

# CHAPTER 163

## SUBHARMONIC COMPONENTS IN HAWSER AND FENDER FORCES

by

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### ABSTRACT

Forces in mooring lines and fenders of a moored vessel exposed to waves have a mixed harmonic and a subharmonic character. The subharmonic oscillations, with periods well beyond the range of wave periods, may cause forces that are as large as, or even larger than the forces associated with the harmonic oscillations.

The origins of the subharmonic oscillations are discussed and it is shown that in model testing, the correct reproduction of both the mooring arrangement and the irregular wave motion is essential.

The subharmonic motions, and consequently the total forces, can be partially reduced by specially adapted mooring arrangements as tests on various mooring systems have shown.

### 1. INTRODUCTION

During the past few decades there has been a rapid development in ship sizes, accompanied by the advent of highly specialised ships, such as LNG carriers, container vessels, etc.. This development has created many challenging problems, not only for nautical engineers in designing the vessels, but also for harbour engineers in providing adequate berthing facilities.

The increase in ship size led to the design and construction of loading and unloading facilities at more and more exposed locations in order to limit the dredging work and the costly construction of protection dams in deep water. Although at these exposed locations the docking itself will still be of importance for the actual design, the behaviour of the moored ship in waves has become an important, and often a determinative, parameter.

On the other hand, special mooring arrangements may be required, even in more or less protected areas, because the tolerable ship motions can be extremely small. For example container transshipment cannot take place if the motions of the container ship are appreciable, and LNG loading arms cannot follow very large ship motions.

The mooring and berthing facilities have to be designed so that the requirements regarding the allowable ship motions, and acceptable downtime, are met at tolerable forces. Wave forces on a ship may easily reach to thousands of tons, even in mild wave conditions. A certain amount of flexibility should,

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therefore, be built in to prevent prohibitively large forces in the mooring lines and fenders.

Clearly the optimum between the dynamic behaviour of the ship and the loads in the mooring system has to be assessed for each situation.

In addition to these developments, the expansion of activities offshore should be mentioned, where drilling, construction and transshipment operations are taking place in extremely hostile conditions, using installations for which the wave-excited forces and motions are extremely important.

In the present study, which is primarily focussed on the problem of a moored vessel, the character of the wave-excited motions of a vessel and the corresponding forces in the mooring system are dealt with. However, the conclusions may also largely apply to any other form of anchored floating object.

## 2. SHIP OSCILLATIONS

In studying the response of a moored ship to a random wave motion, the following types of oscillation can be observed:

- a. Harmonic oscillations.
- b. Superharmonic oscillations.
- c. Subharmonic oscillations.

Ad a. The harmonic motions are the oscillations of the ship (6 components) with frequencies in the range of the wave frequencies. The harmonic wave forces and the harmonic ship motions of a freely-floating ship are generally fairly well linearly related to the wave amplitude. This is even so for the small keel clearances that are usually present at the berths, if only moderate wave amplitudes are considered. As an example one is referred to Figure 1 where the transverse motions (combined roll and sway) at about deck level of an ore carrier at a water depth 17% in excess of the draught are presented. The wave height of the regular waves ranged between about 1.0 m and 2.5 m, and the mooring system was linear. However, if the ship is tightly moored in a non-linear mooring system, the linear approach may no longer be justified.

Ad b. For particular elastic characteristics of the mooring system, oscillations with frequencies higher than the exciting wave frequency (superharmonics) may occur.

Ad c. Finally, subharmonic motions can be observed. These subharmonics have frequencies much smaller than those of the exciting waves. The subharmonic component in the motion or the force often forms a substantial part of the total motion or force. The subharmonic character can even become predominant, and is therefore clearly of importance for practical designs.

In Figure 2, derived from Reference 1, some typical recordings are presented of springline and fender forces for a conventional mooring system, containing appreciable subharmonic components.

The subharmonic motions can have the following origins:

- The non-linear and asymmetric elastic character of mooring lines and fenders. Regular harmonic excitation of a mass in a non-linear and asymmetric spring system leads to frequencies different from the excitation frequency, also lower ones. If instead of one single excitation frequency, two discrete frequencies are applied, the resulting motion can also contain so-called combination tones (Ref. 9). In an actual sea condition, with a wave-energy spectrum covering a certain frequency band, these combination tones will show up as well.
- The second order slowly varying wave drift force. Due to this wave drift force subharmonic motions will occur, also with a strictly linear mooring system. However, only in irregular waves. Figure 3 shows an example of a model ship subjected to head waves, where the surging motion contains a predominant subharmonic component.
- Long periodic waves, like seiches and surf beats, may cause subharmonic motions,

particularly if the harbour configuration tends to amplify these phenomena (Ref. 8).

As the slowly varying wave drift force is an important parameter (and often the dominant one in this respect), this drift force will first be discussed.

### 3. THEORETICAL DESCRIPTION OF WAVE DRIFT FORCE

The study of the wave drift force was originally tuned to the determination of the wave-induced resistance of ships, but in recent years, the computations have been expanded to determine the behaviour of moored vessels. Havelock in the early 1940's published an approximative calculation of the steady drift force (Ref. 4). His computation was based on the observation that the drift force is most predominant near the natural periods of oscillation. It was shown that interaction between first-order effects, which in themselves are purely periodic, may through phase differences give rise to steady drift forces. For head waves:

$$F_D = \frac{1}{2}k(P.z \sin \beta_z + Q.\psi \sin \beta_\psi), \quad (1)$$

where P and Q are amplitudes of buoyancy and pitching moment, z and  $\psi$  are amplitudes of heaving and pitching, and  $\beta_z$  and  $\beta_\psi$  are phase lags. Comparable expressions have been derived by others for beam waves.

