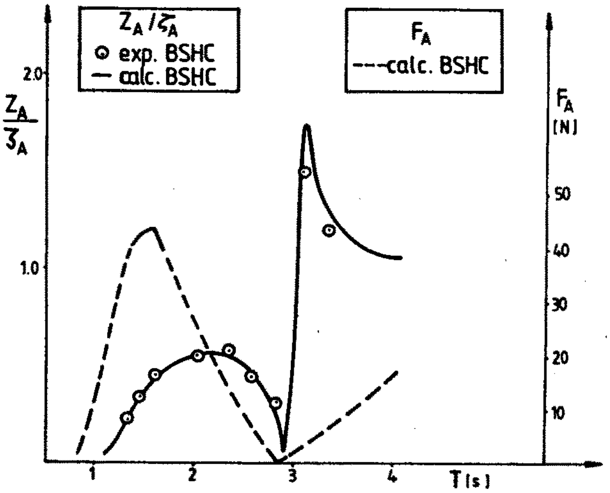

Fig. 3 Amplitude of Heave Motions and Heave Exciting Force at χ = 45°

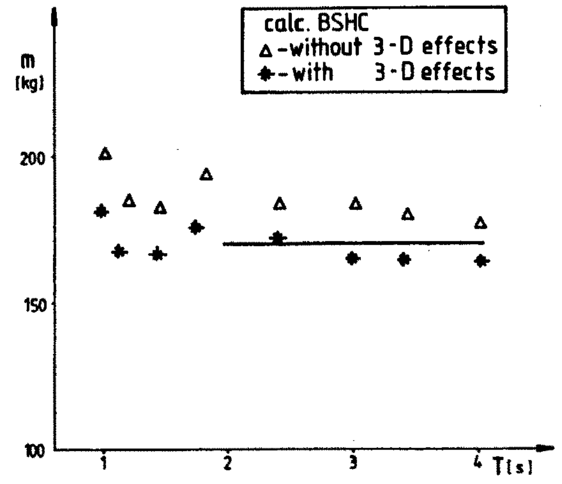

Fig. 4 Heave Added Mass

3. Results and Discussions (Item 8.1.2 of the Report)

The presentation of heave motion amplitude shows the Committee striving for priority for using programs including 3−D and viscous effects. BSHC investigations [2], [3] are in the same direction. Some of the results for χ = 0° and 90° for SR192, are already presented to some Committee members. The results presented here are analogous to those in Fig. 32, 34, 35 [5].

In Fig. 3 the amplitudes of heave motion and wave exciting force are presented. They are in good coincidence with the results given in the Committee Report. In Fig. 4 the added mass coefficient is presented. The calculations are made with and without taking account of the 3−D effects from the geometry and the hydrodynamic interference between platform and the fluid. In this connection we think that a possible way of predicting motion characteristics is analogous to those proposed in [4] (U1, Y2) and [3].

The influence of vortex shedding is very important but in heave motions from wave excitation the Keulegan−Carpenter number is very low (to pontoon width) and in this region it is not positive that the vortex shedding drag predominates over the friction drag. Moreover, some of BSHC investigations show that in low frequency sway motions the influence of motion amplitude or of the added mass coefficient is negligible.

The comparison of the results in Fig. 35 in Table 5 and Table 6 [5] shows only the number of elements, but does not give the element form and the source disposition and their influence on the pressure distribution.

4. Calculation Methods (Items 8.2.4 of the Report) and Conclusion

Fig. 5 gives a nondimensional wave drift force $F_{DRIFT} = F/\frac{1}{2}\rho g L \zeta_A^2$. A comparison shows that the experimental data are nearer to the upper range of F.F.M. and N.F.M. results (Fig. 38 [5]). The measurements are performed without measuring the kinematics for avoiding mechanical excitations [6].

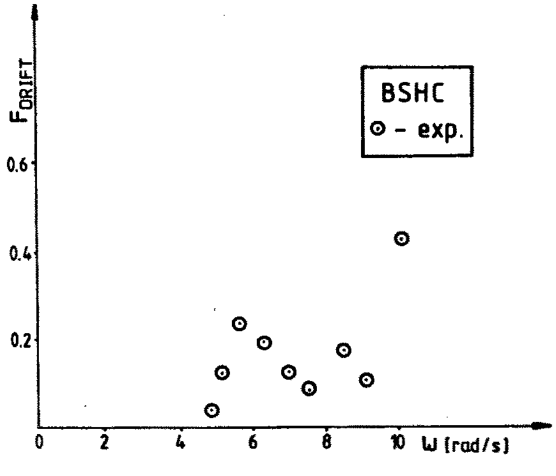

Fig. 5 Wave Drift Force Coefficients in Regular Waves

To conclude our presetation, we would like to thank the Ocean Engineering Committee for the very nice Report and to express our support of the recommended future works of the Committee.

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OE−4

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NECESSITY OF REVIEWING RESEARCHES ON THE STABILITY OF SEMI−SUBMERSIBLES

The researches on the stability of semi−submersible have been performed actively in the world, for example, the casualty research, researches for stability criteria as well as fundamental investigations, though the committee report says that there was little progress to the problem of prediction of extreme response of offshore structures. The results of these researches were presented at various symposia as follows;

a) The 5th Offshore Mechanics and Arctic Engineering Symposium, Tokyo, 1986,
b) Symposium on Stationing and Stability of

Semi−submersibles, Glasgow, 1986,

c) The 3rd International Conference on the Stability of Ships and Ocean Vehicles (STAB' 86), Gdansk, 1986.

In addition, the research projects performed in several countries have been reported at the IMO in order to contribute to reviewing the current MODU Code, which is the international standard for safety of the offshore structures.

Almost of these researches derived conclusions that the time domain simulation technique becomes available to predict extreme behaviours of semi−submersibles in heavy seas either in case of intact condition or under damaged conditions such as progressive flooding or breaking of a mooring line.

The model experiments have been carried out extensively and the results have proven the applicability of such simulation methods.

Therefore, such methods are expected to be effectively applied for evaluating the safety of semi−submersibles in future, instead of the conventional stability criteria.

The present discusser agrees with the recommendation 3.9 of the committee report in expecting that the next committee will summarize achievements gained from these comprehensive research activities in the world which would be very useful for understanding the extreme responses of semi−submersibles in severe sea conditions and for the sake of standardizing the simulation techniques as well as the environmental conditions to be used in the simulation or model testings.

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OE−5

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SLOW DRIFT MOTION AND DAMPING

As stressed in Chapter 5 and Section 7.3 of the 18th ITTC Ocean Engineering Committee Report, the low frequency damping plays a significant role on the slow drift resonant motion of a moored floating body in irregular waves. The purpose of this note is to support the conclusion drawn in the report and also to discuss the relative importance of wave radiation damping, wave drift damping and viscous damping due to current. Although the present discussion is largely based on the study of a two−dimensional floating cylinder in slightly modulated beam seas, it is our opinion that the following statements may equally be valid for a quite broad class of problems of offshore platforms.

