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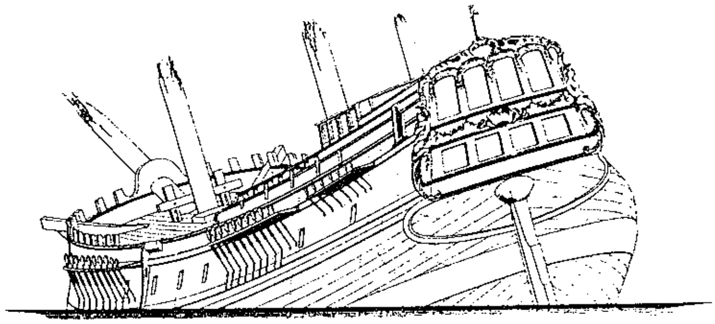

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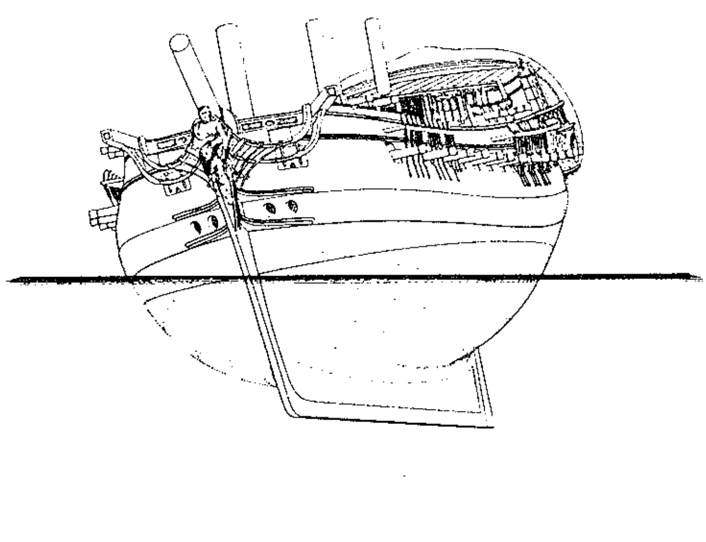

LOSS OF STABILITY OF SHIPS IN FOLLOWING WAVES IN RELATION TO THEIR DESIGN CHARACTERISTICS

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ABSTRACT

Pure stability loss on a wave crest may represent a potentially dangerous situation to a slender, high-speed vessel in following waves. The ship parameters affecting this phenomenon and selected for further parametric study are the following: ship length L_{WL}(m), ship speed by Froude number F_N(-), non-dimensional wind heeling lever l_W(-), flare by vertical prismatic coefficient C_F(-) and dynamic stability by righting energy E_o (mrad).
A modular mathematical model has been developed to estimate the "probability of capsizing" due to this phenomenon. The mathematical model takes into account the heaving and pitching motions of the ship, Smith effect, and ignores for the time being the surge motion and oblique wave directions. The next step consists of the calculation of the time lapse τ during which the residual dynamic stability is negative and the threshold value τ_o for an assumed capsizal. The last stage comprises a computation of long term statistical properties for a given distribution of wave spectra. For this purpose the roll response is considered to be a narrow band process allowing a quasi-harmonic approach of the problem.
Assuming an arbitrary safety level the systematic calculations lead to a stability criterion reflected by the general expression:

$$E_o \text{ (required)} = C_o . \text{Function } (C_F, F_N, l_W, L_{WL})$$

where C_o is a "matching coefficient" to be determined on the basis of casualty statistics and test calculations. The conceptual approach is pragmatic and comparative in nature and the resulting proposal for a stability criterion in principle depends upon the governing ship parameters.

1. INTRODUCTION

A slender, high-speed vessel sailing in following waves may represent a potentially dangerous situation from the stability point of view. Generally a number of capsizing mechanisms are distinguished i.e.:

1. broaching-to, where the wave induced yawing moment exceeds for a considerable time the course-keeping ability of the ship and the ship experiences a violent, forced sheer often associated with the occurrence of large heeling angles.
2. parametric resonance (Mathieu instability), where the ship sailing in a group of almost regular waves builds up a very large rolling amplitude if the rolling frequency is a multiple of half the frequency of encounter.
3. pure stability loss on a wave crest, where the ship due to its slenderness experiences such a substantial reduction of her form stability that the ship may suddenly heel over without any preceding, significant rolling motion.

This study will exclusively deal with the phenomenon of pure stability loss on a wave crest. It may be assumed that the understanding of this capsizing mechanism is sufficiently advanced to describe the physical concept by means of a (modular) mathematical model. This offers the possibility to carry out a parametric study with the aim to develop a stability criterion as a function of relevant ship parameters.
In this connection it may be noted that much attention is paid to the subject on an international level. Quite recently this has resulted in a number of proposals, each based on a different approach, for additional stability criteria covering this specific situation. For further information with regard to these developments on an international level reference is made to [1], [2], [3] and associated papers.

The objectives of this study may be described as follows:

1. to describe the physical elements of pure stability loss on a wave crest in order to assess all essential parameters involved in the occurrence of this phenomenon.
2. to draft a computation procedure, taking account of acceptable simplifications, in order to determine numerically the probability of capsizing.
3. to carry out comparative calculations on the basis of a systematic variation of the parameters selected for this purpose.
4. to develop a stability criterion as a function of the relevant ship parameters on the basis of an assumed arbitrary safety level.

A summary of preliminary studies on the subject is given in [4].

From the problem definition it may be concluded that it is the intention to "isolate" the physical phenomenon in order to enable a quasi-static approach of the problem. This implies that oblique wave directions are ignored because of the following reasons:

1. there is no compelling need to establish the most unfavourable situation because the intention of the study is to assess in a quantitative sense the tendencies of the influence of the parameters from comparative calculations.
2. in the case of oblique wave directions the problem is more complicated as a complete treatment of the dynamics of the ship in a seaway is required. In particular the rolling behaviour, and therefore also the actual stability characteristics, are of paramount importance.

The present simplified approach is based on the assumption of the absence of a wave-induced heeling moment in pure following waves.

During the last four decades a good deal of information on the fundamentals of the subject has been published. An arbitrary choice of literature available, worthwhile for a thorough consideration of the phenomenon, is given in the references [5] - [10].

2. GENERAL CONSIDERATIONS

One feature of stability loss on a wave crest is among other things that the phenomenon occurs in a limited range of wave length to ship length ratios around $\lambda / L_{WL} \approx 1$ which means that the wave frequency acts as a relatively narrow filter.

From a close consideration of the "wetted" hull surface for several wave frequencies it may be recognized that the most critical situation coincides with a wave length approximately equal to the ship length and the wave crest located approximately amidships (see Figure 1).

Likewise it is obvious that the influence of wave height upon the extent of stability loss will increase with increasing wave height. This influence may be augmented considerably by a relatively low freeboard where the wave crest exceeds the level of the exposed deck with of course detrimental consequences for the instantaneous stability. If the flare of a cross-section of the hull is defined by the local derivative with respect to z (z = ordinate in upward vertical direction) the influence of wave height may be argued assuming that the flare is a positive function of z. Reference is made to [11] for a comprehensive discussion on the aspect of flare.

Various methods for the formulation of the stability of a ship are in existence. In this respect one meaningful way of presentation is:

$$GN\sin\varphi = \underbrace{\underbrace{MN\sin\varphi}_{\text{residuary stability}} + BM\sin\varphi}_{\text{form stability}} - \underbrace{BG\sin\varphi}_{\text{weight stability}}$$

In the particular case for very small angles the above expression is reduced to:

$$GM = BM - BG = (KB + BM) - KG$$

where KG performs a "regulating" function and the formula for the metacentric radius is: $BM = I_W / \nabla$.

The passage of a regular wave brings about, to first approximation, a harmonic oscillation of the levers of static stability. On the one hand this is attributable to variations of the instantaneous form of the waterline in the course of time and consequently corresponding variations in the transverse moment of inertia of the waterline, which finds expression in a periodic oscillation of BM around a mean BM-value and of GM around a mean GM-value. This has been represented in a schematic way in Figure 1. On the other hand this effect is compensated to a certain extent by a periodic oscillation of the momentaneous centre of buoyancy around a mean KB-value. From this approach, where the ship is assumed to be continuously in a position of static equilibrium in the wave, it may be derived that the mean KB-value in regular waves is not equal to the KB-value in still water, but exceeds this value. Likewise it may be made plausible that for increasing z an increase of flare will result into a BM-value in still water exceeding the mean BM-value in regular waves. Generally, but certainly not as a rule, this will result in a mean GM-value, which is less than the GM-value in still water. A schematic presentation of above considerations is given in Figure 2.

On account of the foregoing considerations it may be assumed that flare is an essential parameter of the geometry of the ship for the occurrence of this phenomenon. Besides the ship length in an absolute sense is a governing parameter because with this parameter the sensitivity with regard to the prevailing wave lengths and wave heights is determined. Herewith it may be observed that the relation between ship size and wave dimensions in a statistical sense is entirely defined by the wave spectra.

The precision of the above idealized view is diminished to a certain extent by the dynamics of the ship. The ship motions in the longitudinal vertical plane i.e. heave, pitch and surge affect the instantaneous position of the ship in the wave and by that the stability fluctuations in the course of time. In principle the surge motion may be represented by an harmonic oscillation with a period equal to the period of encounter, which leads to the phenomenon that the ship remains relatively longer on a wave crest and relatively shorter in a wave trough [12]. Principally the effect of this phenomenon is dictated by wave steepness and by ship speed, which culminates in the occurrence of the so called "surfing" or riding on a wave crest by which the ship is carried along with the wave ($F_N \approx 0.4$) for a certain time lapse. If the Froude-Krylov hypothesis is assumed to be valid it means that the presence of the ship in the wave does not disturb the wave profile. Then the dynamics of the wave may be taken into account by a simple correction of the hydrostatic pressure distribution in still water for the influence of the orbital velocity in the wave. The effect of the orbital velocity in the wave on the pressure distribution in the wave is known as the Smith-effect. However, if the potential function of the undisturbed incident wave is expanded with potential functions for wave diffraction and wave radiation a deformation of the wave profile along the ship length will occur. On the basis of [13] it may be assumed that the effect of wave diffraction leads to a reduction of the fluctuations of the stability levers in the order of some tens of percentage as a result of the decrease of the effective wave height.

Assuming that the dynamic stability E_o is a dominating parameter for the description of the stability characteristics other form parameters of the ship geometry (with the exception of flare and ship length) represent only a second order effect. In Figure 3 a qualitative indication is given of a variation in the shape of the curve of levers of static stability for a constant E_o-value. On the one hand it concerns an extreme variation of d/B or C_B and on the other hand an extreme variation of F/B (D/B) or the presence of superstructures. Generally it may be assumed that these parameters may substantially influence the process of stability loss on a wave crest, not only by the ship motions and the deformation of the wave profile, but above all by a considerable shift in the energy balance A_r (see Figure 6). For the elaboration towards a practical application for slender high-speed vessels it may be assumed that such exceptional variations of these parameters are of academic interest only. A peculiarity for this capsize phenomenon is the influence of superstructures. It appears that the position in the longitudinal direction plays an important role, which is illustrated in Figure 4 in a schematic way. This is obviated in [3] by taking into account the position of deckhouses and the like in the determination of the "effective" depth.

Pure stability loss on a wave crest should be dealt with in terms of an energy consideration in accordance with e.g. reference [2]. This means that an external moment M_e is assumed to be present for the estimation of the "residual dynamic stability". This external moment is effected by the wind heeling lever h_{wo}:

$$h_{wo} = \frac{M_e}{\Delta} = \frac{p_w l_w}{\gamma} \quad \text{with } l_w = \frac{A_w z_w}{\nabla}$$

and A_w = lateral windage area

z_w = vertical distance between the centre of wind pressure and the centre of the underwater lateral area (hydrodynamic reaction force at ~ 0.5 d)

p_w = wind pressure (which may be approximated to be proportional to B_o^3 where the Beaufort Scale Number is denoted by B_o).

The magnitude of the wind pressure in an absolute sense is of secondary importance because of the comparative nature of the calculations. The wind heeling lever h_w as function of the heeling angle is approximated by the relation:

$$h_w = h_{wo}\,(0.25+0.75\cos^3\varphi)$$

In the situation as described the ship is supposed to travel in longcrested, irregular following waves beam to the wind where the effect of shortcrestedness as a contribution to the external moment by way of a wave excited moment is neglected. It is observed that this situation may actually occur in practice where the ship proceeds in a directional swell perpendicular to the prevailing wind direction. One method to account for wind fluctuations is given in [14]. However, in line with the quasi-static approach it is deemed sufficient to account for the stochastic character of the wind load and to ignore the wind-induced rolling behaviour of the ship. From the foregoing considerations it may be concluded that the magnitude of the external moment is determined by the wind influence coefficient l_w (non-dimensional wind heeling lever).

The element of time is a significant aspect in the occurrence of the phenomenon as such. Likewise this may be distinguished clearly by making a comparison with the behaviour of the ship in head waves. In particular the frequency of encounter is decisive for the time period during which the ship is in a condition of reduced (dynamic) stability. This leads to the conclusion that ship speed is of essential importance because, in addition to the effect on the ship motions, with that parameter the time period of the "critical" stage is established. As an example of how the influence of ship speed can be accounted for in a meaningful way reference is made to the approach presented in [15].

In the same publication Grim introduces the concept of the so-called "effective" wave in order to cater for the probabilistic features of irregular, longcrested following waves. This hypothetical effective wave is a harmonic regular wave with a wave length equal to the ship length and a wave height equal to the average "effective" wave height. This average effective wave height is determined on the basis of the assumption that the variation of the curve of the levers of static stability, which is dependent on wave height in a non-linear manner, is approximately equal to the replacing effective wave amplitude of the considered irregular seaway, which thus is a stochastic parameter with the same statistical properties as the actual seaway. This approach offers also the possibility to provide for the influence of ship speed and ship length in relation to the wave spectrum in a quantitative sense. The method developed along these lines forms the basis for a stability criterion presented in reference [1]. However, preference is given to use the statistical properties of the stability itself instead of those of the effective wave height. Consequently for the purpose of this study the method presented in [16] will be used as the basis for this study.