According to Reference 14 the interaction as described by Havelock only partly explains the steady drift force. The effects of interaction between the waves diffracted by the ship and the incoming waves, as well as the effect associated with the energy dissipation due to outgoing wave radiation resulting from forced ship motions, should be added. To obtain a less complex approach, Hsu and Blenkarn (Ref. 5) developed a description of the drift force, based on the same concept as the radiation stress. The conservation of wave momentum requires that if a wave train is reflected from an obstacle, there is a force exerted on that obstacle equal to the rate of change of wave momentum. The expression for the average drift force on a ship per unit length in regular beam waves becomes:

$$F_D = \frac{\rho g a^2}{2} \left\{ 2(Q-1) \left[ 1 + \frac{z}{a} \cos(\beta_z + kX_1) \right] - Q \frac{z}{a} \left[ \cos(\beta_y + kX_1) + \frac{z}{a} \cos(\beta_y - \beta_z) \right] \right\}, \quad (2)$$

where  $Q = [2 \cosh k(d-T)] / \cosh kd$  and  $X_1$  is the average position of the leeward side of the vessel with respect to the centre line. It is assumed that all wave momentum between the free surface and the keel is reflected (at the average position of the ship side) and that the wave field below the keel is not disturbed. Similarly, Maruo (Ref. 6) in an earlier paper applied the wave momentum analysis on a control surface at a great distance from the ship to find the wave resistance in head waves. This computation is also approximative, as the boundary condition at the surface of the ship does not enter the analysis. Moreover, the expressions derived are for deep water.

The most recent description of the wave drift force, introducing the actual boundary conditions at the ship's surface, is from Pinkster (Ref. 12). The total hydrodynamic force in k-direction on the ship (which is allowed to make small amplitude motions about the mean position) is:

$$F_k = \iint p \cdot n_k \cdot dS \quad (3)$$

The pressure p follows from the Bernoulli equation:

$$p = -\rho gZ - \rho \phi_t - \frac{1}{2}\rho(\nabla\phi)^2 \quad (4)$$

The various parameters are then expanded to the second order using a small parameter  $\epsilon$  (no higher orders, as the low frequency motions are supposed to be generated by the second-order force), so

$$\phi = \epsilon \phi^{(1)} + \epsilon^2 \phi^{(2)}$$

and

$$F_k = \iint \left( p^{(0)} + \epsilon p^{(1)} + \epsilon^2 p^{(2)} \right) \bar{n}_k dS + \iint \left( p^{(0)} + \epsilon p^{(1)} \right) \epsilon n_k^{(1)} dS + \iint p^{(0)} \epsilon^2 n_k^{(2)} dS + O^{(3)} \quad (5)$$

This integration can be carried out for the complete wetted surface S. After elaboration for the horizontal force in longitudinal direction (no roll, no sway) this expression gives:

$$F^{(1)} = \iint_{S_o} -\rho \phi_t^{(1)} \bar{n}_1 dS \quad (6)$$

$$F^{(2)} = \frac{1}{2} \rho g \oint_L \eta_r^{(1)} \bar{n}_1 dS + \psi^{(1)} \iint_{S_o} -\rho \phi_t^{(1)} \bar{n}_3 dS + \iint_{S_o} -\frac{1}{2} \rho (\nabla \phi^{(1)})^2 \bar{n}_1 dS + \iint_{S_o} -\rho \phi_t^{(2)} \bar{n}_1 dS - \rho g z_o^{(1)} \psi^{(1)} A_{wL} + O^{(3)} \quad (7)$$

Apparently the ship motions need only to be known to the first order, unlike the velocity potential. Also the actual wave elevation along the hull ( $\eta_r$ ) should be known. Pinkster concludes therefore that a proper calculation for a practical case is for the time being too complex to be carried out.

In regular waves the drift force is of a constant value. As can be seen in Figure 5, which applies for a model ship in beam waves, the ship drifts away after the waves start to grow, and subsequently attains a constant offset. If the wave amplitude varies, the drift force will vary more than proportionally, and the ship will perform a forced oscillation at a frequency equal to the wave frequency and at a frequency matching the frequency of the wave groups (see also Figure 5).

Various investigators have tried to assess the wave drift force and the resulting motion in the general case of irregular waves, on the basis of the steady drift force as measured in regular waves. Hsu and Blenkarn (Ref. 5) computed the slowly varying drift force in irregular waves by assuming that the irregular sea can be characterised by a sequence of waves, each with a defined height, period and steady drift force, and that each wave acts on the ship as if it was one wave out of a sequence of regular waves. This can be written for a narrow-band spectrum in a general form (Refs. 10, 11) as:

$$\bar{F}_D = \rho g \int_0^\infty S_{\eta\eta}(\omega) R^2(\omega) d\omega \quad (8)$$

and

$$S_{FF}(\mu) = 2 \rho^2 g^2 \int_0^{\infty} S_{\eta\eta}(\omega) S_{\eta\eta}(\omega+\mu) R^4(\omega+\mu/2) d\omega \quad (9)$$

However, it has been shown that, particularly for larger vessels, this will yield incorrect results (Refs. 7, 10), because:

- The mean drifting force can be shown to be only dependent on the first-order potential, whereas the slowly-varying drifting force contains contributions of both the first- and second-order term of  $\phi$ .
- The effect of the dimensions of the vessel relative to the "wave length" of the wave group is not negligible.

Rye et al. (Ref. 13) show that, indeed, already different results are obtained if R is determined in regular waves or in wave groups with constant wave and group period. The same can be seen in Figure 5, where the subharmonic motions are much larger than would follow from the regular wave tests (the amplification due to the response of the ship is only 1.24). It is also pointed out in Reference 13 that the extreme values of the forces or motions are liable to be underestimated if they are derived from the spectrum values  $S_{FF}$ , as the distribution, due to the quadratic character of the drift force, is far from normal.