If, among second−order effects, the low−frequency excitation is due to difference−frequency waves, the excitation frequency can be as low as the natural sway frequency of the moored vessel. Newman (1974) has found that the slow drift force can be written as a quadratic transfer function of the wave components. He suggests that for small frequency differences the coefficients of the function can be approximated by their values at zero difference. As a result the slowly−varying drift force is almost as simple as the steady drift force in a seaway with a narrow−banded spectrum. It is the reason why a seaway can be represented by wave trains with amplitude modulation in the study of slow drift motion.

If both the wave steepness (kA) and the modulation ratio (Ω/ω) are assumed to be small

$$kA=\varepsilon,\ \varepsilon=o(1),\ \Omega/\omega=O(\varepsilon)$$

it is found (Agnon et al., 1987) that the magnitude of slow sway drift (X_0) of a 2−D cylinder can be large when the mooring stiffness (K) is very weak. Because resonance may occur within the framework of potential theory

$$X_0=O(1) \text{ when } K/M=O(\Omega^2),\ M=\text{body mass.}$$

It is to note that the velocity of the slow motion is small due to low frequency ; $X_{0t_1}=O(\varepsilon)$, $t_1=\varepsilon t$.

As a consequence it can be easily shown that not only the wave radiation damping but also the wave drift damping have magnitudes of $O(\varepsilon^3)$, which must be discarded in the consistent second−order equation of slow motion. It implies unbounded sway magnitude at resonance. It is very clear that real fluid effects sofar ignored are most important, in particular, near resonance. The viscous drag can have magnitudes of $O(\varepsilon^2)$ for induced current or even of O(1) for environmental current, if the current speed is not too small and the body dimension is assumed to be O(1). Therefore it is extremely important to study the viscous damping either experimentally or theoretically in order to understand the flow associated with slow drift.

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OE−6

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ON THE SECOND-ORDER RADIATION CONDITION

The second-order radiation condition has been a controversy. Hunt and Baddour claimed that the second-order radiation condition required zero outgoing energy flux unlike the first-order radiation condition of the Sommerfeld type, attacking a second-order diffraction problem for an infinitely-long vertical cylinder [1]. Garrison imposed a second-order radiation condition of the Sommerfeld type in order to solve a second-order diffraction problem for three-dimensional arbitrary-shaped bodies by means of the second-order Green function [2]. The author obtained second-order diffraction potentials as well as second-order forces for an infinitely-long vertical cylinder, discussing the second-order radiation condition [3]. Confining the discussions to the diffraction problem in the finite water depth without modifying the point of the discussions, it seems obvious that waves up to the second order in the far field consist of four wave components:

$$\zeta=\zeta_I^{(1)}+\zeta_D^{(1)}+\zeta_F^{(2)}+\zeta_L^{(2)}$$

where

$\zeta_I^{(1)}$ = first-order incident wave

$\zeta_D^{(1)}$ = first-order diffracted wave

$\zeta_F^{(2)}$ = second-order free wave

$\zeta_L^{(2)}$ = second-order locked wave

$\zeta_D^{(2)}$ behaves like $\zeta_D^{(1)}$. Both $\zeta_D^{(1)}$ and $\zeta_F^{(2)}$ are outgong waves, satisfying the dispersion relation:

$$\omega^2/g=K$$

where ω and K are circular frequency and wavenumber of the relevant waves respectively and g is acceleration of gravity. On the other hand $\zeta_L^{(2)}$ is locked to $\zeta_I^{(1)}$ because $\zeta_L^{(2)}$ results from the product of $\zeta_I^{(1)}$ and $\zeta_D^{(1)}$.
Then it is clear that the second-order radiation condition of the Sommerfeld type should be imposed only to the free-wave potential, not to the locked-wave potetial or the both combined. The author used the integral-equation method with simple sources, where the free surface was discretized as well as the body surface. Attention should be paid to how deeply submerged body-surface elements are dealt with, as shown in Fig. 1. He showed in Fig. 2 that even the body-surface elements in the depth should be taken care of because the second-order potential on the weather side decays much more slowly in the depth directon than the first-order potention does.

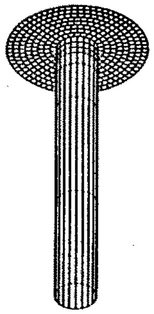

Fig. 1 Element Division

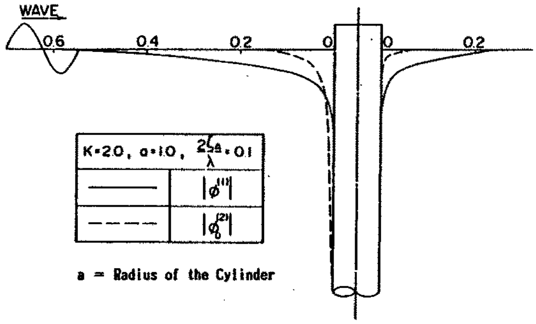

Fig. 2 Variations in the Depth Direction of Oscillatory First-Order and Second-Order Potentials on Weather and Lee Sides

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OE-7

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VARIABILITY OF COMPUTED HEAVE ADDED MASS FOR A SUBMERGED PONTOON

The Ocean Engineering Committee's Report pays considerable attention to the investigation of the heave added mass prediction method in order to explain the discrepancies in the predicted heave natural period which was observed in the last ITTC report [1]. The authors, in this paper, briefly mention the recent study regarding the influence of the facet discretization on the computed heave added mass.

For this study, a 1.0 m long submerged pontoon model is assumed and it has the same sectional shape as that used in the committee (Fig. 1). Two- and three- dimensional source distribution methods [2-5] are usd for the calculation of heave added mass. In the two-dimensional calculation, as is shown in Fig. 2, the geometrical shape of the section is represented by a given number of offset points with straight line segments (6 - 84 per section) between the points. In the three-dimensional claculation, the body shape is represented by a set of quadrilateral elements. An example of a discretization of the body surface per quadrant is shown in Fig. 3. In the figure, P3S-16 stands