3. PARAMETER SELECTION

On the basis of the foregoing considerations it is presumed that the following ship parameters are of importance for the occurrence of this phenomenon and consequently are considered for systematic variation in the parametric study:

E_o(mrad)	=	righting energy (dynamic stability) = area under the righting lever curve in still water.
C_F (-)	=	C_B/C_W = measure for the (average) flare as indicated in [4] and also known as the vertical prismatic coefficient.
F_N (-)	=	$v/\sqrt{gL_{WL}}$ = Froude Number, which represents the dominating time aspect in this phenomenon completely.
l_w (-)	=	$A_w z_w/\nabla$ = wind influence coefficient (non-dimensional wind heeling lever), which is a measure for the external capsizing moment.
L_{WL} (m)	=	ship length (on the waterline), which represents a measure of ship size in relation to the prevailing seastate.

Other ship parameters which affect this phenomenon indirectly, by the vertical ship motions and the deformation of the wave profile, are left out of consideration. These parameters describing the ship geometry such as C_B, d/B, F/B (D/B) are deemed to be of second order importance (having in mind a constant E_o-value) for these comparative calculations. As a matter of course considerations of a practical nature, viz. to keep the number of calculations within reasonable limits, are of importance in this respect. Finally the analysis of the results of the systematic calculations should lead to proposed criterion which may be expressed by the following general expression:

$$E_o \text{ (required)} = C_o \cdot \text{Function } (C_F,\ F_N,\ l_w,\ L_{WL})$$

In this expression C_o is a "matching coefficient" which has to be determined by means of test calculations. These test calculations should be carried out for a great number of vessels with a good safety record taking into account existing stability criteria. Besides a sample of a small number of vessels, which have foundered or have experienced a dangerous list due to this phenomenon, has to be investigated. Such a pragmatic approach for the assessment of C_o is necessary because the present state of the art does not allow an exact estimate of the capsize probability and the corresponding safety level.

4. COMPUTATION PROCEDURE

The computation procedure, summarized also in [4], may be considered to be made up of the following main components:

1. on the basis of a reference ship and a scheme of a systematic variation of the ship parameters affecting this phenomenon, the ship hull forms are generated corresponding with the alternatives of the scheme.
2. calculation of the vertical ship motions, heave and pitch, using a computer programme based on the modified strip theory where it is assumed that the presence of the ship in the wave does not disturb the wave profile.
3. calculation of the curve of levers of static stability, taking into account the Smith-effect, for several instantaneous positions of the ship in the wave around the position where the wavecrest is amidships.
   This calculation should be carried out for several values of wave length and wave height in order to determine in a later stage the relation between the time lapse τ of negative residual dynamic stability on the one hand and wave length/wave height on the other hand.
4. assuming an arbitrary external moment, defined beforehand by means of a systematic variation of the wind influence coefficient l_w, the value of τ is determined. In addition the threshold value τ_o is determined, which inter alia may be dependent upon wind speed and l_w.
   Thereupon the calculation procedure is continued by the estimation of the boundaries of wave length-values, as a function of wave

height, where the situation of negative residual dynamic stability leads to capsizing which is the case if $\tau > \tau_o$.

5. the next stage consists of the calculation of the probability of capsizing, defined as the average number of capsizings per unit of time, for a given wave spectrum. For this purpose the theory developed in [16] is applied. This theory is based on the assumption that the seaway is represented by an energy spectrum with a relatively narrow frequency band, so that the irregular waves may be characterized as quasi-harmonic. This assumption for a quasi-stationary approach allows for an approximate method for the computation of the desired statistical properties.
Additionally, for the purpose of a long term prediction, a standard frequency distribution for the occurrence of the seastates should be introduced.

ad. 1: The lateral profile of the reference ship, including main particulars and other relevant data, are given in Figure 5. The results presented in the references [17] and [18] are also based on the same reference ship. The ship parameters E_o, F_N and l_W may be varied in a systematic manner without any consequences for the basic shipform. That is also true for a variation of L_{WL} since the form parameters remain unchanged. As F_N remains unchanged as well this means that shipspeed (in an absolute sense) is co-varied with $\alpha^{1/2}$ where α = scale factor. A systematic variation of C_F however requires a transformation of the basic shipform. For a constant displacement this means a variation of C_W.

ad. 2: Since the systematic calculations in principle are in principle a sensitivity analysis the computation procedure is simplified by neglecting the effect of the surge motion and the diffraction phenomena due to the presence of the ship in the wave.

ad. 3: In the calculation of the curve of levers of static stability the effect of the orbital velocity of the water particles in the wave, which results in a hydrodynamic pressure distribution different from the hydrostatic pressure distribution, is taken into account.

ad. 4: The residual dynamic stability A_r and the quantity τ to be derived from A_r are defined and clarified in the conceptual diagrams of Figure 6. As no allowance is made for the wave excited rolling motion it is impossible at this stage to make an accurate estimate of the proba-bility of capsizing in the time interval τ for a fixed set of values of A_r and τ. Therefore in this case a stepfunction will be introduced by the threshold value τ_o. In contravention of the methodology applied in [16] and the corresponding test calculations presented in [18] τ_o is defined in such a way that this time limit is a function of parameters describing the windload and the energy balance during the time lapse τ. In an attempt to account for the dynamics of the ship rolling behaviour (in a former phase) the following condition for τ_o has been derived:

$$\tau_o = 0.6\, i_\varphi / \sqrt{|GM_c|} \qquad \text{(sec)}$$

where i_φ = roll radius of gyration (m)
and GM_c = initial metacentric height of the ship poised (amidships) on the wavecrest (m)

This relation is based on the assumption that the "restoring" moment may be averaged over the time interval τ and capsizing occurs when $\tau_o > T_Q/4$ where T_Q is the natural rolling period during the time interval τ. It is quite evident that such a relation for τ_o based on this approach is only realistic for $GM_c < 0$. A sensible approach, which is more appropriate for this particular situation, may be formulated in general as a function of parameters describing A_r and B_o. The expression for τ_o should anyhow reflect acceptable realistic situations. In the situation considered however it is acceptable to ignore the dynamics of the ship on the wave rest under the influence of a beam windload.
In the Appendix a formula for τ_o has been derived using inter alia the data from [19]. The formula reads:

$$\tau_o = \frac{9.3 \cdot 10^{-6}}{u} \exp\left(0.45 \cdot 10^{6} \frac{\delta_h}{l_w u^2}\right)$$

for $\delta_h > 0$

with u = windspeed (at 10 m above sea-level)
and δ_h = stochastic contribution of wind heeling lever = $(h_\varphi - h_W)_{max}$

In order to obtain insight about the impact of this threshold-limit reference is made to the Table produced in Figure 7.
Since all these expressions for τ_o yield variable values for each regular wave the calculation procedure is restricted for the time being to:

$$\tau_o = 0 \quad \text{and} \quad \tau_o = 0.6\, i_\varphi \sqrt{|GM|}$$

and the wave-dependent values of τ_o are calculated only for the purpose of further analysis.

ad.5: Using a slightly different notation the number of capsizings per time unit (n) may, in accordance with [16], be written as follows:

$$n(A_r<0;\ \tau_o,\ M_e) = \frac{1}{2\pi}\sqrt{\frac{m_2}{m_o}} \cdot P\ (\tau > \tau_o,\ M_e)$$

where the probability P should be estimated in accordance with the following numerical integration procedure:

$$P(\tau > \tau_o;\ M_e) = \int_0^\infty d\omega \int_0^\infty d\xi_a \cdot p(\xi_a, \omega)$$

The joint probability density function p reads:

$$p(\xi_a, \omega_i) = \frac{\xi_a^2}{\sqrt{2\pi m_o M}} \exp[-\frac{\xi_a^2}{2M} . (m_2' - 2m_1' \omega_i + m_o \omega_i^2)]$$

with $M = m_o m_2 - m_1^2$, $M' = m_o m_2' - m_1'^2$

and $m_n' = \int_0^\infty \omega^n S(\omega) d\omega$

$$m_n = \int_0^\infty (\omega - \omega_m)^n \, . \, S(\omega) d\omega$$

where $S(\omega)$ = wave spectral density
ω = wave circle frequency
ω_m = wave spectral frequency where maximum of $S(\omega)$ occurs
ξ_a = wave amplitude

In order to achieve a very general validity in the field of application the Pierson-Moskowitz wave spectrum definition for the North Atlantic Ocean has been chosen for the process of systematic calculations (see also Figure 8). The formula for $S(\omega)$ reads:

$$S(\omega) = 0.0081 \, g^2 \omega^{-5} \exp[-0.74 g^4 (u\omega)^{-4}]$$

with g = acceleration of gravity = 9.81 $msec^{-2}$
and u = wind speed (at 19.5 m above sea-level) $\approx 1.065 \, u_{10}$

From the literature is known that the long-term probability density distribution of the wind speed is fitted, as a good approximation, by the Rayleigh distribution:

$$f(u_{10}) = \frac{u_{10}}{\hat{u}_{10}} \exp\left(-\frac{1}{2}\frac{u_{10}^2}{\hat{u}_{10}^2}\right)$$

with $\hat{u}_{10}$ = modal value of u_{10}
= 7.7 $msec^{-1}$ (N.A. Ocean)

5. SYSTEMATIC CALCULATIONS

In Figure 9 a Table has been included showing a summary of a tentative scheme of parameter variations. The values of the reference ship are indicated by an asterisk. As distinct from the other ship parameters two reference values have been selected for the ship length for the purpose of validation. In total about 50 parameter combinations have to be considered in the systematic computation procedure. A basic assumption inherent to this approach of a rather limited number of systematic calculations is that possible interaction effects of the parameters on the calculated safety level are not significant. The presentation of the results of the systematic calculations may be given in accordance with the principle outlined in Figure 9. Such a presentation lends itself for further statistical analysis and a possible formulation of a draft criterion. The results of the systematic calculations, including the associated analysis and a draft proposal for a stability criterion, will be presented in due course in a separate paper.

6. CONCLUDING REMARKS

The approach used in the computation methodology is based on some simplifications and assumptions reflecting the state of the art in order to have control of the entire procedure which is first of all pragmatic and comparative in nature.
Future work in this direction should address the ship motion problem in all six degrees of freedom and in particular the development of an advanced criterion for the threshold value τ_o taking into account the dynamics of the ships rolling behaviour under the influence of a stochastic wave and wind load.

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APPENDIX

From literature is known that windspeed fluctuations follow a Gaussian probability density distribution and the maxima follow a Rayleigh distribution:

$$f(\delta_v) = \frac{\delta_v}{\sigma_v^2} \exp\left(-\frac{\delta_v^2}{2\sigma_v^2}\right) \qquad \text{or}$$

$$F(\delta_v) = \int_{-\infty}^{+\infty} f(\delta_v)\, d\delta_v = \exp\left(-\frac{\delta_v^2}{2\sigma_v^2}\right)$$

From $\delta_p = C_D \cdot \frac{1}{2}\rho\delta_v^2$ and $\delta_h = \frac{l_w}{\gamma}\delta_p$ follows:

$$F(\delta_h) = \exp\left(-\frac{\gamma\delta_h}{C_D \rho l_w \sigma_v^2}\right)$$

with:

C_D = dragcoefficient $\approx$ 1.13 (-)
ρ = air density $\approx 0.125.10^{-3}$ tonsec^{2}m^{-4}
γ = specific mass of seawater $\approx$ 1.025 tonm^{-3}
δ_h = stochastic part of wind heeling lever (m)
l_w = $A_w z_w / \nabla$ (-)
σ_v^2 = variance of windspeed fluctuations
= area of windvelocity (gust) spectrum
= $6.667\ \bar{u}_*^2 = 6.667\,\tau/\rho = 6.667\ \kappa\ u^2 = 0.016\ u^2$ m^{2}sec^{-2}

with κ = surface drag coefficient (-) = $2.4.10^{-3}$
where u = average stationary windspeed at 10 m above sealevel (m/sec^{-1}) and u $>$ 15 msec^{-1} (Beaufort 7)

Substitution yields:

$$F(\delta_h) = \exp\left(-0.45.10^6 \frac{\delta_h}{l_w u^2}\right)$$

It should be noted that the correction for the turbulence intensity, decreasing for increasing windage area, is neglected.