#### 4. EXPERIMENTAL STUDY

##### 4.1 Model set-up

In order to study various aspects of the subharmonic behaviour from a practical point of view, a variety of tests has been made with a model of an ore carrier, in an 8 m wide wind-wave flume at the Delft Hydraulics Laboratory. The wave flume is provided with a flap-type wave generator with the flap being driven by two electronically-controlled hydraulic actuators. The steering signal is random (not a summation of a finite number of components) but can be reset, after which the same sequence of random waves is repeated. The required input signal is obtained by filtering so-called pseudo-random noise (Ref. 8). For the generation of regular waves, a normal function generator is used.

##### Ship

The data of the ship were:

V	=	320,000	m <sup>3</sup>
L <sub>oa</sub>	=	350	m
L <sub>pp</sub>	=	330	m
B	=	55	m
T	=	20.5	m
d	=	24	m
C <sub>B</sub>	=	about 0.85	m

The model (length) scale was 1 to 100.

##### Mooring system

The mooring systems used are shown in Figure 4. Due to the use of a high pretension and long linear springs, no slack occurred and the system was purely linear and symmetric (apart from the head waves for system A). The elasticities were chosen so as to coincide reasonably with the overall elasticity of fenders and mooring lines of a conventional mooring system.

##### Waves

The waves applied were regular and irregular long-crested waves, both beam-on and head-on.

The regular waves had periods between 8 and 20 secs, the wave heights ranged be-

tween 1 m and 2.5 m.

The irregular wave had also significant wave heights between 1 m and 2.5 m. The periods corresponding to the peak of the wave energy spectrum were 12, 15 and 18 secs. At 15 secs a narrower and wider spectrum was also used (see Fig. 8).

#### 4.2 Measurements in regular waves

In the tests with regular waves, the constant drift force has been measured, with the symmetric mooring systems, and the results are shown in Figures 6, 7. Particularly for the beam waves there is a considerable scatter. This is partly due to the small values of the drift force and the inherent measuring inaccuracies, but mainly to the fact that the springs were connected above the center of rotation, at the actual application point of mooring lines and fenders. Consequently there were some secondary effects in the force measurements due to the rolling motion, in spite of the long lines used. Nevertheless, a pronounced increase of the drift force can be observed for beam waves in the period range of 14 to 16 secs (natural periods of roll and heave), obviously due to the interaction between the harmonic ship motions and the wave-excited forces, as mentioned in Section 3. Corresponding results have also been presented in Reference 3.

In head waves the drift force increases with increasing wave period, without any distinct maximum in the period range applied. The harmonic motions (surge, pitch and heave) also increase with longer periods in the corresponding frequency band. Clearly, when studying the subharmonic behaviour of a floating object much attention should be given to the correct reproduction of the mooring system.

Together with the steady drift forces, the harmonic motions have been recorded. From these motions and the proper values for the added mass and the damping, the harmonic wave-exciting forces have been computed. The averaged results for the various tests are also shown in Figures 6 and 7. The steady drift force in regular waves appears to be barely a few percent of the harmonic wave force. For instance, for a wave amplitude of 1 m, the drift force for beam waves is only 2-4%, whereas for head waves in that case the drift force is even less than 1%.

Finally the transferfunctions between the line forces and the waves have been determined for the mooring system A (Fig. 4).

#### 4.3 Measurements in irregular waves

##### Comparison with regular waves, and influence of wave height and period

Additionally to the measurements with regular waves, measurements have been carried out with irregular waves, both head-on and beam-on. In order to demonstrate the limited applicability of the linear transfer functions, and the regular wave tests in general, the 1% exceedance values of the wave-induced forces have been determined from the irregular wave tests, and compared with the 1% values that are obtained applying a linear transfer function  $T_{(\omega)}$  between waves and forces. The results have been plotted in Figures 9 and 10, and some typical spectra, applied for both the computations and the measurements, in Figure 8. The differences between the computed and measured values are fairly well proportional to  $H^2$ , as could be expected theoretically. This also means that the shape of the statistical distribution curve does not change significantly for different wave heights. The differences also increase with increasing wave period, and are particularly for  $T_p = 18$  sec very pronounced. The computed 1% values followed from:

$$S_{FF}(\omega) = S_{\eta\eta}(\omega) \cdot T_{(\omega)}^2$$

and according to the Rayleigh distribution,

$$F_{1\%} = 3.0 \sqrt{M_{OF}} \quad \text{where} \quad M_{OF} = \int_0^{\infty} S_{FF}(\omega) d\omega$$

From some of the recordings with irregular waves, spectra of the ship motions (sway for beam waves and surge for head waves) and the waves have been made (see Fig. 12). The spectra of the motions shown on this Figure are typical. They contain two distinct maxima, one in the harmonic range and one in the subharmonic range, where there is no wave energy. The energy content of this subharmonic part  $M'_{OF}$  has been determined for all tests and normalised by the square of the total energy of the wave motion  $m_{on}^2$ , according to the supposition expressed in (9).

The  $M'_{OF}/m_{on}^2$  can then be supposed to be a function of the peak frequency of the wave spectrum and the energy distribution (shape of the spectrum) for a particular situation. The test results are plotted in Figure 14. It is interesting to note that the value  $M'_{OF}/m_{on}^2$  is a fairly smooth function of the peak frequency of the spectrum. This is particularly surprising for sway, as the value of R in (9) varies very strongly with the wave period (Fig. 6). The spectrum width has not a large influence on this parameter. For beam waves the narrow spectrum results in the largest motions, as opposed to the situation with head waves. The same can be observed in Figures 9, 10 with respect to the subharmonic component of the 1% wave-induced force.