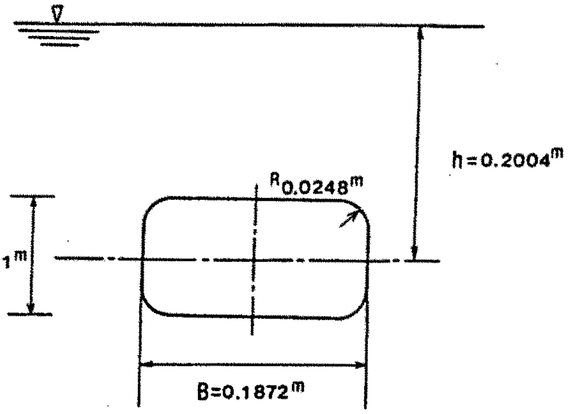

Fig. 1 Dimensions of a Submerged Pontoon

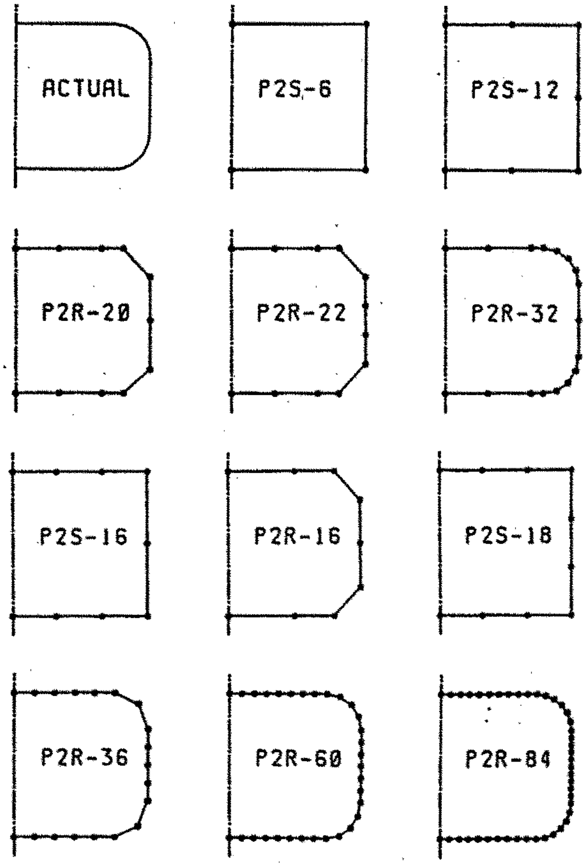

Fig. 2 Panels Approximation of a Cross Sectional Shape for Two-Dimensional Calculations

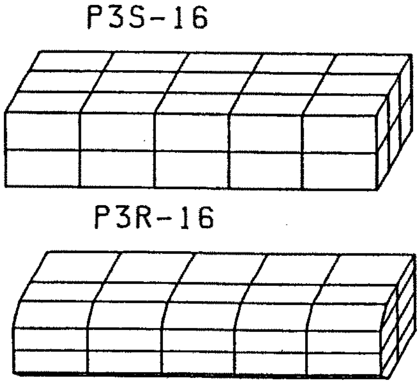

Fig. 3 An Example of Facets Modelling of a Pontoon for Three-Dimensional Calculations

for the discretization having 16 elements around the circumference of the pontoon and it has pointed (sharp) corners. On the contrary, P3R-16 is for the one with rounded corners.

Figs. 4 and 5 show the behavior of the nondimensionalized heave added mass coefficient:
C_a = added mass/(0.25 π ρ B^2L)
where L, B: length and breadth of the pontoon for varying the number of elements around the circumference of the pontoon at the four motion periods. Fig. 4 is for the two-dimensional calculation result and Fig. 5 is for the three-dimensional one.
It can be seen from the figures that:

1) When the number of panels approximating the model shape around the circumference is

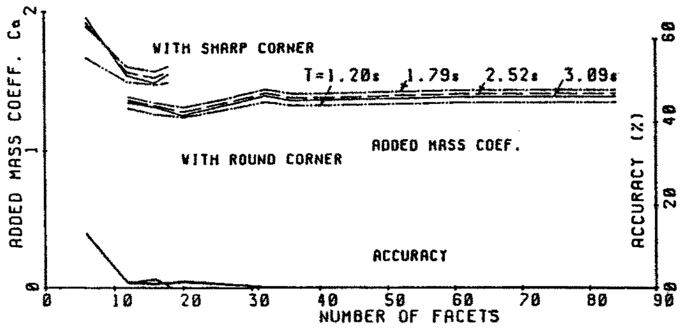

Fig. 4 Variation of Heave Added Mass Coefficients With the number of Panels (Two-Dimensional Source Distribution Method)

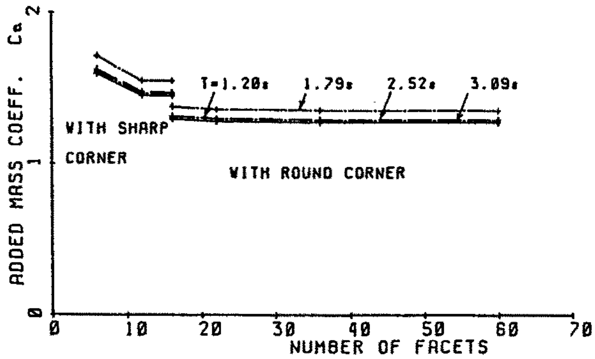

Fig. 5 Variation of Heave Added Mass Coefficients With the Number of Panels (Three-Dimensional Source Distribution Method)

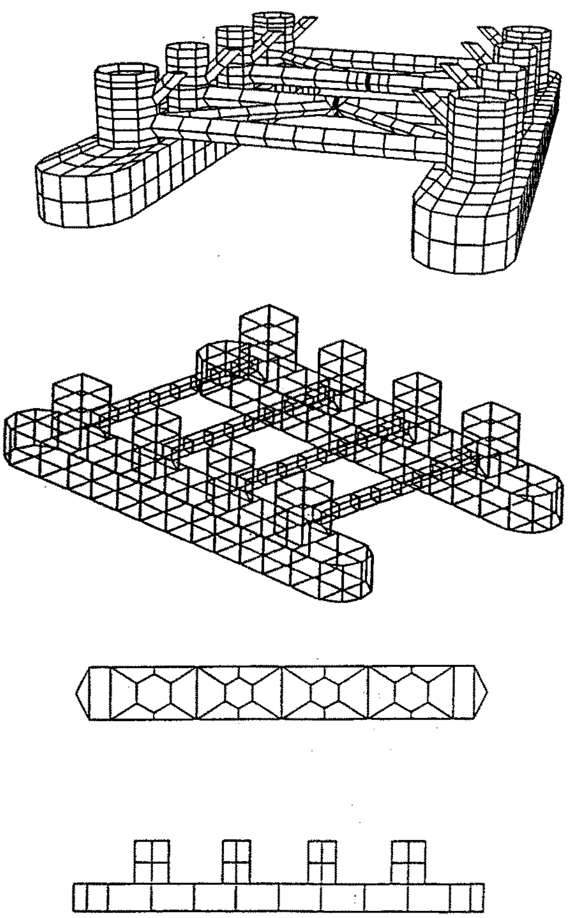

Fig. 6 An Example of Facets Discretization Used by some Participants

small and the corners are pointed, it may cause about 10% difference in the calculated heave added mass when it is compared with the result obtained by using a large number of panels and rounded corners.

2) However, when the corners are approximated by the rounded ones, the result does not depend much on the number of panels around the circumference.

We consider now the ITTC comparative study. Typical panels used by some of the participants are shown in Fig. 6 [6] and it is found that most organizations used small number of panels around the circumference. Lundgren [7] and Standing [8] discussed some reduction of heave added mass with increased panel refinement around the pontoon. Taylor et al. [9] also mentioned the same effect on the hydrodynamic forces for a tension leg platform. These discussions coincide with the result of this paper. However, in this paper it is stressed that the discretization at the corner parts of the pontoon is important and it may be said from the study that: Much finer panels, at least, enough to represent the rounded corners of the lowerhull than those used by most of the participating organizations might decrease the observed discrepancy.

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[6] Takagi, M.: A Follow-up Survey with 17th ITTC Comparative Material for 3rd O.E. Meeting in Houston, 1986.

[7] Letter from Dr. J. Lundgren, SSPA Maritime Consulting AB, to Dr. M. Takagi, Hitachi Shipbuilding Corporation, dated May 1986.

[8] Letter from Dr. R. G. Standing, British Maritime Technology, to Dr. M. Takagi, Hitachi Shipbuilding Corporation, dated October 1986.