Further it may be assumed that:

$$N(\delta_h) = \nu T F(\delta_h)$$

with $N(\delta_h)$ = number of crossings at level δ_h = assumed at L_c^{-1}
where L_c = lifecycle, assumed to be 20 years with 250 days at sea in a sector of wave directions of 45°, which corresponds with $0.54.10^8$ sec
ν = average crossing frequency (sec^{-1})
T = considered time lapse = τ_o (sec)

Using the formulation of the gustspectrum according to Harris:

$$\frac{f S(f)}{u_*^2} = 4 \frac{x}{(1+x^2)^{5/6}} \quad \text{with } x = \lambda \frac{f}{u}$$

and $\lambda \approx 1200$ m

it may be derived that:

$$\nu = \left(\int_0^\infty S(f) f^2 df \Big/ \int_0^\infty S(f) df \right) = 2.42 \frac{u}{\lambda} =$$

$1.98.10^{-3}u$ (sec^{-1})

Substitution of ν and $N(\delta_h)$ yields:

$\tau_o = o$ for $\delta_h \leqslant 0$

$\tau_o \text{ (sec)} = \dfrac{9.3.10^{-6}}{u} \exp\left(0.45.10^{6} \dfrac{\delta_h}{l_w u^2}\right)$
for $\delta_h > 0$

FIGURE 1

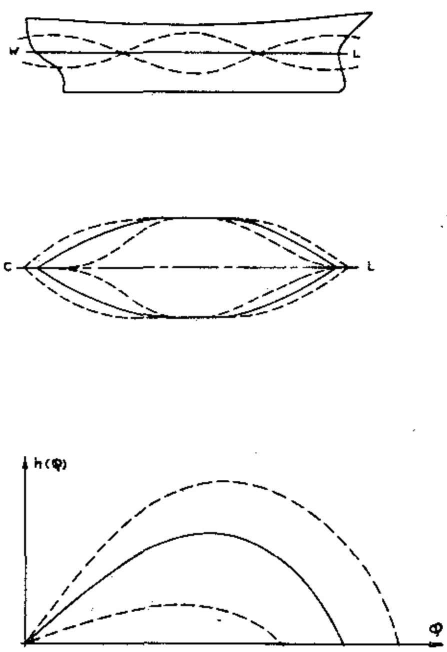


FIGURE 2

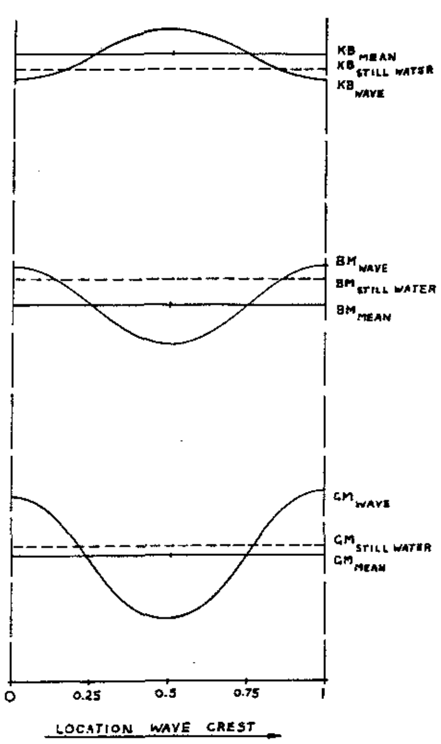


FIGURE 3

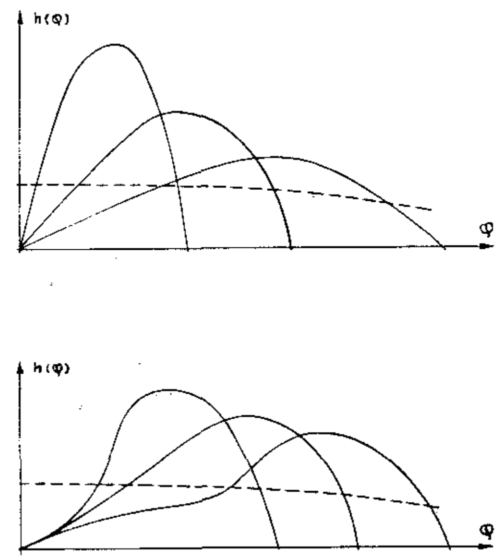


FIGURE 4

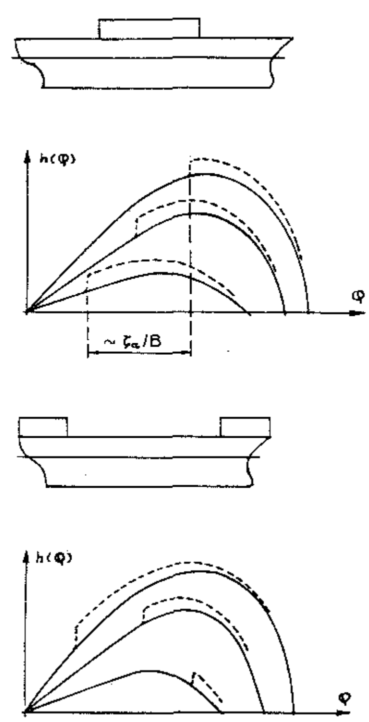
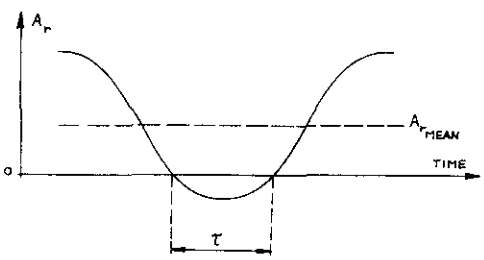


FIGURE 5

$L_{W.L.}$	~	154.6 M
B	~	22.82 M
D	~	12.37 M
d	~	9.14 M
∇	~	17910 M^3
F_n	~	0.33
l_w	~	0.27
C_B	~	0.56
C_W	~	0.70

FIGURE 6



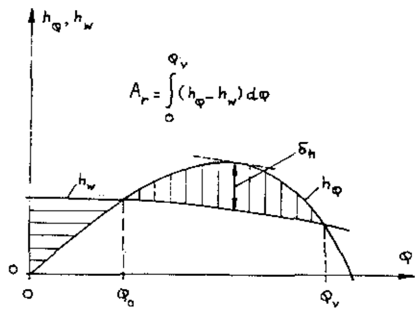
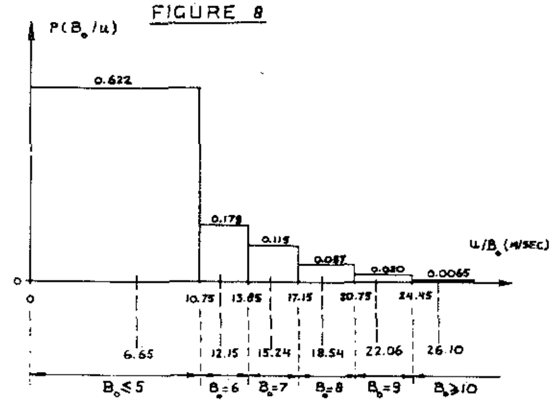


FIGURE 7

TABLE δ_h (M)

τ_o (SEC)	l_w	u=15 M/SEC	u=25 M/SEC	u=35 M/SEC
1	0.5	0.004	0.010	0.021
10	0.5	0.004	0.012	0.024
100	0.5	0.005	0.013	0.027
1	1.0	0.007	0.021	0.041
10	1.0	0.008	0.024	0.047
100	1.0	0.009	0.027	0.054
1	2.0	0.014	0.041	0.082
10	2.0	0.017	0.048	0.095
100	2.0	0.019	0.054	0.108

FIGURE 8



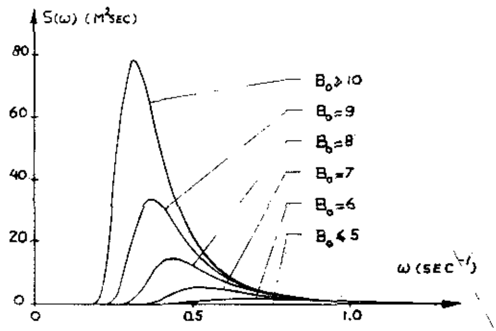
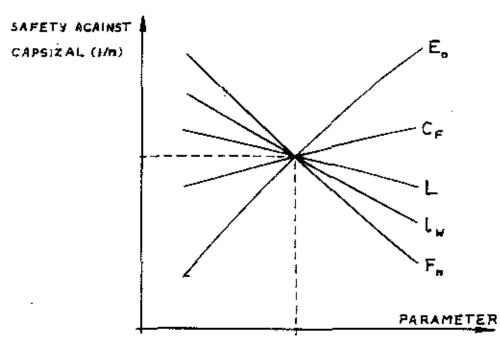


FIGURE 9

SCHEME OF SYSTEMATIC VARIATIONS

L (M)	30	45	65*	90	120	155*	225	325
l_w (-)	0.2	0.3	0.45*	0.6	0.75	0.9	1.2	
F_n (-)	0.2	0.25	0.3*	0.35	0.4	0.5		
C_F (-)	0.7	0.75	0.8*	0.85	0.9	0.95		
E_o (MRAD)	0.1	0.15	0.2*	0.25	0.3	0.4		

* REFERENCE SHIPS



A NEW PROGRAM INVESTIGATING SURVIVAL REQUIREMENTS FOR ALL ANTICIPATED CONDITIONS OF SMALL-SIZED TANK VESSELS

Masataka FUJINO[1], Iwao FUJII[2], Shigeaki HIGASHI[3], Michio NAKATO[4], Takashi OKUYAMA[5] and Isao SUZUKI[6]

We have developed a new damage stability computer program which can be used to decide whether or not a small-sized tank vessel carrying dangerous cargoes satisfies survival requirements for all anticipated conditions of loading and variations in draft and trim. The characteristic features of this computer program are as follows: 1) The maximum allowable height of the center of gravity that satisfies the survival requirements ; e.g. the Amendments to the 1974 SOLAS Convention, can be easily calculated for various cases of draft and trim as well as damaged conditions; 2) The damage stability for any designated loading conditions of a ship can be promptly calculated.

The accuracy of calculated results of the developed program has been confirmed by comparing the results with not only the results of inclination test of a simplified ship model but also the calculation results obtained by an existing widely-recognized computer program. As an example of verifying the effectiveness of the developed program, the maximum allowable height of the center of gravity of a 199 GT chemical tanker is shown for several damage cases.

INTRODUCTION

The 1983 Amendments to the International Convention for the Safety of Life at Sea, 1974 have provided that the damage survival capability should be investigated on the basis of loading information submitted to the Administration for all anticipated conditions of loading and variations in draft and trim of vessels carrying dangerous cargoes [1]. When the existing computer programs of damage survival capability, of which the effectiveness is widely-recognized in Japan, are used to assess the damage stability of tank vessels, the whole process necessary for calculating the damage stability should be carried out repeatedly for every damage case and loading condition. The amount of time and expenditure required for these calculations increases enormously with the increase of damage cases and loading conditions under which the stability calculation sholud be performed.

In view of the present situation, we have developed a new computer program for seeking the maximum allowable height of the center of gravity of small-sized tank vessels, which is required to satisfy the survival requirements of the convention. The characteristic features of the developed program are to calculate the cross curves of righting lever GZ for a limited number of variations in draft and trim for every anticipated damage case, and to

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calculate the maximum allowable KG, which is height of the center of gravity CG above base line, for every case of loading and damaged condition. Then, the damage stability as well as the allowable height of CG for a designated loading and damaged condition is computed by an interpolation.

This conception is more applicable to small-sized tank vessels, because the variations in draft and trim of such vessels are limited remarkably compared with large-sized ones. However, the accuracy of calculated results by the new program should be examined. By comparing the GZ curves, for instance, obtained by the new program with not only the inclination test results of a simplified ship model but also the results calculated by an existing program which has been confirmed by various experiments and widely used for last fifteen years in Japan, the developed program has been verified to be of enough practical use compared with the existing one in spite of adoption of a simplified calculation method.

The advantage of the developed program over the existing one is more manifested as the number of loading conditions for which the damage stability should be investigated increases, because in the developed program the preliminary calculation necessary to compute the allowable KG for all anticipated loading conditions is performed collectively at early steps.

DEVELOPED PROGRAM

Compositon of program

The developed program is composed of the following five steps:

1) Step-1; Calculation of Bonjean's curves of ship's hull and tanks,
2) Step-2; Calculation of cross curves of the righting lever GZ for designated variations in draft and trim, and damage cases,
3) Step-3; Calculation of the maximum allowable KG,
4) Step-4; Caluculation of the damage stability for designated loading conditions, and
5) Step-5; Check of the input data of ship's hull and tanks.

The flow chart of the developed program is shown in Fig. 1 with definition of symbols. As shown there, the preliminary calculation for all anticipated damaged conditions as well as various variations in draft and trim is performed collectively at Step-1 and Step-2. Then, the damage stability for each damaged condition is calculated at Step-4 only. This is the main characteristic features of the developed program in comparison with the existing program. Here it must be noted that the free surface effects of tank liquids on ship's trim is neglected in order to cut down the computation time, because those effects on the damage stability have been confirmed to be negligible by preliminary studies.

Outline of computing process

Step-1: 1) Bonjean's curves and the breadth of water plane are calculated from the input data of ship's hull, tanks, and erection on the deck for each heel angle; 2) For each heel angle, the volume of each tank and the tank heel moment are computed using the Bonjean data of tanks, and put out into the data files. Fig. 2 shows the flow chart of Step-1.

Step-2: 1) Data of designated draft and trim, locations of salt water inflow openings, and damaged tanks are inputted; 2) Various hydrostatic data such as displacement Δ , distance from midship to center of buoyancy ⊠B, distance from midship to center of floatation ⊠F, tons per cm immersion TPC, and moment to change trim one cm MTC before damage are calculated and put out into the data files for each case of draft and trim; 3) For each of loading condition, damage case and heel angle, the convergence calculation of draft and trim is carried out based on lost buoyancy method with trim free; 4) For each damage case, cross curves of GZ are obtained for designated variations in draft and trim on the assumption that KG=0 and ℄G=0, where ℄G denotes the distance between CG and the vertical center line of ship; 5) The angles at deck immersion and salt water inflow are computed.