#### Influence of mooring system

The steady drift force in regular waves has been shown to be only a few percent of the harmonic wave forces. Although the regular-wave tests tend to underestimate the drift forces, the slowly-varying drift force in an actual case with irregular waves is still of second order. So the reason for the large subharmonic motions can only be found in the occurrence of a resonance type of motion of the vessel in its mooring system.

The conditions for resonance are available, as the natural frequencies of oscillation are within the band-width of the slow drift excitation and, moreover, the damping (both linear and quadratic) is extremely small in the low-frequency range (see Fig. 11<sup>\*</sup>). If the low damping is partly responsible for the large subharmonics, it may be expected that improvements can be obtained if special elastic characteristics of mooring lines and fenders are used. In particular if some energy dissipation is applied. With regard to this, some tests have been carried out with other than linear mooring systems. Some typical recordings are presented in Figure 15 together with the wave recordings. For each test the same motion control programme for the wave machine has been applied. Apart from minor details, the wave recordings were similar. The mooring system was changed in each test. The upper curve (System I) shows a subharmonic surging motion, which is reduced for System II where some energy dissipation was built in. In System III the restoring force at positive displacements was constant at a value of 400 tf (average of forces recorded for System I), moreover an appreciable energy dissipation was built in. Nevertheless, the subharmonic motion increased dramatically, but only for parts of the recording. In the remaining parts the subharmonics were of the same order as, or smaller than those for System I.

It is surprising that there was no direct correlation between the occurrence of these extremely large subharmonics and those for System I, as also can be seen in Figure 15. So the adverse effect is not only very large, but also fairly unpredictable. This is the more important, as System III is a type of system that may be found in practice, applying the widely-used constant tensioning winches,

\* The damping and added mass coefficients have been derived from data for a ship which is only slightly smaller, and are expected to be accurate within 10%.

with different pull-in and pay-out loads. In Figure 13 the energy spectra of the motions are given for the various mooring systems. In beam waves, the motions for System III became so large that they were beyond the range of the displacement transducers, and could not be analysed.

#### Influence of a steady force

Not only will the elastic properties of the mooring system depend on the particular types of lines and winches applied, but they may also vary for a particular lay-out as a result of an extra "pretension" caused by a steady wind or current. The effect on the ship motions can be quite significant, as shown by Hsu and Blenkarn (Ref. 5). The wind- and current forces, if present, should therefore always be reproduced in the model experiments. Due to the subharmonic motion, the relative speed of the wind or the current will vary, which may also affect the subharmonic motion in another way. Let  $U$  be the permanent velocity of wind or current, and  $\dot{x}$  the momentaneous ship velocity, the resistance is then,

$$F_R = \frac{1}{2} \rho C_d A (U - \dot{x})^2 = \frac{1}{2} \rho C_d A (U^2 + \dot{x}^2) - \rho C_d A \dot{x} U$$

if  $U$  is a constant, the last term at the right hand side has the character of a linear damping,  $2F_R \dot{x}/U$ . Due to the higher value of  $U$  this virtual damping will for the wind generally be rather small. However, for a steady current this term may be of the order of magnitude of the linear damping in the subharmonic range as shown in Figure 11. In that case it will be required to reproduce not only the steady current force, but the whole current pattern.

#### 5 CONCLUSIONS

Subharmonic components form, particularly in more exposed locations, a substantial part of the total horizontal ship motions or mooring forces. The second order slowly varying wave drift force is an important parameter in this respect.

The slowly varying drift force and consequent subharmonic motions occurring in irregular waves are larger than follow from the steady drift force in regular waves. Moreover, the so-called combination tones in irregular waves may introduce subharmonic motions that will not be found in the case of regular waves.

The second-order drift force in regular waves depends strongly on the harmonic ship motions, as a result of interactions between incident, diffracted and radiated waves.

As opposed to the harmonic motions, subharmonics can to some extent be reduced by applying special elastic characteristics for the mooring system, or more particularly energy dissipating devices. The effect of a variation of the elastic properties, however, seems to be fairly unpredictable.

In irregular waves the subharmonic components increased proportional to  $H_s^2$ . They also continuously increased with increasing peak period of the energy spectrum ( $T_p$ ), for both sway and surge.

LIST OF SYMBOLS

A	- area	p	- pressure
$A_{WL}$	- area of water plane	$R_{(\omega)}$	- transfer function
a	- amplitude of wave	$S_{xx}$	- energy density of x
B	- breadth	T	- wave period; draught
$C_B$	- block coefficient	$T_{(\omega)}$	- transfer function
$C_d$	- resistance coefficient	t	- time
c	- spring constant	U	- wind or current velocity
d	- depth of water	X	- horizontal coordinate
$F_D$	- drift force	x	- surging motion
$F_k$	- force in k-direction	y	- swaying motion
$F_R$	- resistance	z	- heaving motion
f	- frequency	Z	- vertical coordinate
g	- acceleration of gravity	$\beta$	- phase lag
$H_s$	- significant wave height	$\epsilon$	- small parameter
k	- wave number = $2\pi/\text{wave length}$	$\phi$	- velocity potential
$L_{oa}$	- length over all	$\eta$	- surface elevation
$L_{pp}$	- length between perpendiculars	$\rho$	- mass density
$M_{oF}$	- total energy of harmonic force	$\psi$	- pitching motion
$M'_{oF}$	- total energy of subharmonic force	$\omega, \mu$	- angular frequency
$m_{o\eta}$	- total energy of wave motion		
$n_k$	- directional cosine of surface element in k-direction		