[9] Taylor, R.E. and Jefferys, E.R.: "Variability of Hydrodynamic Load Predictions for a Tension Leg Platform", Ocean Engng, Vol. 13, No. 5, 1986, p.449-490.

OE-8

J.A. PINKSTER and H.J.J. VAN DEN BOOM, Maritime Research Institute Netherlands (MARIN), The Netherlands

1. Test duration

The purpose of carrying out model tests of floating offshore structures is, among others, to obtain design loads for the mooring system.
Mooring loads generally contain significant, large amplitude, low frequency components mainly caused by mean and slowly varying wave drift forces combined with the low damping characteristics of the moored system.

From the point of view of statistical reliability of the results of model tests in irregular waves it can be assumed that in some cases the test duration is insufficient i.e. the low frequency components of mooring forces and horizontal motions contain relatively few oscillations.

Advances in measuring equipment, data collection and analysis systems and techniques allow test durations to be increased while still containing costs at an acceptable level. This is leading to a steady increase in the duration of model tests.

Whilst it is recognized that an increase in model test duration will be beneficial from the point of view of statistical reliability of the results, no general guidelines exist on which the selection of the test duration can be based.

It is recommended that attention be devoted to this issue and that, based on knowledge of the behaviour of moored systems on the one hand and on recent developments with respect to the determination of design loads from model test results on the other hand, the ITTC Ocean Engineerng Committee develop some guidelines to this effect.

2. Modelling and Simulation of Wind and Current

The importance of wind loads on offshore platforms is widely recognized. Wind may contribute significantly to the excitation of resonant horizontal motions of moored and compliant structures. As outlined in Section 3.6.2 numerical simulation of wind gust forces may be basd on the quasi—static forces.

As investigated by the SNAME MS—3 panel existing calculation procedures as recommended by certifying authorities provide great discrepancies when compared with windtunnel tests.

Although theoretical flow calculations may not be feasible for complex structures at present, it is believed that mathematical description of wind forces should be encouraged by the ITTC.

3. Low frequency damping

The non—potential damping effects dominate the resonant low—frequency motions of moored structures. An impressive list of damping components is presented in Section 5.2 and discussed separately in Section 5.3. In order to obtain recommendations for model test and calculation procedures quantification of these components for selected configurations can be of great help. Analogous to trial measurements for ship resistance and propulsion, basic information may be derived from full scale measurements e.g. extinction tests.

4. Analysis of "Flexibles" in offshore Systems

The non—linear dynamic behaviour of mooring lines is mainly due to the geometry of the line and to the drag in normal direction. Contrary to riser forces, extreme mooring/tow line tensions are hardly affected by the choice of fluid force coefficients. This conclusion is also important for model experiments.

Finite difference methods based on lumped mass discretization have proven to provide remarkable efficient but also very clear procedures to model multi—component flexibles featuring clumps, buoys, tethers etc.

OE—9

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STOCHASTIC BENDING STRESSES AND ITS DISTRIBUTION OF MARINE RISER

Marine Risers are employed between a vessel and a subsea well. For some of productional risers the required design life and working time are much greater than conventional drilling risers. Fatigue life may thus become an important Criterion in engineering design.

A mathematical model for analysing the stochastic process of bending stresses, during the top end motion with large deflection, nonlinear and time domain situation, basd on the so called lumped mass method has been developed. It's an extended research of ref [1].

The results of this paper indicate that the largest bending stresses along the riser moving stochasticly would not be higher than the bending stresses under the situation of harmonic motion with large amplitudes and higher frequencies, shown in Fig. 1. It implies that the fracture occurance mostly relates to the factors of fatigue.

Statistical distribution of bending stresses along the riser is described in this paper, the cumulative probability distribution of extreme bending stresses has been drawn nearly to the theoretical distribution of Weibull, shown in Fig. 2.

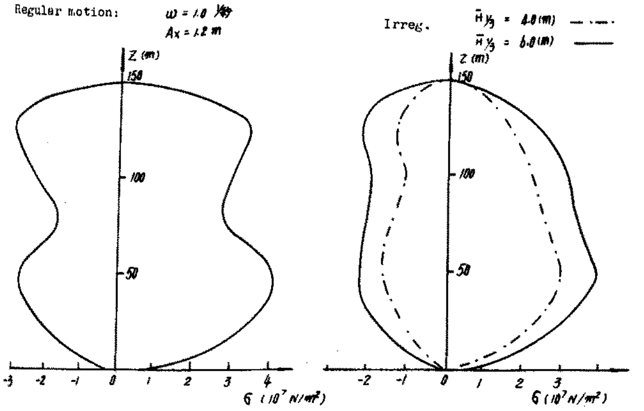

Fig. 1

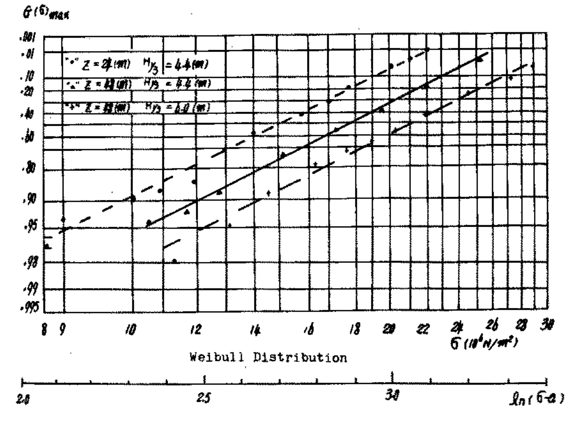

Fig. 2

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OE—10

W. B. MORGAN, David Taylor Research Center, U.S.A.

Recommendations 3.1, 3.2 and 3.3 on page 525 and recommendation 3.7 on page 526 are not really recommendations for future work of the Committee. They should either be dropped or reworded.

OE—11

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MODEL/FULL SCALE CORRELATIONS WITH REGARD TO SECOND ORDER WAVES

1. This discussion is in relation to the draft recommendation to the conference 2.3, which states "Members performing tests on moored systems are recommended to take special note of the lack of model/full scale correlation of the low frequency motions of the systems". Although some wave tanks are equipped with wave generation systems with 2nd order long wave compensation, most tanks do not. Besides all beaches designed for oridinary wave absorbtion do not function effectively for group—induced long waves and parasitic long waves. Consequently, there are doubts over reflection build—up for these long waves in the tank as mentioned in sec-

tion 3.3 of the 17th ITTC O.E. report. The following is a contribution to quantify such effects.

2. The three−probe correlation method

Goda and Suzaki [2] used a two−probe method to extract information regarding reflection coefficient of primary waves from a beach. We extended Goda's concept to a three probe linear array and attempt to separate the three systems of long waves.

Let three wave probes be arranged in a linear array as shown in Fig. 1, the location of the array is defined by the first probe which is placed nearest to the wave generator.