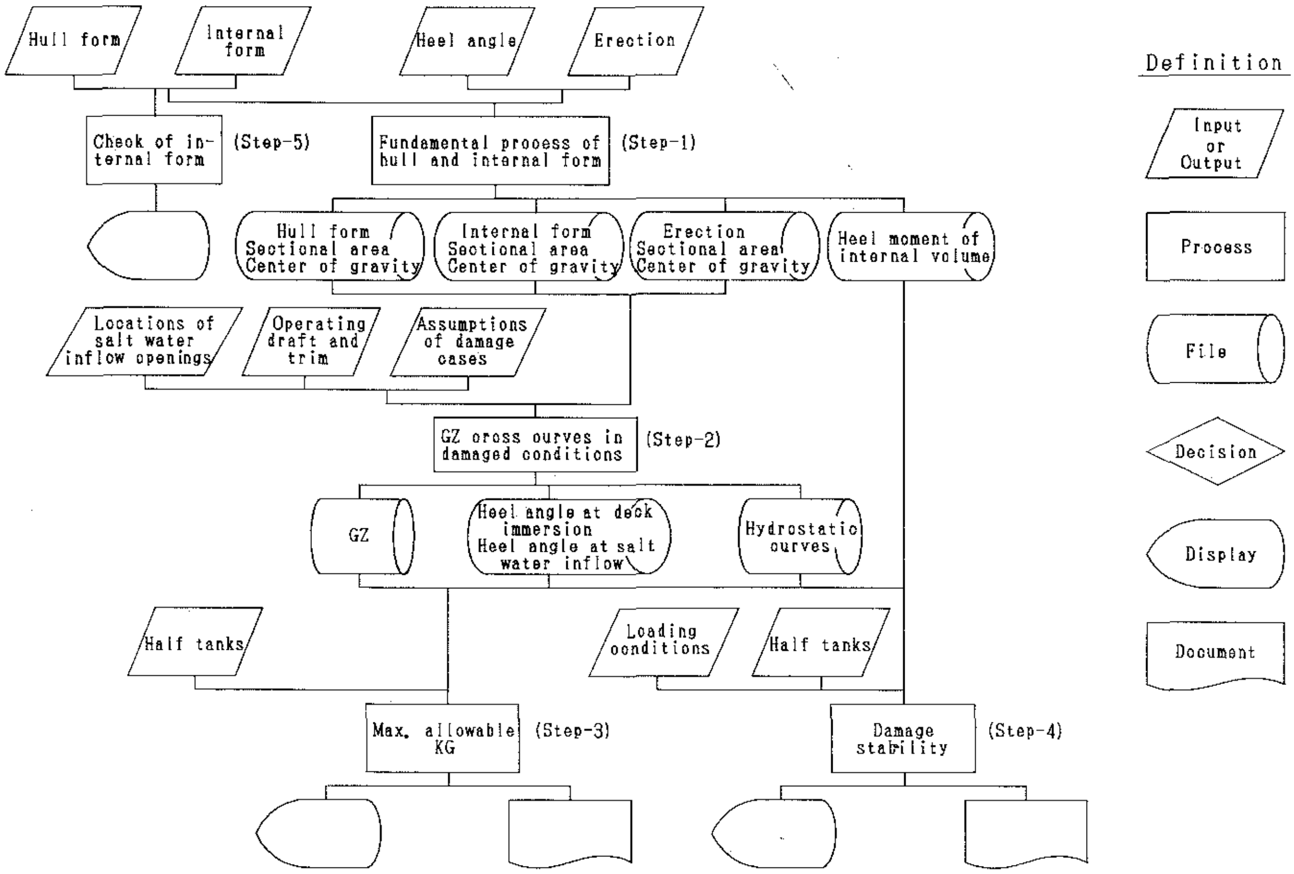

Fig. 1 Flow chart of developed program

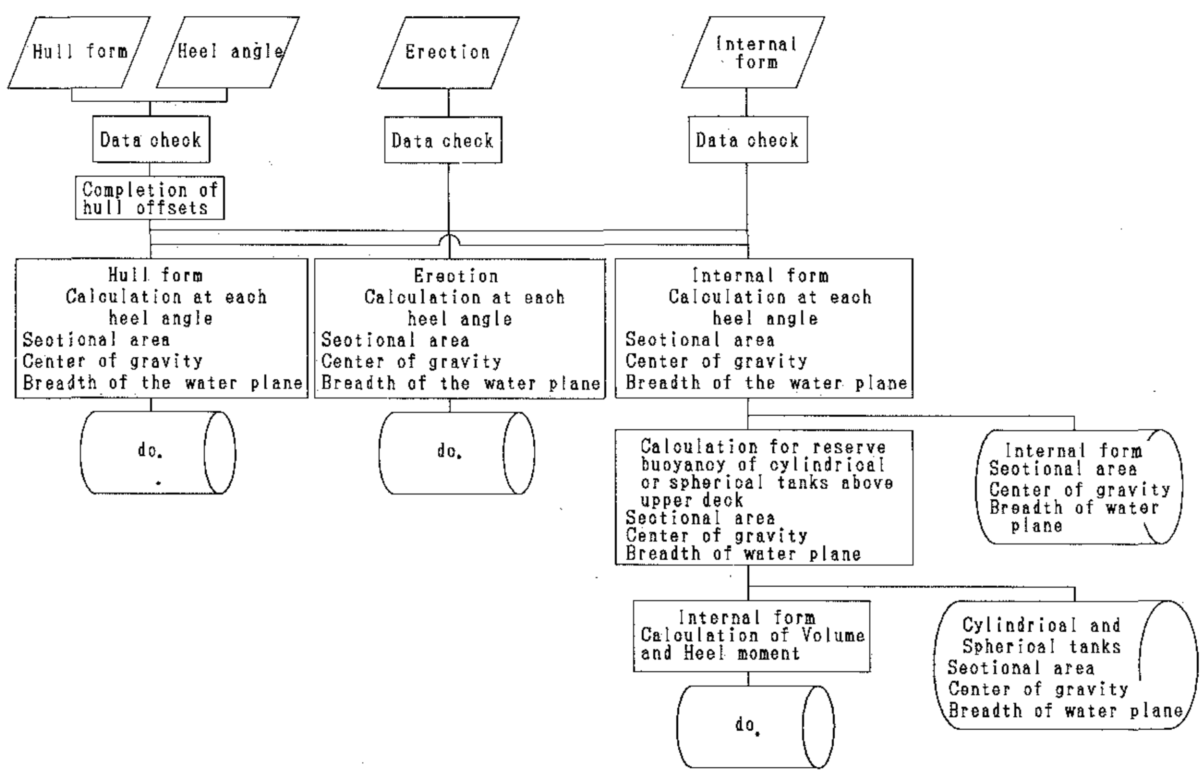

Fig. 2 Flow chart of Step-1, Bonjean's curves

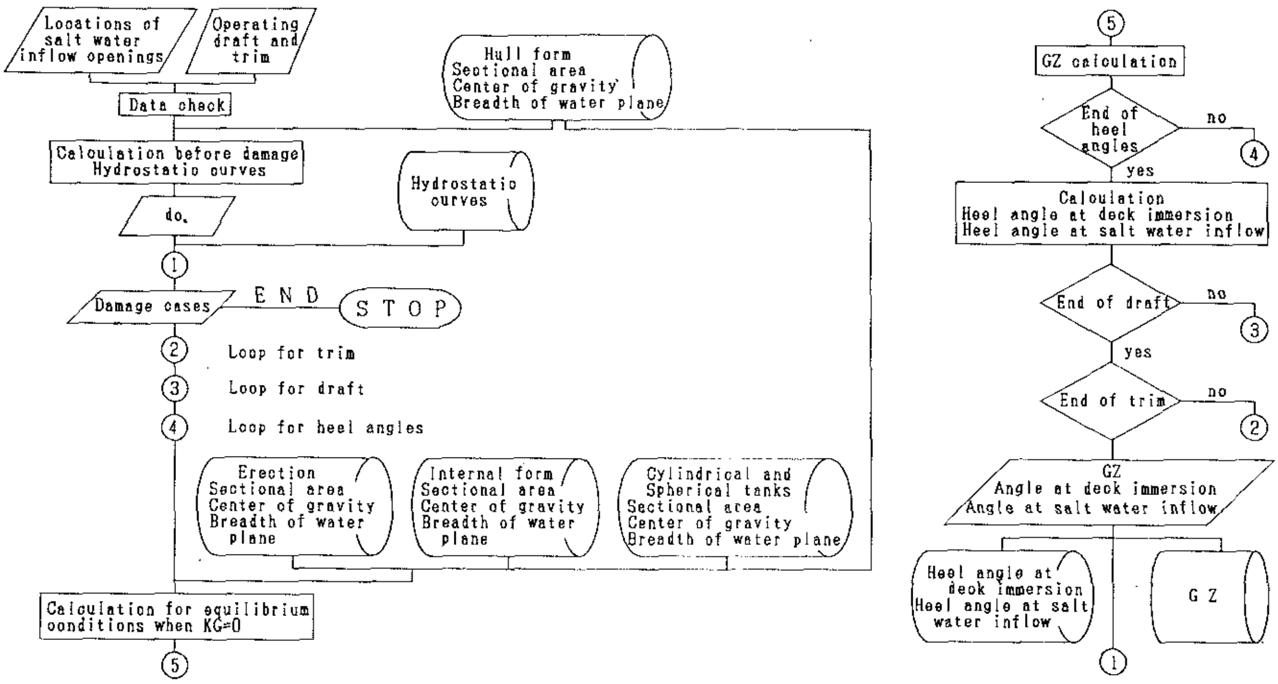

Fig. 3 Flow chart of Step-2, Cross curves of GZ

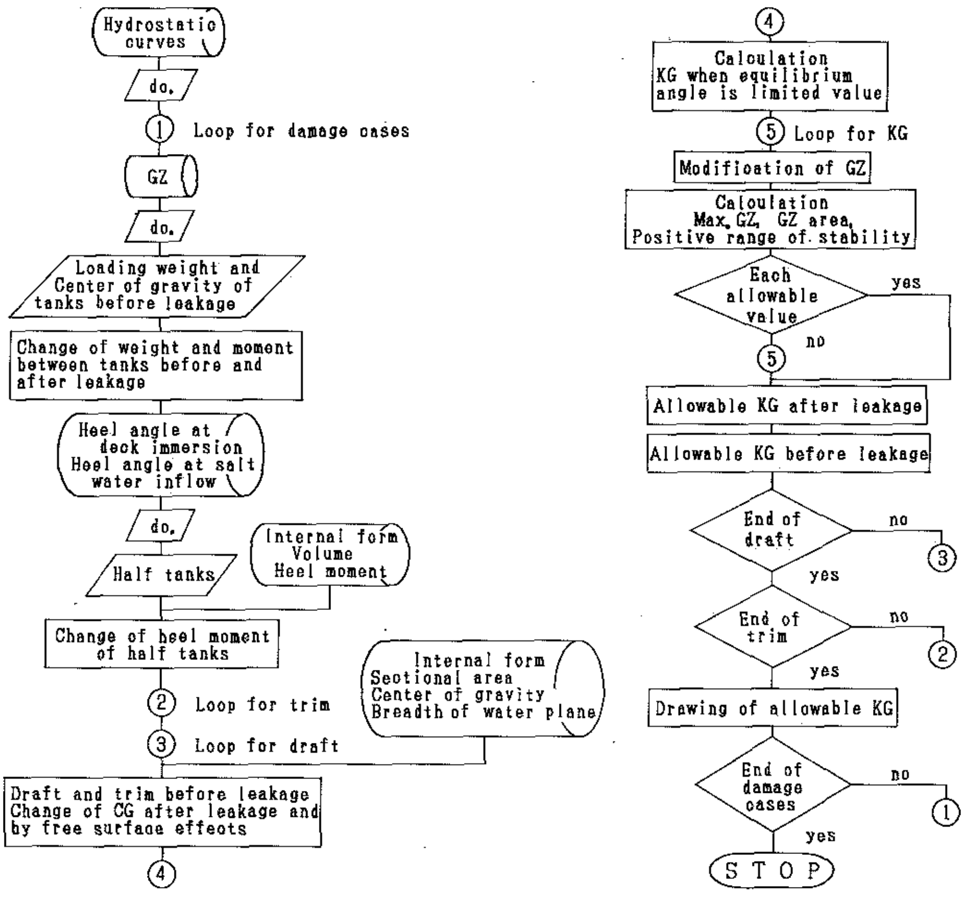

Fig. 4 Flow chart of Step-3, Max. allowable KG

At Step-2, unloaded tanks are assumed for the calculation. Fig. 3 shows the flow chart of Step-2.

Step-3: 1) Designation of damage cases is read, and data of damaged tanks and tanks having the free surface of liquids, which are defined as half tanks here, are inputted; 2) The leakage of liquids out of damaged tanks is calculated; 3) The change of heel moment of half tanks is calculated; 4) For each case of draft and trim for which the cross curves have been obtained at Step-2, the draft and trim before the leakage of tank liquids are calculated, and the change of KG and ȽG due to the leakage of tank liquids, and the free surface effects of liquids in tanks on KG are computed; 5) Inputting the value of GZ for an assumed case of KG=0, ȽG=0, and heel moment=0, the value of ȽG after the leakage of tank liquids is computed taking account of the free surface effects; 6) The maximum allowble KG is obtained based on the survival requirements of the 1974 SOLAS Convention [1], and then the value of KG is modified by taking account of the change of KG due to the leakage of tank liquids; 7) The values of the maximum allowable KG for the intact draft and trim conditions are obtained by the interpolation, using the computation results obtained at the above steps 4) to 6).

At Step-3 and the following Step-4, it is assumed that the liquids in damaged tanks are completely lost from those and replaced by salt water up to the level of the final plane of equilibrium [1]. Fig. 4 shows the flow chart of Step-3.

Step-4: 1) Data relating to loading conditions and designation of damaged tanks as well as half tanks are inputted; 2) The change of heel moment of half tanks and the leakage of liquids out of damaged tanks are calculated for designated loading conditions and damage cases; 3) The draft, trim, KG and ȽG after the leakage of tank liquids are calculated; 4) The damage stability for designated loading conditions is obtained taking account of the effects of liquids leakage as well as the free surface of tanks. Fig. 5 shows the flow chart of Step-4.

Step-5: Map data of tanks are displayed and put out in order to check the input data of ship's hull and tanks.