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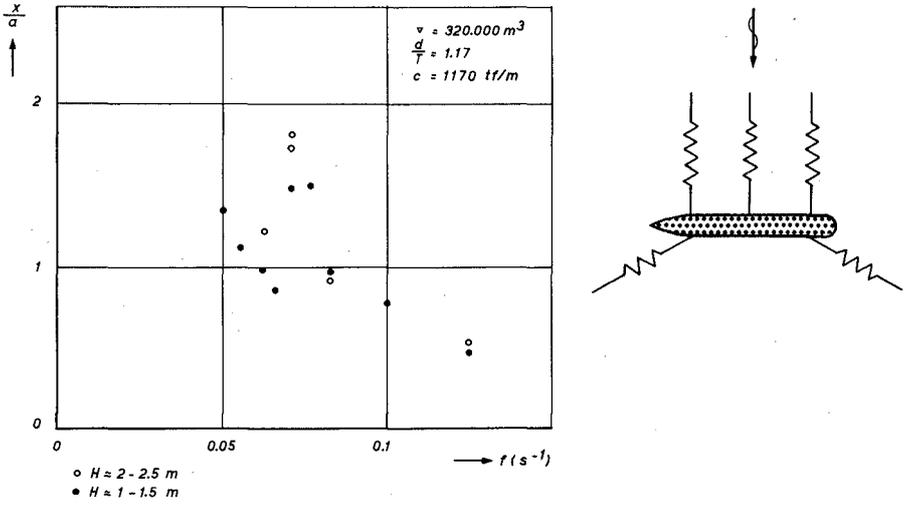