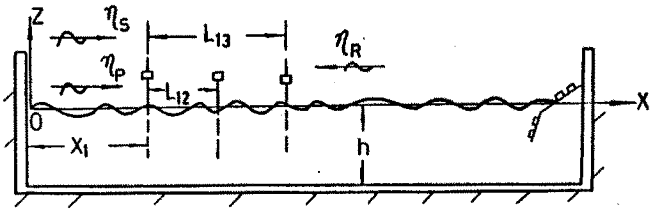

Fig. 1 The Three−Probe Correlation Method

We express the three systems of long waves as

$$\eta_s(t)=\sum_{n=1}^{N} A_{s,n}\cos[(k^-)_n x-(\omega^-)_n t+\varepsilon_{s,n}] \qquad (1)$$

$$\eta_p(t)=\sum_{n=1}^{N} A_{p,n}\cos[\hat{k}_n x-(\omega^-)_n t+\varepsilon_{p,n}] \qquad (2)$$

$$\eta_r(t)=\sum_{n=1}^{N} A_{r,n}\cos[\hat{k}_n x+(\omega^-)_n t+\varepsilon_{r,n}] \qquad (3)$$

where the positive direction of x is taken as from the generator to the beach. $(\omega^-)_n$ is the nth component of angular frequency of set−down and parasitic waves in their respective spectra. $(\omega^-)_n$ is defined as the difference $(\omega^-)_N=\omega_N-\omega_{N'}$ $(N > N')$, similarly, $(k^-)_n = k_N - k_{N'}$ and $\hat{k}_n$ satisfies the dispersion relationship $(\omega^-)^2_n = \hat{k}_n g\,th(\hat{k}_n d)$, d is the water depth, $A_{s,n}$, $\varepsilon_{s,n}$, $A_{p,n}$, $\varepsilon_{p,n}$, $A_{r,n}$, $\varepsilon_{r,n}$ are respectively the wave amplitude and phases of the nth component of set−down, parasitic and reflected long waves.

Spectrum in real life corresponds to a significant wave height $H_{1/3}$ of 5.74m. The location of the first wave−probe of the array is at 15m, 35m, 45m, 55m respectively from the wave generator end. Since the 3−probe method depends very much on the phase relationship between measurements at each probe, it is of utmost importance to start and terminate the recording of 3 probes simultaneously by an external trigger circuit. Ultrasonic wave probes, and RTP−520 tape recorders are used. The first 5 min. of recording was not used, the total recording time was 30 min.

Fig. 2 is a complete spectra of the low frequency (LFS) and primary frequency waves in the tank, (X_l = 35), the LFS break down is obtained by 3 probe analysis. By measuring the area under the respective LFS, the significant wave heights of these long waves are obtained as shown in Table 1.

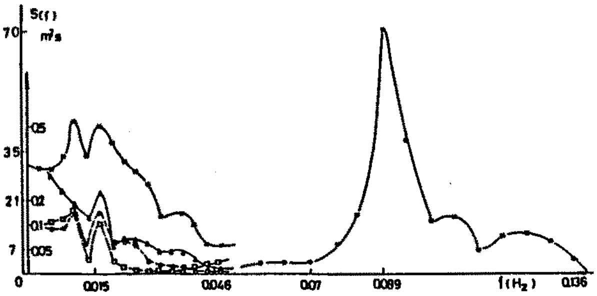

Fig. 2 The Low−Freq. Spectra and Primary Wave Spectrum in Full−Scale (−−Δ−−Set−Down, −−o−−Refl. Wave, −−□−−Parasitic, −−*−−Low−Freq. and Primary Wave, X_l = 35m)

Table 1

X_1(m)	energy wave height direct measure H_e	 derived H_c	set-down H_s(m)	reflected H_r(m)	parasitic H_p(m)
15	0.29	0.32	0.22	0.15	0.18
35	0.45	0.37	0.26	0.19	0.17
45	0.42	0.39	0.28	0.20	0.19
55	0.41	0.39	0.28	0.22	0.18

3. Analysis by square—law operation

As described in section 1 and 2, the wave system in a tank may be considered to have the following block diagram of transfer:

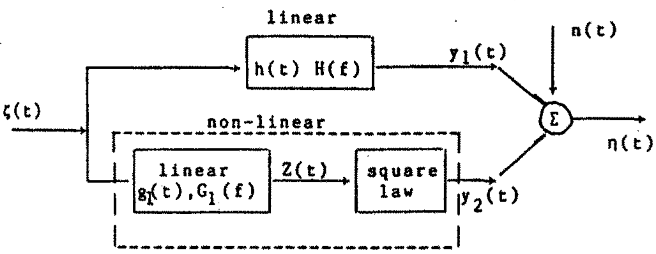

Fig. 3

Table 2 shows comparisons of set—downs $H_{1/3}$ obtained by square—law operation (calculated) and by 3—probe correlation method (experiment). They compare reasonably well.

Table 2

wave probe at	$H_{1/3}$ by 3probe meas.	$H_{1/3}$ by square la
35 m	0.26 m	0.31 m
45	0.28	0.26
55	0.28	0.30

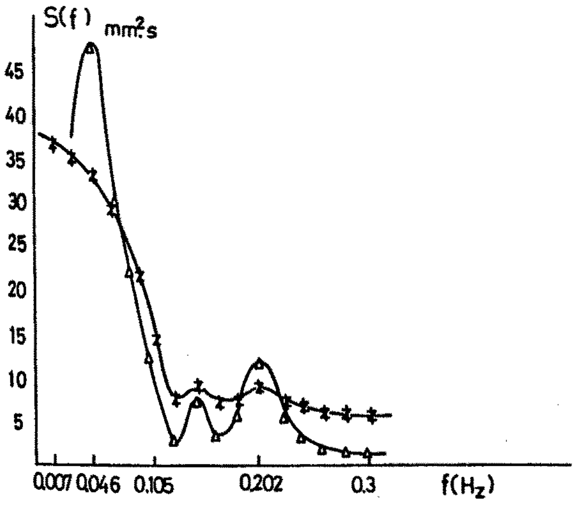

Fig. 4 The Comparison of Set—Down Spectra Obtained by Square—Law (——≠——) and by 3—Probe (——Δ——) Operations, x = 55m

4. Concluding remarks

We may never be able to have a wave tank for ideally generating random waves with wave grouping and set—down simulated as in real life. The parasitic waves and reflected waves accompanying set—down, though theoretically removable, still prove to be very expensive in practice. Again, what we obtain as a result of tank test of moored structures concerning low—frequency excursions may be very difficult to validate against full—scale experiment. The present paper uses low frequency waves as the target, and investigated the feasibility of 1) to separate the three systems of long waves by 3—probe correlation method, the spectra obtained appeared reasonable, 2) by square—law operation in which set—down spectrum is extracted directly from 1st—order wave spectrum without going into complicated quadratic transfer functions. The parasitic and reflection waves are treated as noise. This system identification technique seems to give results that cross—checked the 3—probe analysis. We admit that the noise is rather large. But the cross—checking of the methods is encouraging. The inherent indication of this investigation is that square—law operation may be also useful in correctly analyzing the low—frequency forces acting on, and motions of, a moored structure in a wave tank.

OE—12

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The importance of dynamic wind and second—order wave loading acting on floating and compliant offshore strctures is referred in sections 3 and 5 of the report respectively. In order to quantify the effects of various parameters in calculating the second—order loading due to wind and waves two examples will be illustrated in this contribution using an articulated tower and a coupled articulated tower and ship system. In the study reported here the effects of variations in the definition of spectral shapes and in inertia and damping coefficients were considered.