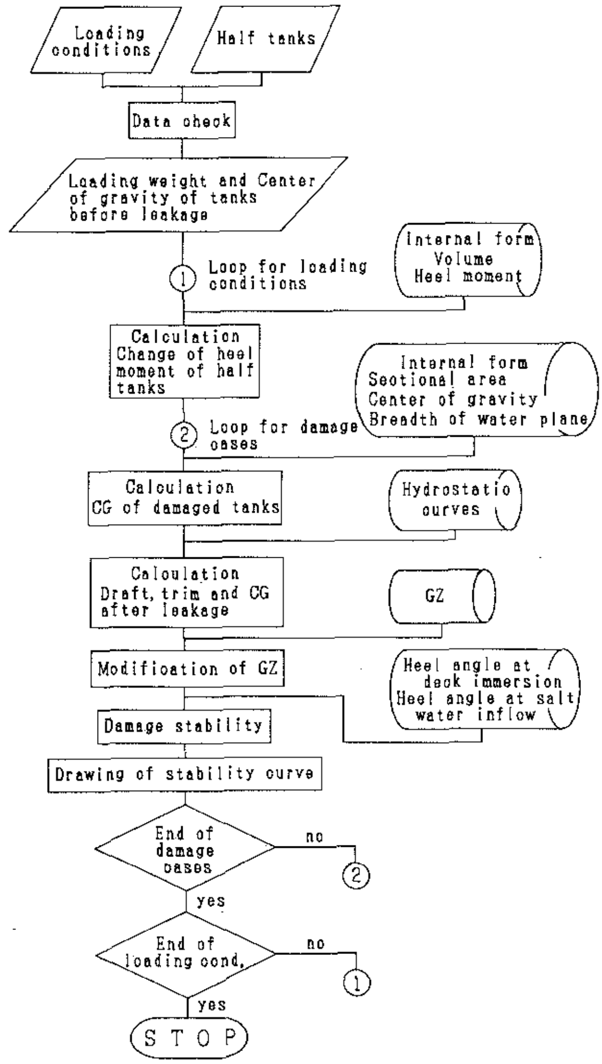

Fig.5 Flow chart of Step-4, Damage stability

EXAMPLES OF CALCULATIONS

In order to verify the effectiveness of the developed program, the righting lever GZ computed by this program has been compared with the results of inclination test of a simplified ship model of wall-side, and furthermore compared with the computation results by the existing program for various small-sized vessels. As the result, the developed program has been proven of practical use although the computed results by the new program may differ, as the case may be, slightly from the results obtained by the existing program in a case where excessive trim is caused by flooding.

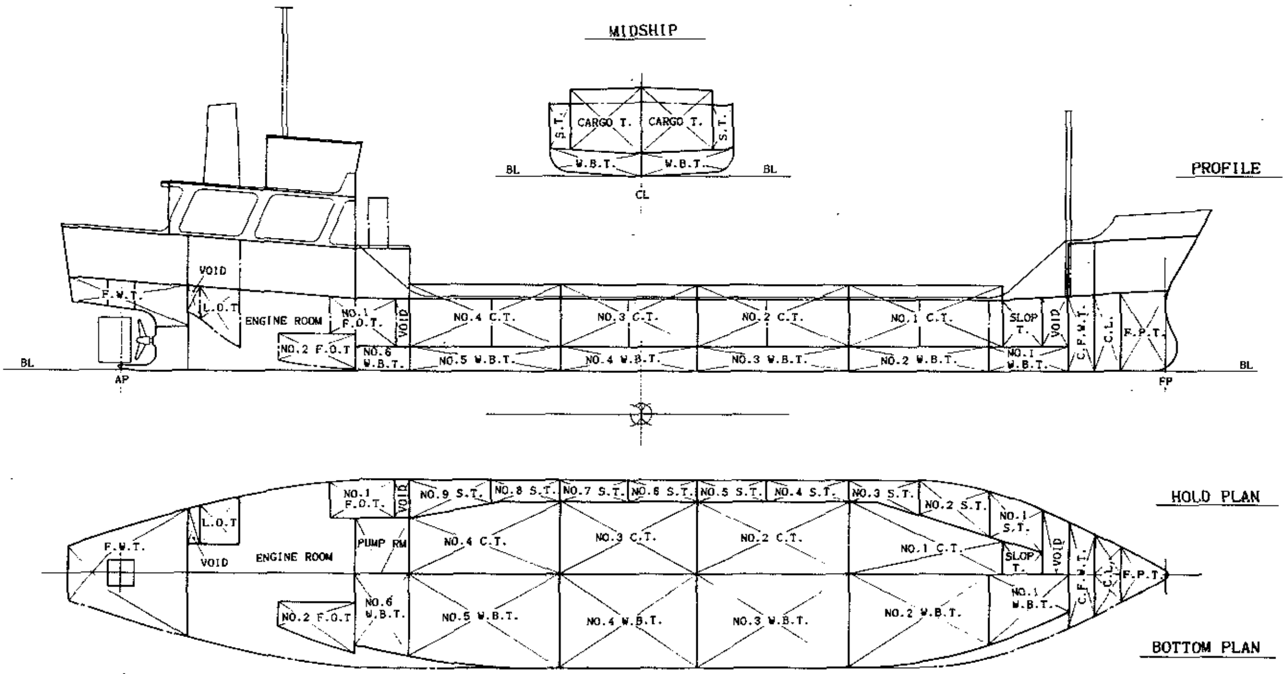

Fig. 6 General arrangement of 199 GT chemical tanker

TABLE 1
Principal particulars of
199 GT chemical tanker

Lpp (m)	B (m)	D (m)	d (m)	Δ (t)	GT (t)
44.00	7.80	2.95	2.45	323	199

Sample ship

The damage stability has been calculated for various small-sized vessels to confirm the usefulness of the developed program as mentioned previously. Since we do not have enough space to describe fully the computation results, some typical examples of damage calculation of a sample ship will be mentioned in the following. The sample ship is a 199 GT chemical tanker for coastal service, of which the principal particulars and the general arrangement are shown in Table 1 and Fig. 6 respectively.

Examples of damage calculation

The damage calculation, of which the results will be referred in this paper, has been carried out for nine damage cases of two-compartment flooding. Out of nine cases, three cases of the damage of tanks in the vicinity of the midship are shown in Table 2 and Figs. 7 to 9. Table 2 shows the detailed information of flooded tanks, and the draft, trim, and angle of equilibrium after flooding computed by the developed and existing programs. And the curves of righting lever GZ versus heel angle ϕ after flooding are compared in Figs. 7 to 9. As shown there, the computation results by the developed

TABLE 2
Examples of damage cases and of calculation results for two-compartment flooding of full load condition

Damage case number	Damage compartment	Program	Draft (m)	Trim (m)	Angle of equilibrium (deg.)
4	No.5 S.T. (S) No.3 W.B.T. (S) No.2 C.T. (S) No.6 S.T. (S) No.4 W.B.T. (S) No.3 C.T. (S) No.7 S.T. (S)	New Existing	2.90 3.01	−0.66 −1.02	30.60 32.44
5	No.7 S.T. (P) No.4 W.B.T. (P) No.3 C.T. (P) No.8 S.T. (P) No.5 W.B.T. (P) No.4 C.T. (P) No.9 S.T. (P)	New Existing	2.68 2.76	0.87 0.84	14.62 20.35
6	No.9 S.T. (S) No.5 W.B.T. (S) No.4 C.T. (S) VOID 21-22 (S) No.6 W.B.T. (S) No.1 F.O.T. (S) PUMP ROOM (C) No.8 S.T. (S)	New Existing	2.87 2.86	1.18 1.23	20.90 20.36

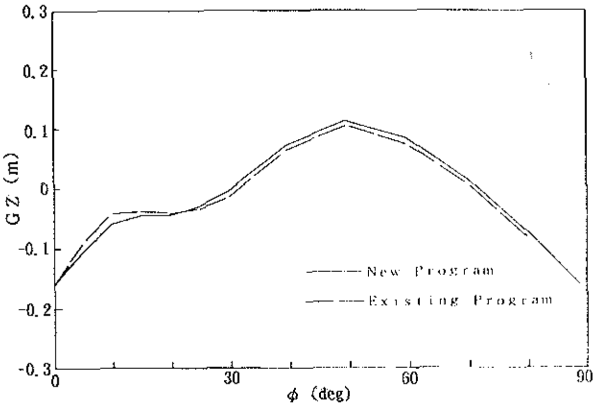

Fig. 7 Values of GZ vs. ϕ for damage case 4 of full load condition

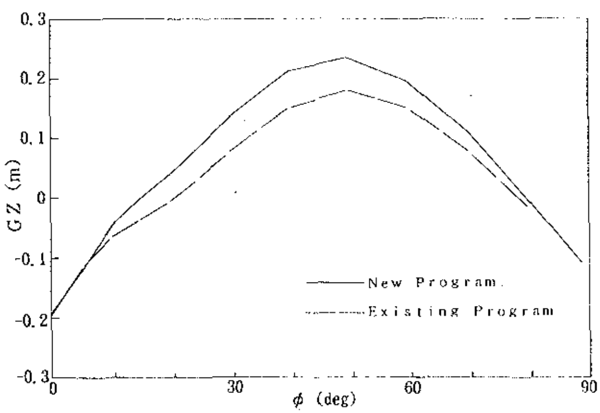

Fig. 8 Values of GZ vs. ϕ for damage case 5 of full load condition

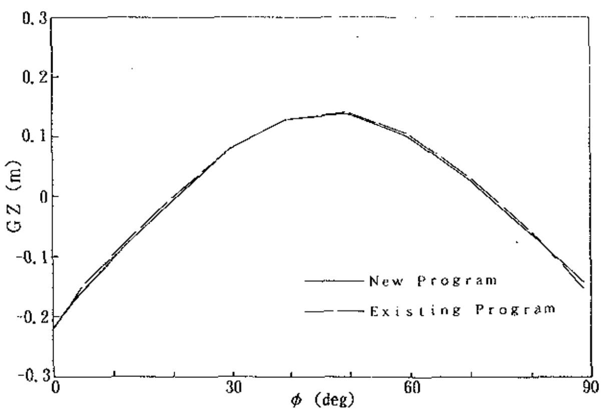

Fig. 9 Values of GZ vs. ϕ for damage case 6 of full load condition

program and the existing one are in fairly good agreement in spite of the difference between the two calculation methods.

The maximum allowable KG curves, which are obtained by the developed program

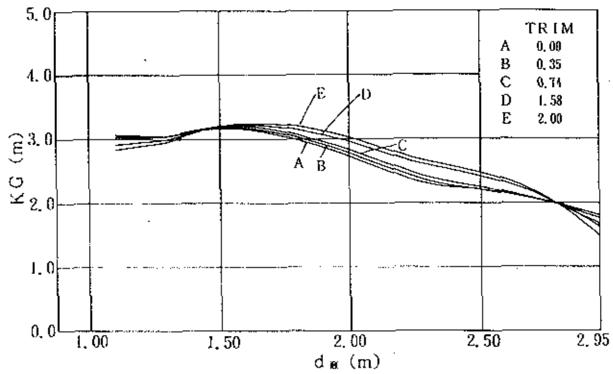

Fig.10 Max. allowable KG curves vs. d_W for damage case 4

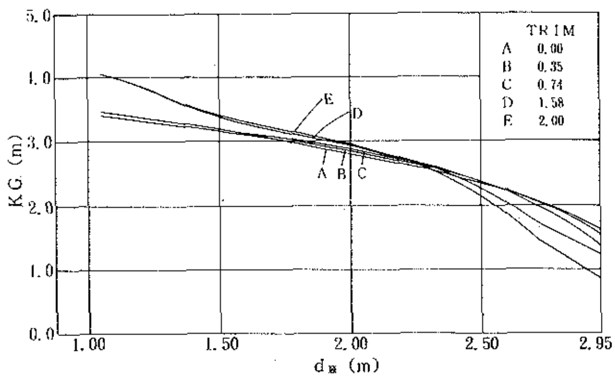

Fig.11 Max. allowable KG curves vs. d_W for damage case 5

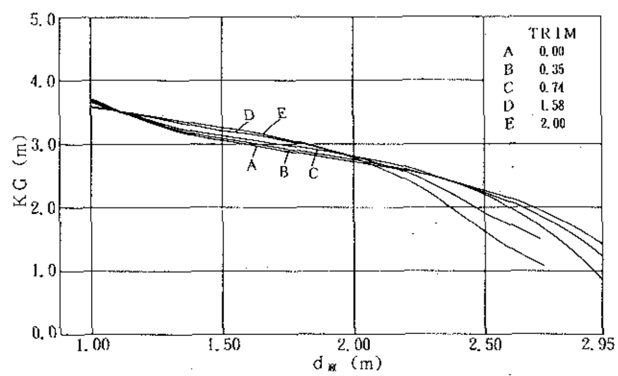

Fig.12 Max. allowable KG curves vs. d_W for damage case 6

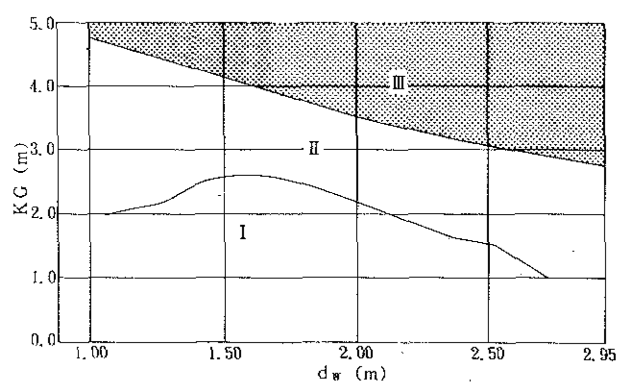

Fig.13 Summarized max. allowable KG curves vs. d_W for nine damage cases

to satisfy the survival requirements of the 1983 Amendments to the 1974 SOLAS Convention for the above three cases, are shown in Figs. 10 to 12. The abscissa and ordinate of these figures denote the midship draft d_M and KG before flooding respectively, in other words, the draft and KG for the intact condition, and five curves in each figure correspond to five assumed cases of trim before flooding.

In the damage cases shown in those figures, there is a slight effect of the trim on the maximum allowable KG. In Fig.13 summarized are the maximum allowable KG curves for nine damage cases. The region I marked in this figure is a safety zone which means that the vessel is able to survive in any damage case of two-compartment flooding. On the contrary, the region III is a risk zone which means that the vessel can never satisfy the survival requirements for any damage case of two-compartment flooding. In the region II, the vessel may satisfy or may not satisfy the requirements depending on the damage case. If we want to obtain such results as Fig. 13 using the existing program, it is necessary to perform at least twenty-five calculations for the product of five cases of draft and five cases of trim. It takes approximately eight-fold time over the developed program.

CONCLUSIONS

The following conclusions are obtained through the development of a new computer program investigating survival requirements for all anticipated conditions of small-sized tank vessels carrying dangerous cargoes.