FIG. 1 HORIZONTAL MOTION OF SCHIP SIDE AT 5 m ABOVE WATER-LINE, IN IRREGULAR BEAM WAVES

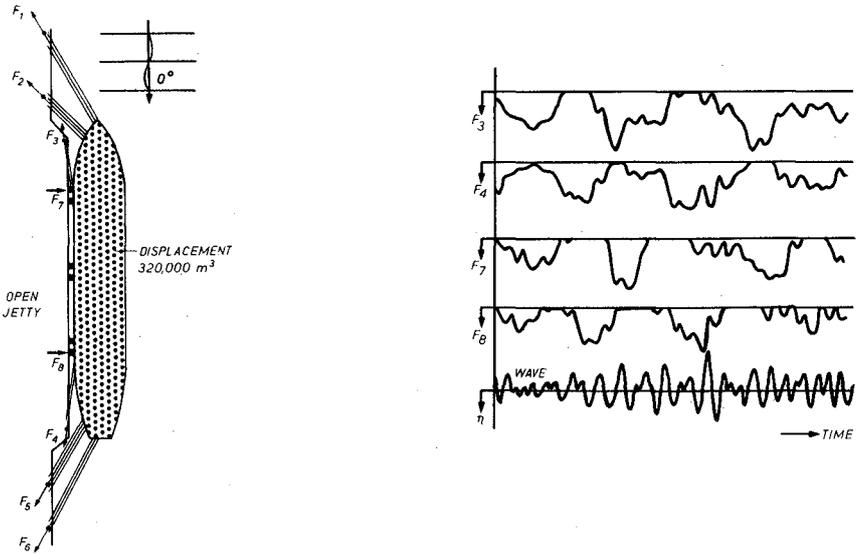


FIG. 2 IRREGULAR WAVES, TYPICAL FORCE RECORD

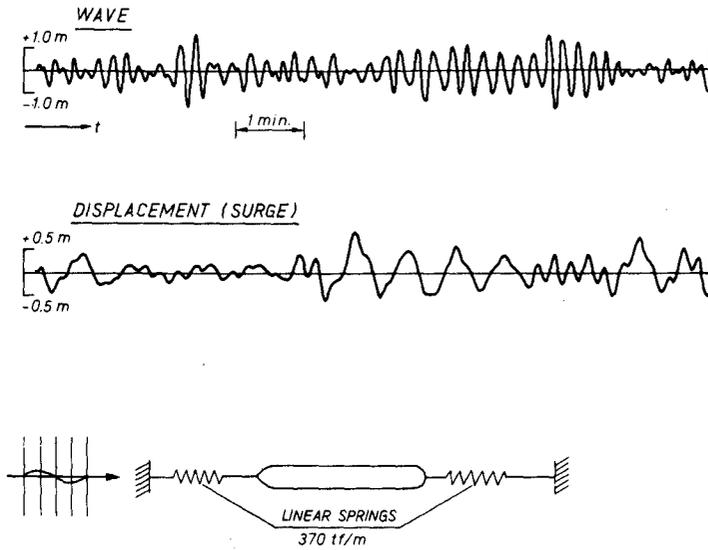


FIG. 3 TYPICAL SURGE RECORD FOR IRREGULAR HEAD WAVES

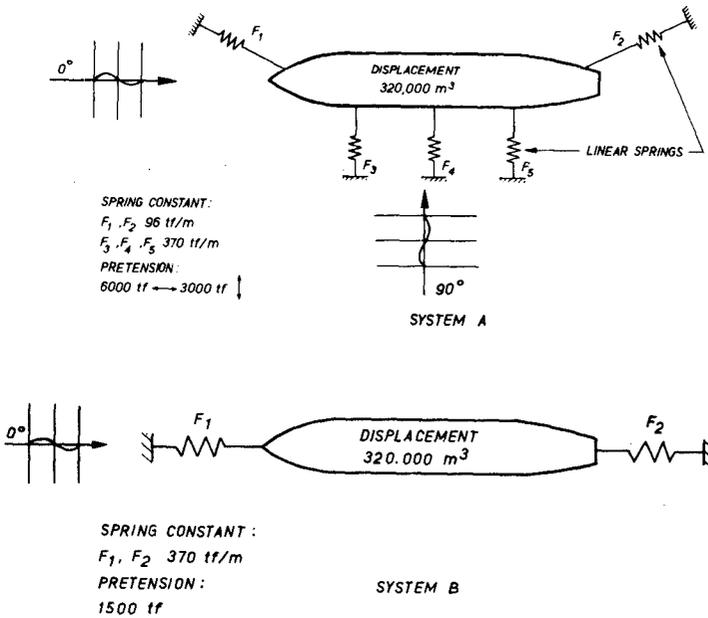


FIG. 4 MODEL SET-UP

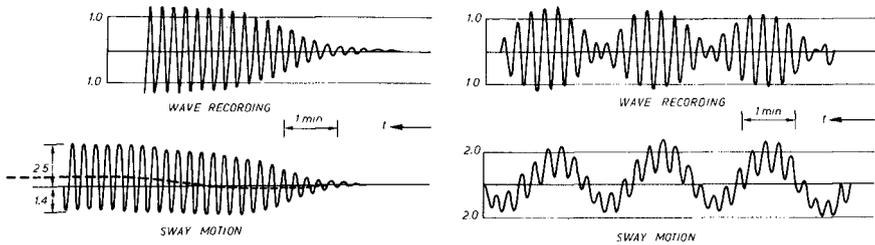


FIG. 5 SWAY MOTION IN REGULAR WAVES AND REGULAR WAVE GROUPS

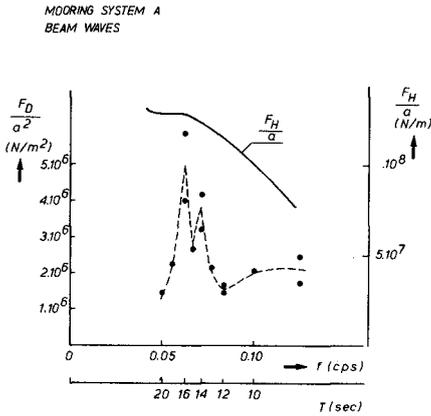


FIG. 6— WAVE DRIFT FORCE FOR SWAY, FROM REGULAR WAVE TESTS ( $F_D$ )  
— HARMONIC WAVE FORCE FOR SWAY ( $F_H$ )

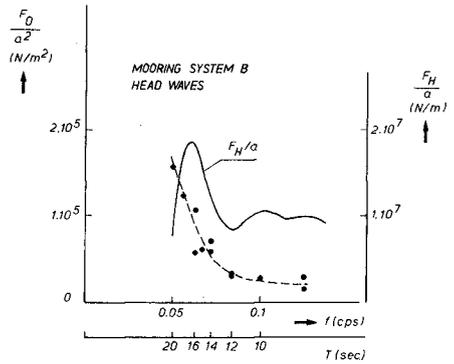


FIG. 7— WAVE DRIFT FORCE FOR SURGE, FROM REGULAR WAVE TESTS ( $F_D$ )  
— HARMONIC WAVE FORCE FOR SURGE ( $F_H$ )