Table 1 Geometrical Characteristics and Weight Distribution for an Articulated Tower Platform

Structural Member	Length [m]	Diameter [m]	Weight [Tonnes]
Deck	6	30	500
Upper Column	45	7	772
Buoyancy Tank	20	15	735
Lower Column	141.65	8	2,777
Ballast	18.35	18.35	12,963

Table 1 shows the geometical details and weight distribution of an articulated tower structure used in the study. The sensitivity of the pitch response values to changes in different wind spectral forms (i.e. Harris and Davenport Spectra) as well as in inertia and damping coefficients, the mean wind velocity, the wind velocity distribution factor β, the reference length L, and gustiness (or surface drag coefficient) K was indicated in Table 2. The following points may be concluded from Table 2.

- ∗ Harris Wind Spectrum generally yields higher response values.
- ∗ The motion response values increase significantly as the mean velocity, L and K increase.
- ∗ The motion response values increase as inertia and drag coefficients decrease.
- ∗ The motion responses are not sensitive to β
- ∗ The motion responses due to dynamic wind loading are in the same order as the responses due to first−order wave loading.

Table 2 R.M.S. Pitch Response(∗) Spectral Values due to Dynamic Wind Forces for the Platform Given in Table 2

SPECTRUM	TYPE			HARRIS				DAVENPORT			
C_M	$\bar{V}_{w,10}$	β	L	1200		2000		1200		2000	
			d \ K	.0015	.0033	.0015	.0033	.0015	.0033	.0015	.0033
1.5	35.	.075	.05	.175	.259	.189	.280	.170	.252	.190	.282
			.10	.124	.185	.136	.202	.119	.176	.135	.200
			.20	.089	.132	.100	.148	.082	.122	.096	.143
		.125	.05	.180	.268	.195	.290	.176	.261	.196	.291
			.10	.129	.191	.141	.209	.123	.182	.140	.207
			.20	.092	.137	.103	.153	.085	.126	.099	.147
	48.	.075	.05	.299	.444	.342	.507	.276	.410	.340	.504
			.10	.212	.315	.245	.363	.192	.285	.239	.254
			.20	.151	.224	.176	.262	.132	.195	.167	.247
		.125	.05	.309	.459	.354	.525	.286	.424	.351	.521
			.10	.220	.326	.253	.376	.199	.295	.247	.366
			.20	.156	.232	.182	.271	.136	.202	.172	.255
2.0	35.	.075	.05	.165	.245	.185	.275	.156	.232	.185	.274
			.10	.117	.174	.133	.197	.109	.161	.131	.194
			.20	.084	.124	.096	.143	.075	.111	.092	.136
		.125	.05	.171	.253	.191	.284	.161	.239	.191	.284
			.10	.121	.180	.137	.204	.113	.167	.135	.200
			.20	.087	.128	.100	.148	.077	.115	.095	.141
	48.	.075	.05	.278	.413	.328	.486	.245	.363	.319	.473
			.10	.197	.293	.234	.347	.170	.252	.223	.331
			.20	.140	.208	.167	.248	.116	.172	.154	.229
		.125	.05	.288	.427	.339	.503	.253	.375	.330	.490
			.10	.204	.303	.242	.359	.176	.261	.231	.343
			.20	.145	.215	.173	.257	.120	.178	.160	.237

(∗) : Given in degrees

$\bar{V}_{w,10}$: Hourly mean wind speed at 10 m (m/s)

β : Wind velocity distribution factor

L : Reference Length (m)

K : Gustiness or surface drag coeffecient

The prediction of motion responses of the articulated tower geometry given in Table 1 was also performed under slowly varying second−order wave loading. Table 3 shows r.m.s. angular response predictions carried out using Bretschneider, Jonswap and ITTC Spectra as the inertia and damping coefficients as well as the significant height and zero crossing wave periods were varied. The results show that motion amplitudes are sensitive to changes in damping coefficients.

It should also be pointed and that the magnitude of motion response under second−order wave and dynamic wind loading are very similar.

Finally Table 4 illustrates surge motions of the

Table 3 R.M.S. Pitch Response(*) Spectral Values due to Slowly Varying Second−Order Wave Excitation for the Platform Given in Table

SPECTRA	C_M	1.5		2.0	
	H_s/T_z \ d	10%	20%	10%	20%
BRETSCHNEIDER	8.0m/ 8.0s	0.1852	0.1303	0.1640	0.1156
	10.0m/10.0s	0.2109	0.1488	0.1883	0.1330
	12.0m/12.0s	0.2263	0.1601	0.2037	0.1442
	14.0m/14.0s	0.2366	0.1680	0.2146	0.1523
	16.0m/16.0s	0.2439	0.1737	0.2229	0.1586
JONSWAP	8.0m/ 8.0s	0.1726	0.1241	0.1586	0.1139
	10.0m/10.0s	0.1851	0.1336	0.1704	0.1230
	12.0m/12.0s	0.1933	0.1400	0.1780	0.1290
	14.0m/14.0s	0.1996	0.1447	0.1838	0.1334
	16.0m/16.0s	0.2044	0.1484	0.1881	0.1374
ITTC	8.0m/ 8.0s	0.1832	0.1311	0.1660	0.1190
	10.0m/10.0s	0.2039	0.1461	0.1844	0.1324
	12.0m/12.0s	0.2176	0.1559	0.1969	0.1416
	14.0m/14.0s	0.2274	0.1631	0.2065	0.1487
	16.0m/16.0s	0.2346	0.1685	0.2142	0.1542

(*) Given in degrees

H_s: Significant Wave Height (m)

T_z: Average zero crossing period (seconds)

coupled articulated tower and ship system (see Fig. 1) under first and second−order wave loading using P−M spectrum and under dynamic wind response using Davenport, Harris and Kaimal Spectra. Table 4 indicates that this particular compliant system is very sensitive to the definition of wind spectra and to damping coefficients in predicting the motion responses due to wind loading.

In conclusion author would like to suggest that parametric studies to identify and quantify the uncertainties associated with second−order wave and dynamic wind loading should be encouraged by the ITTC.

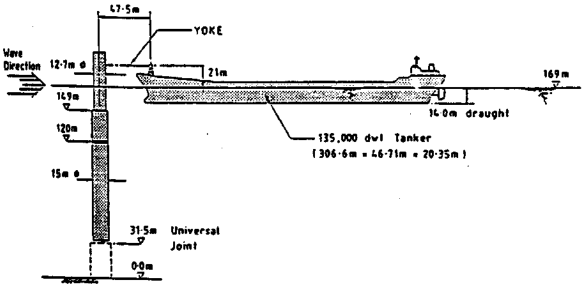

Fig. 1 Tanker/Column Tests (Head Seas)

Table 4 R.M.S. Surge Displacement of the Ship (in Metres)

		ζ=0.05	ζ=0.10	ζ=0.15
First Order Wave (P-M Spectrum)	U=20m/s	0.30	0.30	0.30
	U=30m/s	2.06	2.06	2.06
	U=40m/s	5.40	5.40	5.40
Second Order Wave (P-M Spectrum)	U=20m/s	0.68	0.67	0.66
	U=30m/s	2.24	2.23	2.22
	U=40m/s	4.38	4.37	4.34
Davenport Wind Spectrum	U=20m/s	0.39	0.29	0.23
	U=30m/s	0.61	0.46	0.35
	U=40m/s	0.82	0.61	0.47
Harris Wind Spectrum	U=20m/s	0.63	0.47	0.36
	U=30m/s	1.16	0.87	0.67
	U=40m/s	1.78	1.34	1.03
Kaimal Wind Spectrum	U=20m/s	0.50	0.38	0.29
	U=30m/s	0.95	0.71	0.55
	U=40m/s	1.48	1.11	0.86

U : wind velocity at 10m above the water surface
ζ : damping factor

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II. REPLY BY THE OCEAN ENGINEERING COMMITTEE

The Committee thanks all the discussers for their valuable contributions.