1. The maximum allowable KG can be easily calculated for various loading conditions and damage cases. Simultaneously, the damage stability for any designated loading condition of small-sized tank vessels is able to be computed promptly.
2. In comparison with the existing program, the developed program enables one to save the computing time remarkably. This advantage is more manifested as the number of cases of damage calculation increases.
3. The accuracy of computation by the developed program has been confirmed by comparing the computed results with results by a well-recognized existing program. That is to say, the accuracy of the developed program is satisfactory in most cases except when excessive trim is caused by flooding.
4. When the value of KG for a loading condition is close to the maximum allowable KG of the loading condition obtained by the developed program, it is recommended to examine minutely the safety margin by more detailed computaion.

Further calculations by the developed program will be performed for various ships of different size and different type in order to verify the applicability of the developed program.

ACKNOWLEDGMENT

This development was sponsored by a grant from The Japan Shipbuilding Industry Foundation (Ryoichi Sasakawa, Chairman) as a project of Tsukuba Institute, Japan Foundation for Shipbuilding Advancement under the leadership of the Ministry of Transport of Japan.

The programming of the new computer program was carried out by Suuri-Keikaku Co., Ltd. We would like to thank the persons concerned.

REFERENCE

1. International Code for the Construction and Equipment of Ships Carrying Dangerous Chemicals in Bulk (IBC Code), Chapter 2-Ship Survival Capability and Location of Cargo Tanks, Resolution MSC. 4 (48), IMO, adopted on 17, June 1983.

PREDICTION OF CRITICAL WAVE CONDITIONS FOR EXTREME VESSEL RESPONSE IN RANDOM SEAS

Kirsi K. Tikka[1] and J. Randolph Paulling[2]

ABSTRACT

The paper describes a method for predicting the occurrence of conditions which can excite extreme ship motions. Earlier experimental studies have suggested that large roll motions and capsizing are often associated with the ship encountering a number of steep and almost regular waves. Most capsizes have been observed in following and quartering seas. In this work, the wave conditions encountered by a ship are characterized in terms of wave groups that consist of a high run and a low run of waves. In order to study the waves encountered by a ship, the spectral information measured at a fixed point was transformed to a moving coordinate system. The wave groups and the high runs were then predicted in the moving coordinate system using envelope theory. These and other properties of the encountered waves were found to depend strongly on the ship speed. Using these results, the speeds and the headings at which the ship is likely to encounter long high runs were predicted. Critical conditions were determined by comparing the period and the length of the dominant encountered waves with the characteristics of the ship. The probability of the ship encountering a critical high run was then determined for a given speed and seastate.

INTRODUCTION

The theory for predicting linear ship motions both in regular waves and in random seas is well developed, and in most practical applications the predictions obtained from linear analysis are sufficient. However, in the case of resonant roll, large angles are frequently found to occur and, for such angles, the effects of nonlinearities appear to be significant. Large roll motions can also result from parametric excitation which is associated with time dependent coefficients in the equation of motion.

Unfortunately, a complete solution of the hydrodynamic problem, which would be valid for large excitations and large motions, is not yet available. Therefore, experimental and numerical methods are usually applied to study extreme roll motions. The disadvantage of these methods is that no techniques are available which would allow generalization of the results obtained from a single wave record to any possible wave conditions encountered by the ship. To obtain statistical information on the motion behavior of a ship in realistic conditions, a large number of long simulations or experimental runs would be required.

In the work described here, the objective was to find a simplified method to predict the occurrence of wave conditions that can lead to undesirable roll motions or even to capsizing of a vessel. Based on the results of earlier experimental and theoretical work, low cycle resonance caused by parametric excitation in following and quartering sea conditions is often found to be critical for ships.

For purpose of the present analysis, the waves are assumed to be long-crested and the wave frequencies are assumed to lie in a narrow band. The method developed can then be used to predict speeds and headings at which a ship moving in

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[2] Professor of Naval Architecture, Department of Naval Architecture and Offshore Engineering, University of California, Berkeley

following and quartering seas is likely to encounter high runs of waves that, in turn, can excite low cycle resonance. A high run is defined as a sequence of waves with amplitudes higher than a given threshold value and a low run is defined as the following sequence of waves until the next exceedence of the threshold.

RANDOM WAVES ENCOUNTERED BY A SHIP

To study waves encountered by a ship, the wave spectrum observed at a fixed point was transformed to a moving coordinate system. The properties of the encountered wave process were found to depend strongly on the velocity of the ship. At velocities in the vicinity of the group velocity of the dominant frequency component, the spectrum becomes very narrow and peaked. The dominant frequency, ω_d, is defined as the mean zero upcrossing frequency of the original process and the corresponding group velocity is referred to as the dominant group speed. The record observed at these velocities appears nearly regular for extended periods of time. The same regularity of the encountered waves at certain velocities has been observed earlier in experiments [1].

Figure 1 illustrates wave records observed at various nondimensional velocities K=V/Vcr. V is the dimensional velocity. V_{cr} is called the critical velocity because at that velocity, the peak component of the spectrum at a fixed point becomes infinite in the encounter spectrum. The critical speed is equal to half of the phase speed of the peak component of the spectrum and it is generally referred to as the group velocity of the spectrum. Figure 2 shows a Pierson-Moskowitz spectrum $\bar{S}(\bar{\omega})$ and the encounter spectrum $\bar{S}(\bar{\omega}_e)$ observed at various nondimensional velocities. The spectra are in a nondimensional form $\bar{S}(\bar{\omega}) = S(\omega)\omega_{cr}/H_s^2$, where H_s is the significant wave height and ω is the frequency. At the frequency $\omega_{cr}=g/4V$, called the critical frequency, the encounter spectrum has an infinite value. The nondimensional frequency ($\bar{\omega}$) is equal to ω/ω_{cr}.

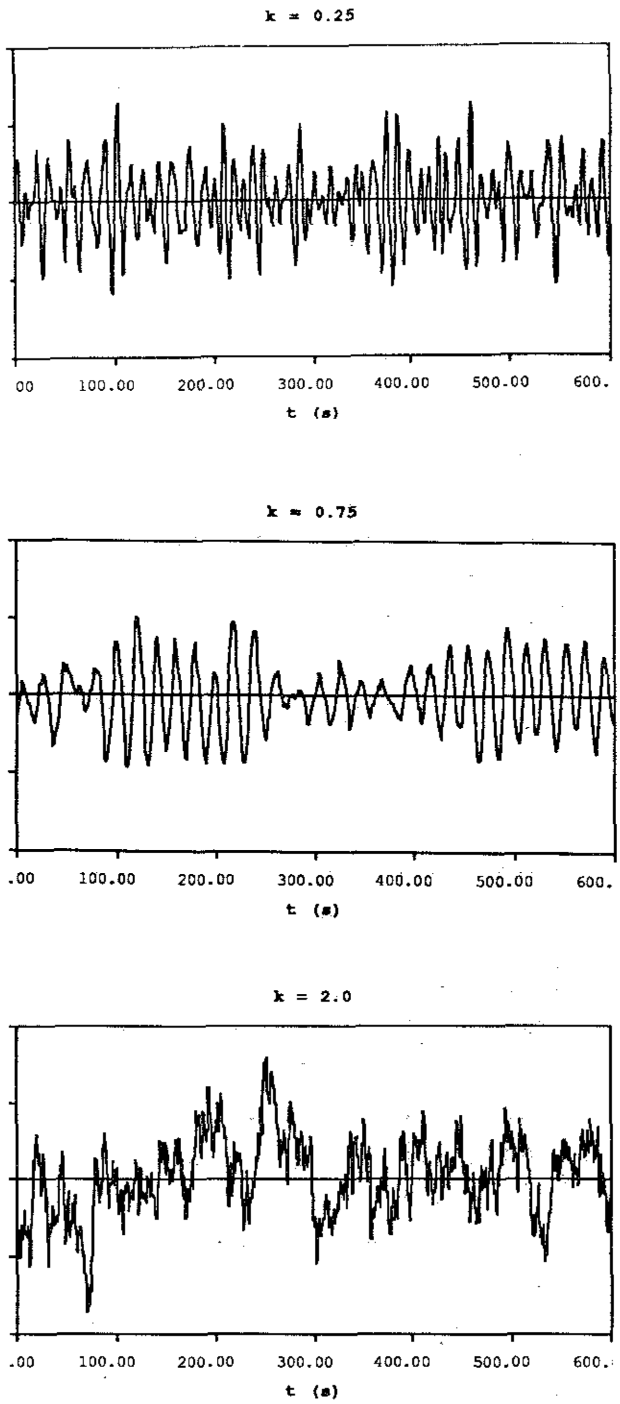

Figure 1. Ten minute samples of simulated time series at various nondimensional velocities.

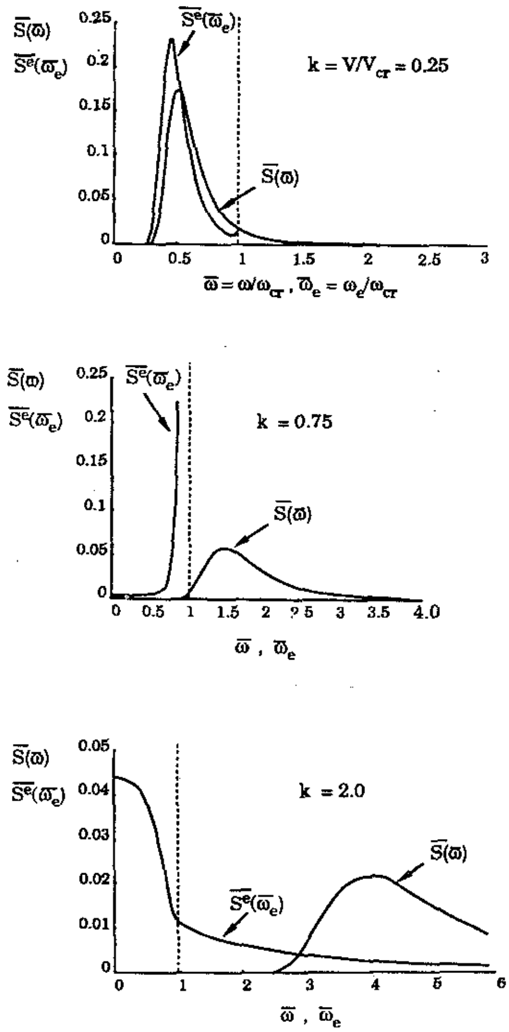

Figure 2. Mapping of the spectrum to a moving coordinate system at various nondimensional velocities.

The encounter spectrum was found to be narrow-banded up to a certain limiting velocity, which depends on the shape of the original spectrum. At these velocities the bandwidth of the encounter spectrum is smaller than the bandwidth of the original spectrum. Therefore, the encountered wave record can appear nearly regular even if the spectrum in the fixed coordinate system is relatively broad-banded. Above the critical velocity, the faster the ship travels, the more broad-banded the observed wave process becomes, thus the narrow-band assumption cannot be applied. Analysis shows, however, that most practical ship speeds are below this critical speed in which case the observed wave process in following seas will be relatively narrow-banded.

For standard wave spectra, such as the Pierson-Moskowitz spectrum, the second spectral moment becomes infinite in the moving coordinate system and in computing of properties which depend on the spectral moment of order two or higher, the frequency range of the encounter spectrum must be limited. The truncated spectrum is different depending on whether the frequencies are limited in the moving coordinate system or in the fixed coordinate system prior to the transformation of the spectrum to a moving coordinate system. The cutoff frequencies should be selected carefully, particularly if the truncation is done in the fixed coordinate system since some of the high and low frequencies may map into frequencies of interest in the moving system. At some velocities the energy neglected this way may result in a significant error in statistical estimates.

Low cycle resonance is associated with the ship encountering a sequence of steep almost regular waves in following or quartering seas. For this condition the critical conditions are defined in terms of the length of a wave group, the amplitude which is exceeded by the individual waves, and the wave frequency. The statistical prediction of wave groups and high runs was based on the theory developed by Longuet-Higgins [2]. In the prediction of wave groups and high runs, the truncation of the spectrum that gave the best agreement with simulation results was found to depend on the speed of the vessel. With the selected filtering, the simulation results and the theoretical predictions were in good agreement. The best agreement was found at velocities which correspond to the longest high runs and wave groups (see [3]).

OCCURENCE OF CRITICAL WAVE CONDITIONS IN FOLLOWING AND QUARTERING SEAS

The prediction of the occurrence of critical conditions is based on a comparison between the wave conditions encountered by a ship and the dynamic properties of the ship. The speeds and the headings, at which the vessel is likely to encounter long high runs, can be predicted in a given sea state. If the parameters of a critical wave group are known, the seastates and the speeds at which these conditions occur can be identified. Once the seastate and the speed of interest is known, the probability of a ship encountering, in time T_s, a high run which is longer than some critical run length, can be studied.

It is assumed that the behavior of a ship in regular waves is known from either model tests or from simulations. The wave parameters considered are the period, the amplitude and the number of waves required for critical motions to be excited. The initial conditions of a ship encountering the critical waves are neglected, although their importance is recognized.

An example case from simulation studies by De Kat [4] is adopted to illustrate the procedure for predicting potentially dangerous conditions in random seas once the behavior in regular waves is known. De Kat has studied the motion behavior of a vessel with the following characteristics

Ship length: L = 530 ft
Metacentric height: GM = 0.56 ft
Natural roll period: T_θ = 37 s.

The example ship has the full scale characteristics of the model of the American Challenger class cargo ship which was tested in the San Francisco Bay experiments, reported in [5]. The motions of the vessel were studied in regular waves with an amplitude of 9.8 ft and a frequency of 0.61 rad/s. The ratio between the wave length and the ship length was 0.975 and the speed of the vessel in the direction of the waves was 22.7 ft/s, which results in an encounter period of 18.1 seconds. The vessel capsized after experiencing resonant roll motions at approximately the natural roll period of the vessel. The number of waves encountered by the vessel before capsizing was 4.

SPEEDS AND HEADINGS ASSOCIATED WITH LONG HIGH RUNS OF WAVES

Both the theoretical considerations and the simulations predicted that a ship encounters the longest high runs in following seas at the dominant group speed, V_d, which can be computed using the following equations

$$V_d = \frac{g}{2\omega_d \cos\theta} \tag{1}$$

$$\omega_d = \sqrt{\frac{m_2}{m_0}} \tag{2}$$

Here θ is the angle between the ship heading and the direction of waves, m_0 is the zero spectral moment and m_2 is the second spectral moment.