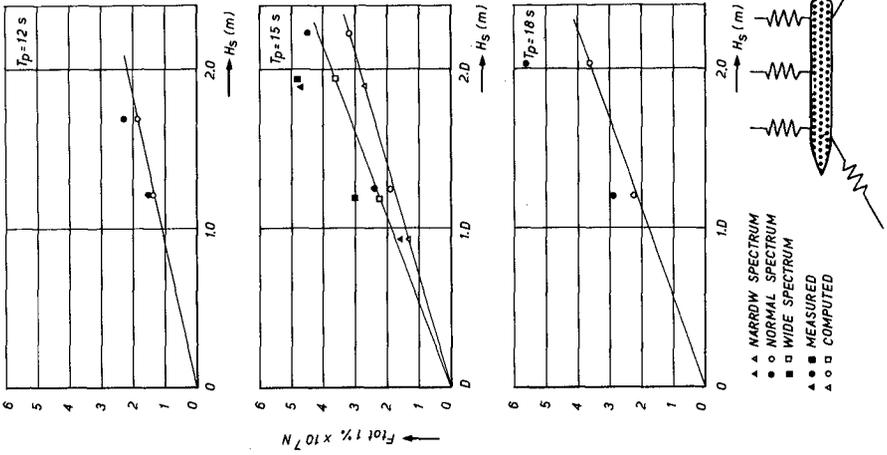


FIG. 9 TOTAL WAVE INDUCED FORCE VERSUS WAVE HEIGHT BEAM WAVES

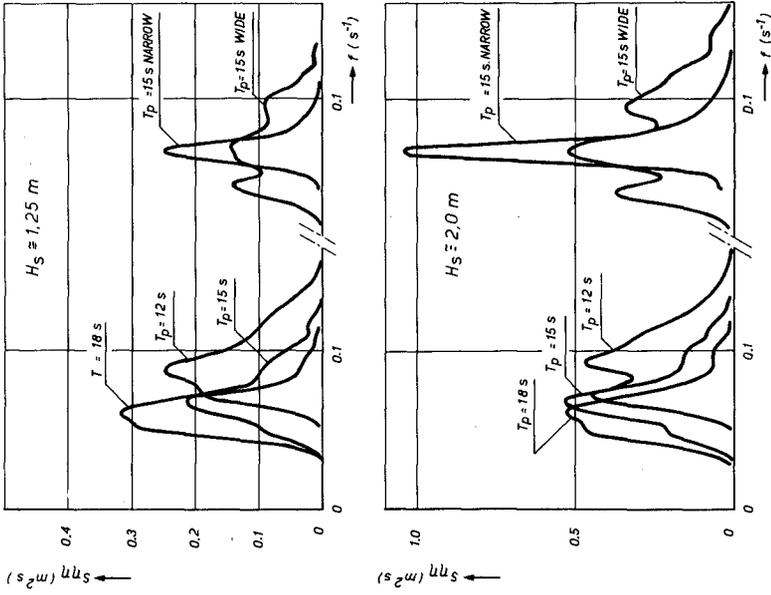


FIG. 8 TYPICAL WAVE SPECTRA FOR COMPUTATION AND MEASUREMENT OF WAVE INDUCED FORCES

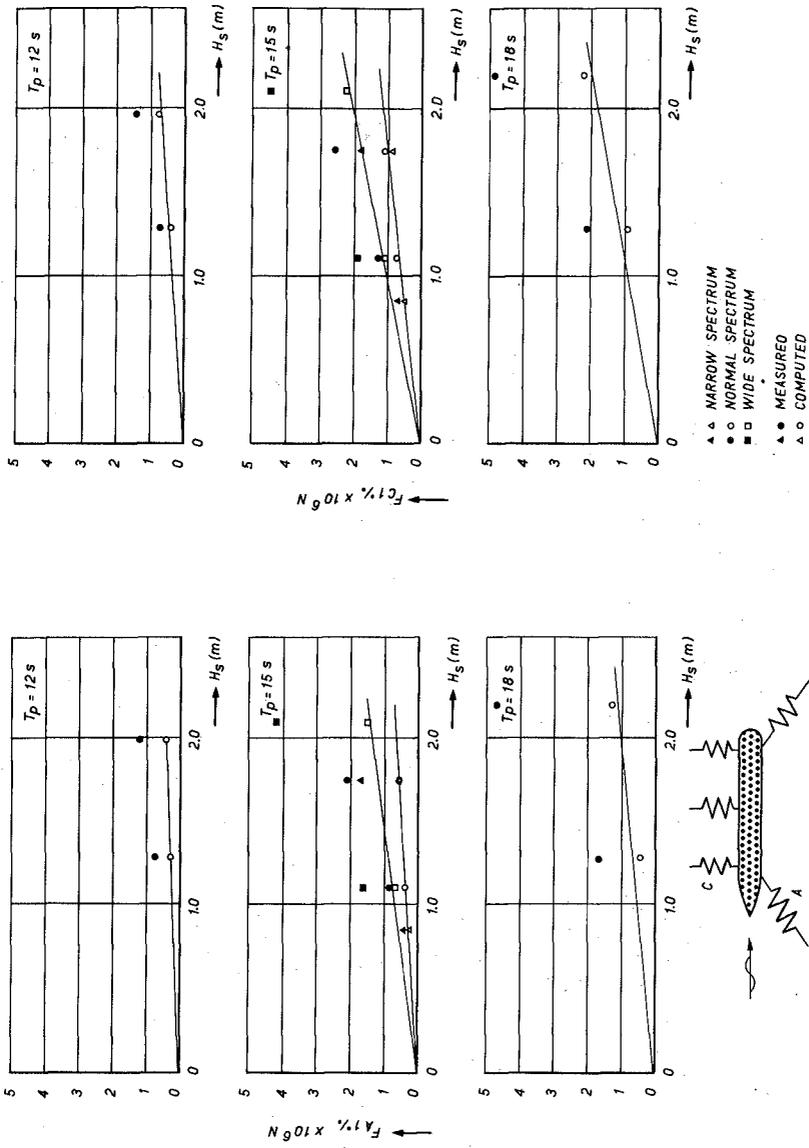


FIG. 10 TOTAL WAVE INDUCED FORCE VERSUS WAVE HEIGHT, HEAD WAVES

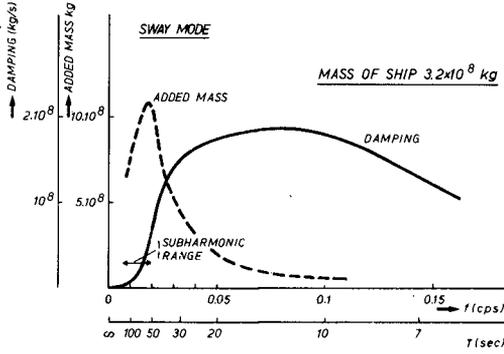
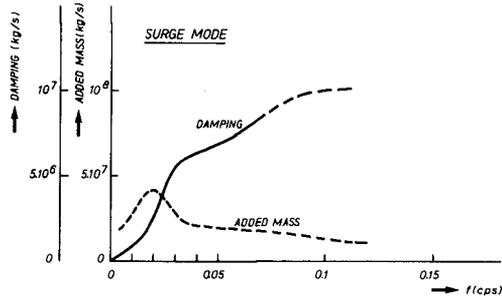