With regard to the comments on the draft recommendations for the future work of the Committee which were made by **Dr. Morgan (OE−10)**, the Committee should like to state that the recommendations 3.1, 3.2, 3.3 and 3.7 were clearly intended as recommendations for future work of the Committee. The Committee agrees with Dr. Morgan that the wording of recommendations 3.1, 3.3 and 3.7 is not quite appropriate and proposes to reword these as follows:

3.1 The Committee should monitor work that will lead to the identification of cases where it is necessary to model multidirectional waves.
3.3 Attention should be given to the definition of typical joint probabilities of extreme wave, wind and current conditions.
3.7 The Committee should continue to monitor research work on flow around cylindrical members. Particular attention should be given to new theoretical developments based on vortex modelling and the correlation of these methods with model tests and full scale.

The Committee feels, however, that the wording of draft recommendation 3.2 is appropriate. There is a clear need for more data on multi−directional wave fields. In several cases data are collected on behalf of the industry, but such data are usually not disclosed. The future Ocean Engineering Committee should actively encourage the industry to make such data available to the engineering community for the general benefit of our profession.

With regard to the discussion by **Dr. Takaishi (OE−4)** the Committee should like to make the following comments. The Committee fully agrees with Dr. Takaishi that in the last few years a lot of research work has been carried out on the subject of stability of semi−submersibles. When, however, the Committee stated that there was little progress to report, we were referring to the much wider topic of the behaviour of ocean structures under extreme environmental conditions in general, and the prediction of extreme values based on model tests or computer simulations of limited duration in particular. The Committee has been unable to prepare a substantial contribution on this topic and therefore recommends the next Committee to pay special attention to this topic. It is noted that Dr. Takaishi is supporting draft recommendation 3.9 to the next Committee.

Professor Koterayama (OE−1), in his first comment, draws attention to one of the publications referenced in our report, which shows reduced drag with increasing ratio of current to wave orbital velocity. His own experiments, published at the PRADS Symposium in June this year, show the opposite. The Committee certainly agrees with Prof. Koterayama, that the combined effect of current and waves can influence the drag coefficient of a cylinder both ways.
It all depends on:
- orientation of the cylinder relative to current and waves;
- the relative magnitude of waves and current;
- Kc and Rn, etc.

Up to now, information on these dependencies is purely empirical, and due to the number of parameters involved the information may seem to lack systematic tendencies. At least this has been the case until now.
In connection with this we should like to draw attention to a paper by Huse and Muren [1]

which was published recently, a paper which attempts to provide physical interpretation and systematic order into the data. In this paper the idea is introduced of a drag coefficient which is physically the same in oscillatory flow as it is in stationary flow. In oscillatory flow, however, one has to correct the traditional inflow velocity for the wake generated by the previous cycles of motion. This correction can be done on the basis of turbulent wake theory together with momentum considerations. The interesting thing is that this theory now allows us to calculate theoretically the influence of superposed current and waves upon the drag forces.

Prof. Koterayama's paper at PRADS as well as Huse's OTC paper came too late to be discussed in the Committee report. However, it is supposed that the next Ocean Engineering Committee will pay due attention to these papers.

In his second discussion **Prof. Koterayama (OE−2)** refers to his paper at the OMAE Conference early this year, which also came too late for inclusion in the report. The Committee is glad to see that his findings are in agreement with those of the Committee, regarding the influence of mooring line damping upon the resonant surge motions of moored vessels.

Mr. Van den Boom (OE−8) asks for a quantification of the various components of the low frequency damping. The Committee agrees that such a quantification would be most valuable. A reasonably complete survey, however, would have to consider various types of offshore structures, various directions of motion, various environmental conditions, etc. etc., and would thus be very extensive. We regret that it has so far been beyond our capacity to do such an excercise. The best we can do is to pass the suggestion to the next Committee, with our full support indeed. We also agree regarding the need for full scale measurements.

The Committee very much appreciates the contribution from **the Bulgarian Ship Hydrodynamics Center(OE−3).** Undoubtedly our colleagues at BSHC have already had considerable success in calculating the drag on pontoons of semi−submersibles by a discrete vortex method. Such drag is of course very important in predicting low frequency damping. The Committee is somewhat surprised, however, that this drag should influence the sway motions of a non−moored semi in long waves as indicated in Fig. 2 of the discussion. We do not understand how this can be the case.

In this frequency range the platform follows closely the orbital motion of the waves and drag forces should be negligible compared to the total wave force.

The Committee thanks BSHC for providing more experimental data on both the linear motions and wave drift forces on the SR 192 semi−submersible and appreciates the efforts they made with regard to computation of motion characteristics. The results computed by BSHC show good correlation with experimental data.

This good correlation was obtained by using two−dimensional theory and applying some correction to account for three−dimensional flow effects. The results are in line with those of similar methods which are referred to as "group 2" in the report of the 17th ITTC Ocean Engineering Committee.

The Committee should like to thank **Prof. Saito and Dr. Higo (OE−7)** for their co−operation with the Committee in the follow−up survey to the 17th ITTC comparative calculation work. After completion of the present report they found that in order to avoid a large discrepancy a large enough number of panels is needed, at least, to represent the rounded corners of the lower hull section. This helps us to prescribe a minimum number of panels. We cannot, however, draw definite conclusions about improvement due to the increased number of panels at this moment. Although increasing the number of elements and modeling of the round corners improves the correlation between theory and experiment, there remains a discrepancy between calculated and measured heave motion. In studying this problem further, also viscous effects should be considered. The Committee will follow the continuation of these studies with much interest.

The contribution by **Prof. Choi and Prof. Kim (OE−5)** considers the special case of the slow drift of a floating horizontal cylinder in narrow−banded beam seas, which has been investigated as a two−dimensional problem. The discussers clearly support the conclusion in the Committee report concerning the relative importance of wave radiation damping, wave drift damping and viscous damping. Reference is made to a paper submitted for publication this year, which is not available to the Committee. It is not easy to say if conclusions made here for the special case of the sway drift motion of a cylindrical cross−section can be generalized for arbitrary body geometries and other motion modes. Especially the statement concerning the range of the magnitude of the wave radiation damping and the wave drift damping could not be followed by the Committee on the basis of the information now available. The Committee agrees with the authors, however, with regard to the great importance of the viscous damping in connection with slow drift motions.