The nondimensional dominant group velocity is defined as $K_d = V_d \cos\theta / V_{cr}$, where the critical velocity V_{cr} can be written in terms of the peak frequency of the spectrum, $V_{cr} = g/(2\omega_p)$. For the Pierson-Moskowitz spectrum, K_d was found to be equal to 0.71 for $\theta = 0$. The average high run lengths do not change very much for speeds around the dominant group velocity. Therefore the conditions which correspond to long high runs are characterized by a velocity range rather than by a single velocity.

The speeds which correspond to long encountered high runs can be dangerous to the vessel if the encounter frequency of the waves in the high runs is such that it can excite unstable motions. The motions depend also on the amplitude of the waves in the high run. In the case of extremely steep and high waves, a single wave can be sufficient to induce an instability.

In experiments and simulations, unstable motions have been observed at encounter frequencies which are approximately twice the roll frequency. The instability phenomenon, which causes large roll motions at low frequencies, results from parametric excitation. The capsizing simulated by De Kat is an example of instability due to parametric excitation.

To predict the random sea conditions in which the example vessel may experience unstable behavior, the periods of the dominant encountered waves, T_d^e, are plotted in Figure 3 as a function of the nondimensional velocity, K, for Pierson-Moskowitz spectra with significant wave heights equal to 10, 20 and 30 feet. The period of the encountered waves is equal to one-half the natural period of the roll motion of the example vessel at the nondimensional velocity 0.51 in the seastate having a 30-foot significant wave height. The corresponding nondimensional velocity is equal to 0.69 in the seastate of a 20-foot significant wave height. At both of these velocities a ship is likely to encounter long high runs. The dimensional speeds, $V\cos\theta$, corresponding to K=0.51 and H_s=30 ft and to K=0.69 and H_s=20 ft are 19.7 ft/s and 21.7 ft/s respectively.

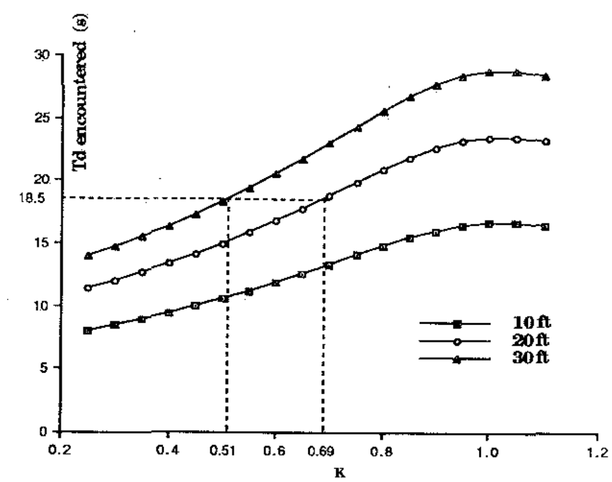

Figure 3. Periods of the dominant encountered waves as a function of the nondimensional speed $K=V/V_{cr}$ for Pierson-Moskowitz spectra with significant wave heights 10, 20, and 30 ft.

In addition to the frequency and the amplitude of the encountered waves, the length of the wave components should be considered. If the wave lengths are short or long relative to the ship length, the exciting force becomes small and the risk of large motions is reduced. To study this aspect in the seastate with a 20-foot significant wave height, the ratio between the wave lengths and the ship length are shown in Table 1 for the components which lie in a range of encounter periods around 18.5 s (half of a roll natural period). The tables show also the corresponding dimensional speeds. The wave lengths are computed using the dispersion relation for deep water waves.

$$\lambda = \frac{2\pi g}{\omega^2} \tag{3}$$

K=V/Vcr	T_d^e(s)	λ_1/L	λ_2/L	λ_3/L	Vcosθ(ft/s)
0.65	17.72	1.31	0.36	0.11	20.5
0.69	18.55	1.37	0.42	0.12	21.7
0.70	18.77	1.41	0.43	0.13	22.0
0.75	19.82	1.52	0.51	0.14	23.6

Table 1. The wave lengths of the frequency components which are observed at frequencies close to twice the natural roll frequency of the example vessel. The significant wave height is equal to 20 ft.

There are three wave lengths corresponding to one encounter period as a result of the mapping from a fixed coordinate system to a moving coordinate system (see [6]). The wave lengths λ_1 and λ_2 correspond to frequency components which overtake the ship, and the wave length λ_3 corresponds to the frequency component which is overtaken by the ship. At each speed there is at least one component which can have significant effect on the motions. These results would indicate that in a seastate with a significant wave height of 20 feet, a ship is likely to encounter long high runs of waves if it is travelling with a combination of a speed and a heading angle for which Vcosθ is in the neighborhood of 21 ft/s.

Another example case is considered to compare the theoretical predictions with simulation results in random seas. The ship is the same as the one in the previous example but the loading condition is different.

Metacentric height: GM = 0.86 ft
Natural roll period: T_θ = 30 s.
Heading angle: θ = 15°

The parameters of the seastate to be considered are the following

Peak frequency: ω_p = 0.53 rad/s
Significant wave height: H_s = 28 ft

The motions of the vessel were simulated by De Kat [4] at speeds 5, 10, 15, 20, 32, 33.68, 35 and 40 ft/s. Capsizing due to low cycle resonance was observed at the speeds 10 ft/s and 15 ft/s. Capsizing due to loss of static stability was observed at the speed 33.68 ft/s. This speed is the same as the speed of the model at the same load condition which resulted capsizing in a San Francisco Bay experiment.

To predict the conditions for low cycle resonance in the above wave conditions, a two-parameter Bretschneider spectrum was used. The nondimensional dominant group speed K_d for this spectrum is equal to 0.71.

The nondimensional velocities, periods of the encountered waves, and the wave length to ship length ratios corresponding to the speeds used in the motion simulations are shown in Table 2. The assumption that the observed wave record is narrow-banded is not valid at the speed of 40 ft/s, and therefore it is not included in the table.

K=V/Vcr	T_d^e(s)	λ_1/L	λ_2/L	λ_3/L	V (ft/s)
0.32	12.23	0.80	0.09	0.04	10
0.48	14.43	0.70	0.34	0.08	15
0.64	17.15	-	-	0.14	20
1.02	23.12	-	-	0.34	32
1.07	23.18	-	-	0.37	33.68

Table 2. The velocities used in the motion simulations and the corresponding wave data. The original spectrum is a two parameter Bretschneider spectrum with H_s = 28 ft and ω_p=0.53 rad/s.

The period of the encountered waves at the speed 10 ft/s (K = 0.32) is considerably smaller than half of the natural period of the roll, but the wave length of one of the frequency components is close to the ship length, which could be related to the capsizing due to low cycle resonance at that speed.

At 15 ft/s, the ship is likely to encounter long high runs, the period of the encountered waves is close to half of the natural roll period, and one of the wave lengths is close to the ship length. Therefore 15 ft/s is a critical speed for low cycle resonance which was observed in the earlier motion simulations.

Long high runs would be predicted at nondimensional velocities close to K=0.71, which would include the velocities 20, 32 and 33.68 ft/s. The wave length corresponding to 20 ft/s is small relative to the ship length, and therefore large motions excited by the high runs would not be expected at this speed. The length of the wave overtaken by the ship at velocities 32 and 33.68 ft/s is closer to the length of the ship, and the capsizing observed at 33.68 ft/s can be related to an encounter of a long high run. However, the loss of static stability is not as clearly related to the frequency of the encountered waves as the low cycle resonance. It must be also remembered that the present method cannot predict an occurrence of capsizing. It can only predict the combination of speed and heading at which the ship is likely to encounter long high runs, and the speed at which the frequency of the dominant encountered waves is equal to a critical frequency.

PROPABILITY OF ENCOUNTERING A CRITICAL HIGH RUN OF WAVES

The probability of a ship encountering a high run of waves, which is as long or longer than the critical high run, was predicted. The seastate and the speed of the ship were selected so that the period of the dominant encountered waves was close to the period of the waves in the critical high run. Again it was assumed that the wave process observed from the ship has a narrow band of frequencies, and that the high runs are statistically independent.

The number of waves in a high run H is assumed to have the following exponential distribution

$$p(H) = \frac{1}{H_a} \exp\{-\frac{H}{H_a}\} \tag{4}$$

where H_a is the average number of waves in a high run. If the ship encounters a high run, the probability that the run has a smaller number of waves than the critical number H_{cr} is determined using the following equation

$$\begin{aligned} p_{cr} = P\{H_j < H_{cr}\} &= \int_0^{H_{cr}-0.5} p(H)\, dH \\ &= 1 - \exp\{-\frac{H_{cr}-0.5}{H_a}\} \end{aligned} \tag{5}$$

Here both H_{cr} and H_j are assumed to have an integer value.

The number of wave groups observed from the ship in time T_s, N_g, can be obtained by the following equation

$$N_g = \frac{T_s N_0}{G_a} \tag{6}$$

where G_a is the average number of waves in a wave group and N_0 is the mean zero upcrossing period. The number of high runs encountered by the ship in a given time can be assumed to be equal to the number of wave groups if the period in question is much longer than the average length of a wave group.

The probability that at least one of the high runs is as long or longer than the critical high run is given by

$$P(\text{All high runs in } T_s < H_{cr}) = (p_{cr})^{N_g} \tag{7}$$

The probability that the example ship encounters a critical high run of waves in approximately 30 minutes was predicted. The probability of the encounter was computed for seastates described by Pierson-Moskowitz spectra with 20-foot and 30-foot significant wave heights. The theoretical predictions were compared with results estimated with simulations. The critical high run was determined based on motion simulations in regular waves and the parameters were the following

Amplitude: 9.8 ft
Period: 18.1 s
Number of waves: 4

The speed was selected so that the dominant encounter period was 18.1s. The encounter spectrum was narrow-banded.

The theoretical and the simulation results for the two seastates are shown below. The theoretical results are given for two ranges of cut-off frequencies which define a lower and an upper bound for the predicted values.

$H_s = 20$ ft
$V = 21.1$ ft/s
$K = 0.67$
$T_s = 1841$ s

Variables	Prediction		Simulation
Cutoff freq.:	A	B	
p_{cr}:	0.779	0.838	0.825
$1 - (p_{cr})^{Ng}$:	0.786	0.730	0.640

A: $0.5\omega_p - 1.5\omega_p$
B: $0.25\omega_p - 1.75\omega_p$

$H_s = 30$ ft
$V = 18.9$ ft/s
$K = 0.49$
$T_s = 1829$ s

Variables	Prediction		Simulation
Cutoff freq.:	C	D	
p_{cr}:	0.733	0.792	0.793
$1 - (p_{cr})^{Ng}$:	0.993	0.988	0.988

C: $0.75\omega_p$-$1.25\omega_p$
D: $0.375\omega_p - 1.625\omega_p$

The probability p_{cr} agree well with the simulation results for the choice of cutoff frequencies. The prediction of the probability of a ship encountering critical high runs in a given time is close to the simulated value. This prediction could be improved by studying a possible dependence between high runs.

CONCLUSION

The method described provides a relatively simple way to predict the occurrence of critical wave conditions for ships in following and quartering seas. The theoretical predictions on the statistics of encountered waves agree well with simulation results and a comparison with earlier motion simulations indicate that the predictions are realistic. A similar method could be used to predict critical conditions for other types of instabilities, such as subharmonic resonance.

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THE ROLE OF NUMERICAL SIMULATION IN THE STUDY OF EXTREME PLATFORM RESPONSE

J. R. Paulling, Y. S. Shin

The motion response of an intact or damaged floating vessel in extreme waves is a dynamic process involving several types of nonlinear phenomena. Some of these nonlinearities lead to responses that are essentially similar to those that would be predicted by a purely linear theory. Others result in effects that are not predicted by linear analysis. The means available for estimating the hydrodynamic loads and motion responses are discussed and comparisons are made of results of linear and nonlinear analysis.

INTRODUCTION

The extreme motion response of an intact or damaged floating vessel in high waves and strong winds is a dynamic process involving several different nonlinear phenomena. These include nonlinear terms in the equations of motion of the body and nonlinear dependence of certain forces on the wave, wind and body motions. If the motions are small or if the source of the disturbance is small, the forces corresponding to certain of the nonlinear terms may be ignored and the problem reduces to a linear one. Some important features of the vessel's motion response in the severe conditions that may lead to extreme motions and capsize are fundamentally nonlinear and, consequently, are absent from a linear prediction model.

The effects of motion nonlinearities may have two different characteristics. First, the nonlinearity may modify the magnitude without changing the general characteristic of a response that would be predicted by a purely linear analysis. A common example is viscous damping represented by a force proportional to the square of the velocity. Such quadratic damping is found in the roll damping of a ship undergoing moderately large motions or the heave damping of a twin hull semisubmersible. The resulting motion with quadratic damping appears very similar to the linearly damped motion in the same wave environment except that the amplitude is not proportional to the wave amplitude, i.e., a doubling of the wave amplitude does not double the roll amplitude. Resonance is found to occur at nearly the same frequency and the appearance of the motion is similar to that predicted by linear theory.

Some nonlinear effects introduce motion responses that are totally unlike those predictable by a linear theory. Of particular interest, are motion instabilities and sub- or superharmonic responses. As an example, a ship moving in head or following seas experiences a time-varying roll restoring moment coefficient as the wave crest moves along the ship length. This is a result of the wave and motion induced variation in the ship's underwater geometry. Typically, the stability is least when a wave crest is near amidships and greatest when a trough is near amidships. In pure head or following seas, there is no wave-induced exciting moment in roll and the equation of motion is of the form of Mathieu's equation. The solution of this equation becomes unstable for certain frequencies of the time varying term, and in the case of ship rolling, this instability is represented by a spontaneous roll that may grow to a value sufficient to endanger the vessel. Such "autoparametrically induced" response is found to occur in several other types of floating vessel motions as well.