FIG. 11 ADDED MASS AND DAMPING FOR MODEL SHIP DEPTH/DRAUGHT = 1.17

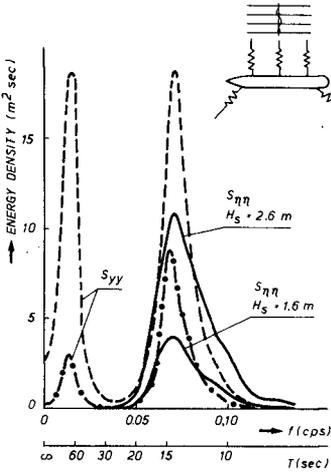


FIG. 12 ENERGIE SPECTRA OF WAVES AND SWAY MOTION

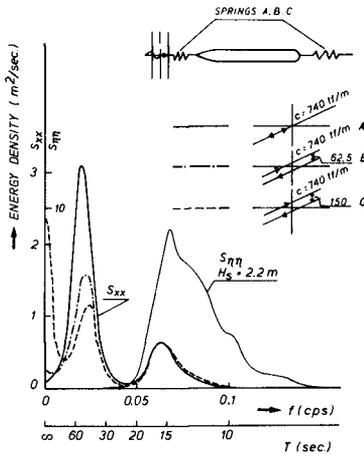


FIG. 13 ENERGY SPECTRA OF WAVES AND SURGE MOTION

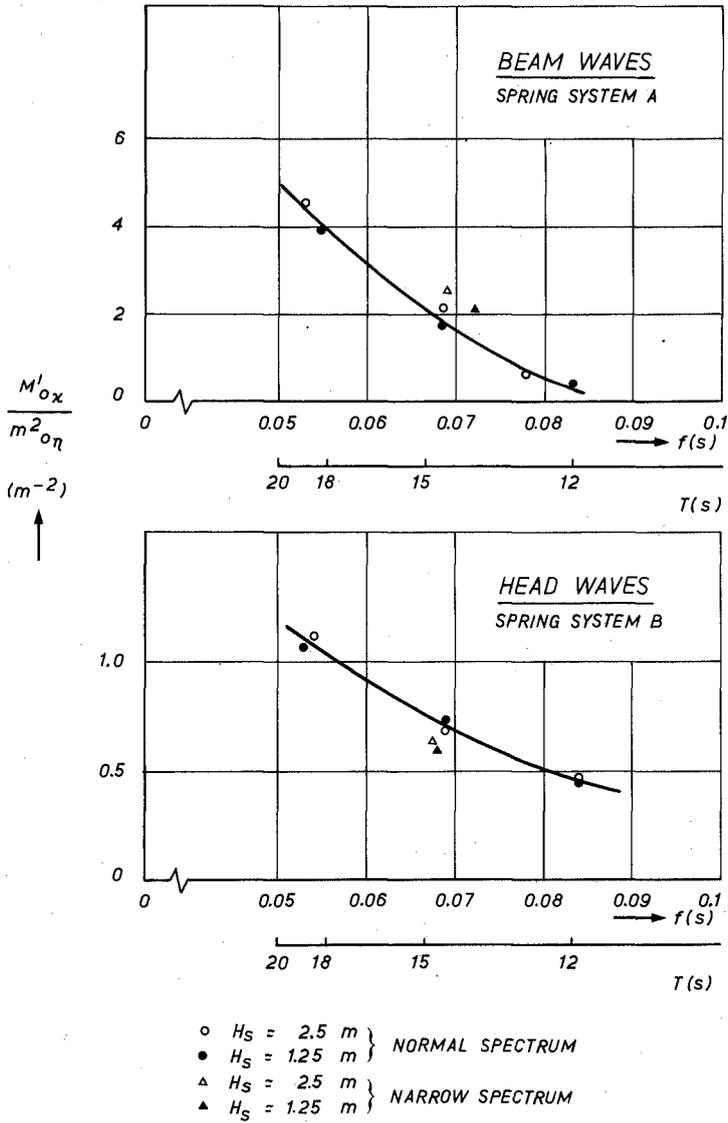


FIG. 14 NORMALISED ENERGY OF SUBHARMONIC MOTION

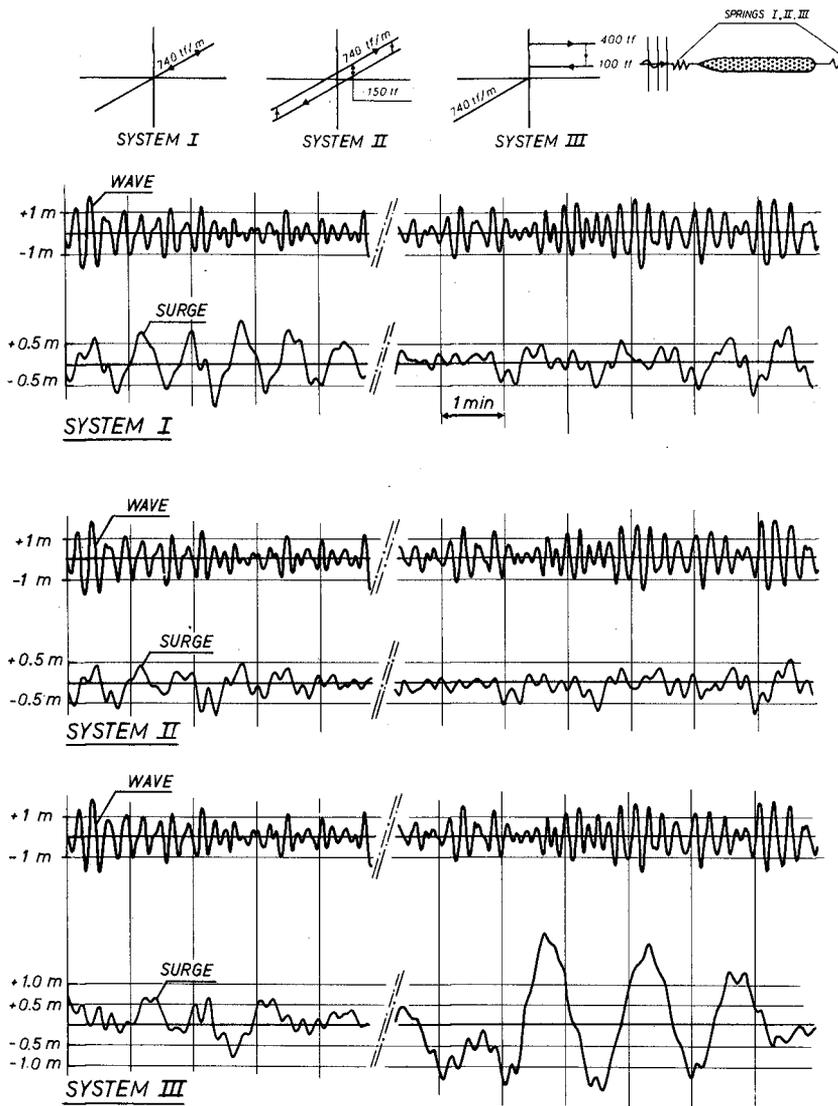


FIG. 15 SURGE RECORDINGS AT SAME PLACES IN WAVE SEQUENCE