The discussion by **Dr. Shimada (OE−6)** addresses an important problem within the second order diffraction theory, namely the fulfillment of the second order radiation condition. It is known that the physically correct formulation and the degree of satisfaction of this condition affect directly the quality of the second order solution. Also in the case of this discussion the corresponding publication came too late to be included in the Committee report.
The details of the theory developed by Dr. Shimada are in a paper presented at the spring meeting of the Society of Naval Architects of Japan in May this year. The Committee received this paper yesterday. The Committee agrees with Dr. Shimada that the second order radiation condition of the Sommerfeld type should be imposed on the free wave potential only and not on the locked wave potential. Molin formulated in 1979 a solution which uses far−field expressions of the second order velocity potential, Green's third identity and Haskind's reciprocal relations. Molin regarded also the second order diffraction waves as consisting of two components, the "free waves" which behave independently of the first order wave system as outgoing waves travelling at infinity, and the "phase−locked waves" accompanying the first order waves. Dr. Shimada used for the second order diffraction problem simple sources or boundary elements and eigen functions in a similar manner as adopted by Yeung for linear problems. It is very interesting to see that this approach works, at least for the special case considered here.
A numerical analysis has been carried out for an infinitely long vertical cylinder. The results are in remarkably good agreement with those presented by Hunt and Baldour. We should like to recommend Dr. Shimada to give, probably in his next paper, more information on the computational expenses of his method in order to demonstrate more clearly its advantages. Furthermore, comparisons with other results obtained by Garrison, Molin and Rahman are recommended. In this context, the extension of Dr. Shimada's method to cylinders in finite water depth, bottom mounted and piercing the water surface or floating vertically, is of direct practical importance. Especially the case of the floating vertical cylinder, which has been investigated by Molin last year, could be a good subject for the extension of this approach.

Concerning **Mr. Van den Boom's (OE−8)** remarks we agree with him with regard to the importance of the correct choice of fluid force coefficients within the non−linear analysis of the dynamic behaviour of mooring lines. These coefficients can only be obtained from experiments. It is also our opinion that the finite difference methods associated with the lumped mass discretization are an efficient tool for the numerical analysis of flexibles. This is clearly indicated in the Committee report. However in several cases both the simplified semi−analytical methods as well as more complicated finite element techniques could be useful too. For example, the bending stiffness of a flexible riser being neglected in first approximation, catenary equations can be used, by means of which the shape and tension over most of the riser length can be calculated. The bending moment and the associated curvature at the ends can be calculated using beam theory expressions. On the other hand, for some complex flexible riser systems in which information about the situations in certain points, i.e. in critical components (connectors, terminators, bundles, bending restrictors) is required, finite element techniques, especially in the hybrid formulation, can be very efficient.

Further experimental and theoretical work is needed for flexible risers, in particular considering bundles and more complex systems.

The method **Mrs. Sun Yi−Qing (OE−9)** mentions in her discussion, is a lumped mass discretization, developed together with Mr. Van den Boom. This method has been discussed in the Committee report. We agree with Mrs. Sun Yi−Qing that these methods can lead to the estimation of the maximum bending stresses in the line in the deterministic way, and from there, in a stochastic concept. The statistical distribution of the stresses along the riser is of direct importance for the designer. Experience shows, however that the deterioration and fatigue occur at the points of local stress concentration, i.e. primarily at the terminators, connectors, bundles, flanges, bending restrictors etc. The mechanisms of deterioration and fatigue for the flexible risers used today are very complex and lay beyond the scope of the report of this Committee. The structural type of the pipe is decisive for this mechanism. However, the results of the global dynamic analysis of the riser are the basis for the computation to estimate the operational life of a flexible riser.

Prof. Gu and Dr. Zhou (OE−11) address two subjects in their discussion. The first is a method to separate the group−bounded long waves (also referred to as the set−down), and reflected and parasitic long waves. The second subject deals with calculation of the long wave set−down spectrum directly from the first order wave spectrum. Concerning the first subject, the Committee agrees that the presence of parasitic and reflected long waves creates problems in extrapolating model test results to full scale, when testing systems such as moored vessels which are sensitive to long−periodic excitation. The results presented indicate, that the long−periodic excitation caused by reflected and parasitic long waves is of the same order of magnitude as that due to the natural set−down. However, the Committee wants to emphasize, that the set−down is only one of several contributions to the total long−periodic wave drift force, and that the effect of the set−down is more pronounced in shallow water than it is in deep water.

In the second part of their discussion Prof. Gu and Dr. Zhou compare significant values of the set−down obtained from the developed 3−probe correlation method with those from square law operation. The good agreement between experimental and theoretical results supports the validity of the 3−probe correlation method developed by CSSRC. We look forward to read the description of the method, which is to be published in the Ocean Engineerng magazine, and the Committee hopes that other members of the ITTC community will use the method to evaluate the influence of the different long wave components in model tests with moored offshore structures.

Dr. Pinkster (OE−8) asked the Committee for some guidelines regarding the duration of model tests in irregular waves in order to obtain statistical reliability of the results when testing moored offshore structures where the mooring loads contain low frequency components.

When first order wave−induced motions and loads are investigated a model test duration long enough to obtain 100−300 cycles is considered to be sufficient. For the second order motions and loads the same number of cycles is needed, thus as a minimum, 100 cycles.

This means, for example, when testing a mooring system with a resonance period of 3 minutes in full scale, with a scale factor of 50, a model test duration of at least 45 minutes. This is in most cases not acceptable due to the high costs involved. At such very long test durations one should also be aware of possible problems of reflections from beaches or models. To overcome these problems it might be possible to combine experiments and computer simulations. The Committee is aware of a recent paper by Dr. Pinkster presented at OTC 1987 [2] which discusses the problem and the Committee will follow the continuation of this work with much interest.

Mr. Van den Boom (OE−8) discussed the importance of wind loads on offshore platforms, especially wind gusts may contribute to the excitation of resonant horizontal motions of moored and compliant structures and he is proposing that mathematical descriptions of wind forces should be encouraged by the ITTC. The Committee believes that in the coming years experiments will still be needed to obtain wind forces on complex structures. Wind spectra information is also needed. It might, however, be possible to establish semi−empirical formulas for estimation of wind forces.

Dr. Incecik (OE−12) has presented very interesting results from calculations of motion responses for an articulated tower and a coupled articulated tower and ship system in different commonly used wave and wind spectral forms. He has also investigated the effects of the responses of changes in inertia and damping coefficients.

The Committee agrees that all these factors are important and should also like to refer to the reply to Mr. Van den Boom that more information on wind spectra is needed. Dr. Incecik has also pointed out that the magnitude of the motion response of the articulated tower due to second order wave forces and dynamic wind loading is of the same order of magnitude. The Committee is aware that this may be the case for several types of offshore structures.

References

[1] Huse, E. and Muren, P.: "Drag in Oscillatory Flow Interpreted from Wake Considerations", OTC Paper No. 5370, Houston, 1987.

[2] Pinkster, J. A. and Whichers, J. E. W.: "The Statistical Properties of Low Frequency Motions of Non−Linearly Moored Tankers", OTC Paper No. 5457, Houston, 1987.