Many of the effects that may endanger the vessel are transient in nature, resulting from one of a kind events such as episodic waves, extreme wind gusts, operational accidents and violent maneuvers. The analysis of the response to such events requires solving for the transient response of the floating vessel, given the time history of the triggering event. These dangerous motion effects also usually involve large amplitudes of waves, forces and the motions themselves. Consequently, the analysis of extreme motions and capsize must usually be based on nonlinear theories. In practical engineering applications, this normally requires solving the equations of motion by numerical integration in the time domain.

The state of the art is such that an exact general solution does not exist for the hydrodynamic forces on a body of arbitrary geometry undergoing motions of large amplitude in steep waves. In predicting the hydrodynamic forces for large motion analysis we must usually resort to a process of synthesis in which existing linear procedures are combined intuitively with more exact computations of the flow of real fluids over bodies of simple geometry. The validity of the results of such a synthesis must then be tested by means of model experiments. A procedure is described in [1] for the simulation of ship capsizing in waves in which linear strip theory is combined with a quadratic model of the viscous forces and exact computations of the hydrostatics of the underwater hull.

THE HYDRODYNAMIC FORCE SYSTEM

A commonly used linear procedure for computing the hydrodynamic forces on a three dimensional floating body is the distributed source Green function or "panel method". In this procedure one solves a boundary value problem in ideal fluid potential theory for the flow about the body undergoing small oscillatory motion in waves of small steepness. Implementation of the procedure requires that the underwater surface of the body be subdivided into a number of rectangular or triangular panels. Distributed pulsating sources are assumed to be located on each panel and the strengths of the sources are adjusted to satisfy the body kinematic boundary condition. There are two approximations in this procedure that limit its usefulness for solving large motions problems. They are, first, an assumption of small wave and body motions and, second, the assumption of an ideal inviscid fluid.

As a consequence of the first approximation, the underwater shape of the body is assumed to remain unchanged with time and the boundary conditions are satisfied on the mean underwater shape. The change in the immersed geometry with passage of waves combined with body motion is neglected. This is clearly a major impediment to the prediction of capsizing behavior since capsize motions are large by definition, with corresponding large changes in the immersed shape.

Current research in numerical hydrodynamics is being directed towards extending panel methods in the direction of solving the large amplitude hydrodynamics problem. These use panelized source distributions both on the body and on the water free surface in a time-domain mode of solution. In order to accommodate large motion amplitudes, both the panel geometry and the source strengths must be updated at each time step requiring considerable computational resources.

An approximate procedure that is suitable for approximating the forces on many types of fixed and floating offshore structures employs the so-called Morison formula [2], [3]. This formula, which is intuitive in origin, was intended for computing the wave forces on a stationary vertical pile but is now extended to the case of a moving slender cylinder at an arbitrary orientation in space. In applying this to the computation of forces on a space frame platform made up of slender cylinders as shown in Figure 1, two assumptions are necessary. It is first assumed that the cylinders are sparsely distributed so that flow interference between members may be neglected, and, second, each member is assumed relatively long and slender so that the flow is essentially two dimensional at each cross section. This procedure has been adapted to both linearized frequency-domain solution schemes and to the numerical time-domain type of solution of the equations of motion.

SOME COMPARISONS OF LINEAR AND NONLINEAR RESULTS

Figure 1 depicts a generic six column twin-hull semisubmersible that was used as the experimental example in the recent ABS MODU stability study, references [4] and [5]. The motion response characteristics were determined for a wide variety of wind and wave conditions by experiment and both linear and nonlinear computations. Extensive comparisons were made of the computational and experimental results in order to verify computational procedures used in later parametric studies.

The hydrodynamic characteristics of this platform were computed both by a panel procedure and by the Morison formula, and it is useful to compare the results in order to illustrate the features of each. The panel computation used a mesh defining 476 panels distributed over the surface of the body. The Morison formula computation involved treating each of the six columns and the two pontoons as an independent cylindrical body.

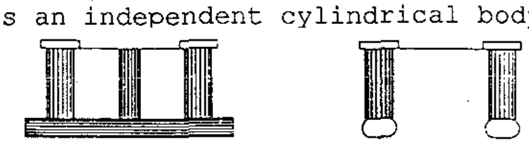

FIG.1 GENERIC SIX COLUMN SEMISUBMERSIBLE

The computed damping and added mass coefficients for the platform are illustrated in Figure 2. The added mass computed by the panel method is nearly constant while the radiation damping by this method (dashed line) shows a strong frequency dependence and nearly vanishes for periods greater than 15 seconds. In the Morison slender member model, an added mass coefficient of 1.0 is usually used for cylinder members and this is seen to give an overall structure added mass somewhat greater than the panel computation. The difference between the two can be explained by the hydrodynamic interference between the columns and pontoons which is neglected in the Morison model. Making an intuitive reduction to the C_M value to account for the shielding of the pontoon by columns brings the two methods into much closer agreement.

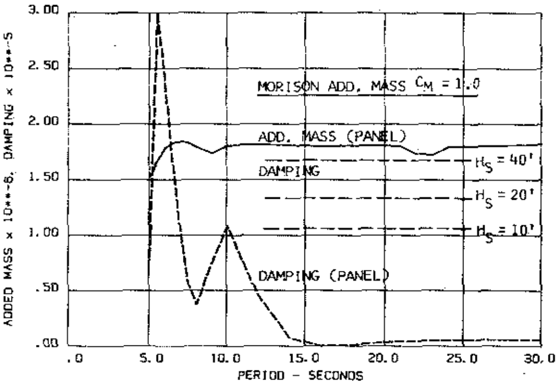

FIG. 2 GENERIC SEMI., HEAVE ADDED MASS AND DAMPING (Ft-Lb-Sec Units)

The damping displays a greater discrepancy between the two but we recall that the Morison model includes a viscous term that is missing from the ideal fluid panel model. The viscous damping is represented here by an equivalent linear damping coefficient, defined to have the same mean energy dissipation as the quadratic viscous term. The equivalent linear coefficient is found to be dependent upon the motion amplitude as shown in [6]. Three different values are shown corresponding to the platform motion in random seas of 10, 20 and 40 foot significant height. The values plotted here are based upon model scale C_D values although the graph is plotted for full scale wave period. The use of a value of C_D appropriate to full scale would reduce the equivalent linear values by about one-half to two-thirds. The significance of this damping discrepancy between the panel method and the Morison formula is most apparent in the heave resonant response.

Figures 3 and 4 contain the components of heave exciting force computed by the two methods. Here we

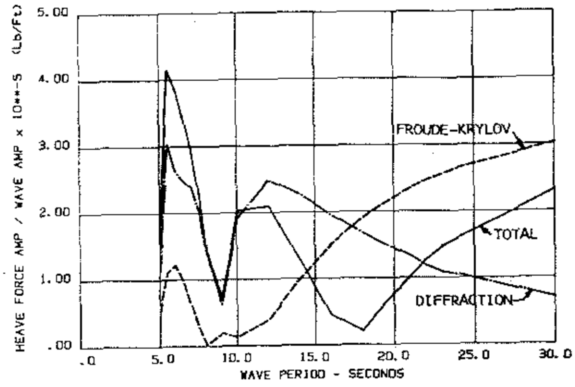

FIG. 3 GENERIC SEMI., COMPONENTS OF HEAVE FORCE BY PANEL METHOD

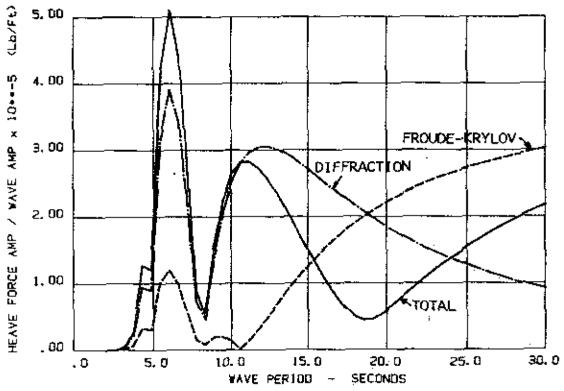

FIG. 4 GENERIC SEMI., COMPONENTS OF HEAVE FORCE BY MORISON FORMULA

see that the general behavior of the force is nearly the same by both methods although the values given by the Morison formula are somewhat higher than those by the panel method. In particular, points of cancellation or reinforcement of forces are replicated quite well, and the low frequency limit is nearly the same by each method.

The heave response of this platform, shown in Figure 5, has been computed by a quasilinear procedure which includes the equivalent linear damping of the Morison model. The highest resonant response is obtained using the panel method, and the height of the resonant peak can be attributed to the low damping near resonance of this procedure. Results are shown for the Morison formula model using two different values of C_M

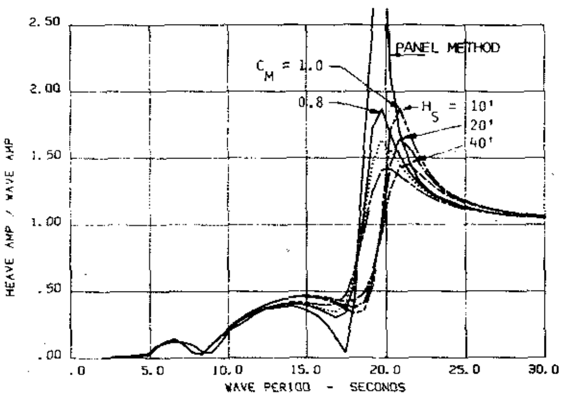

FIG. 5 GENERIC SEMI., HEAVE RESPONSE BY QUASILINEAR METHOD

and for the three sea states mentioned earlier. The lower value of C_M brings the Morison model added mass into close agreement with the panel method resulting in nearly equal resonant frequencies.

Experimentally determined heave responses for a 1:50 scale model of this platform are shown in Figure 6. The plotted points were obtained from tests in waves representing full scale wave heights between 16 and 106 feet. On the basis of past experience, the lower waves would be expected to yield results in good agreement with the quasilinear computations. The highest waves were used in an attempt to ascertain the upper limit of applicability of the procedure. It is seen that the quasilinear computations, shown by the solid line in Figure 6, are in good agreement with experiments over the entire range of wave periods although the limited number of experiments in extreme waves is insufficient to clearly illustrate the effect of wave height.

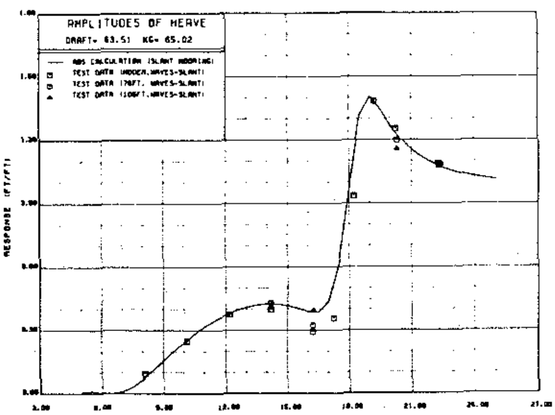

FIG.6 GENERIC SEMI. EXPERIMENTAL HEAVE RESPONSE

An example of the second type of nonlinear response, i.e., a response not present in a purely linear analysis, is shown in Figure 7. This response was computed using the nonlinear time-domain integration procedure described in [7] and [8]. It consists of a low frequency rolling motion occurring at the frequency of the envelope of a composite wave system formed by the superposition of two sinusoidal wave trains of slightly different frequencies. Similar behavior was observed in the model response in experiments conducted in such composite waves. The explanation for this effect lies in the phenomenon of wave drift forces on a surface-piercing spar. Here, it is known that drag forces on the surface piercing part of the spar combined with wave scattering about the spar result in a nonzero mean force in the direction of wave motion. In the simple group waves, this force and its moment have maximum and minimum values at the maxima and minima of the wave envelope. If the envelope period coincides with the natural period of roll, resonant rolling motion may be excited. These effects will be present in random seas as well, having a frequency content similar to that of the envelope of the random waves.

The above is a nonlinear dynamic phenomenon, not amenable to analysis by a purely linear procedure but clearly and accurately depicted in a nonlinear time-domain simulation. Other effects that are most easily studied by this type of simulation are responses to complex multiple disturbances such as combined wind and waves, transient effects such as those due to impulsively applied wind, and effects of damage which may include modelling of flooding of the interior of the ship or platform. A simulation of the latter behavior was described in [9] in which the procedure took into account water ingress through downflooding openings which were intermittently immersed by wave and platform action.

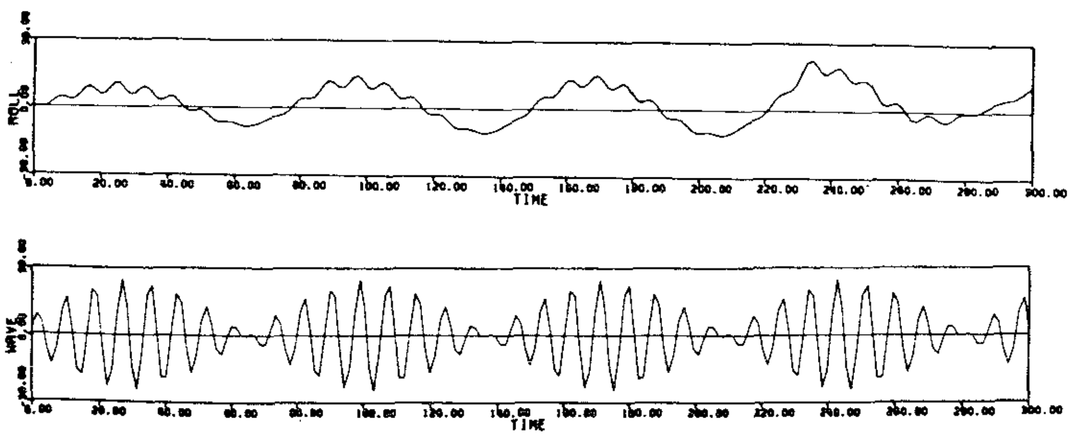

FIG.7 GENERIC SEMI., RESPONSE TO GROUP WAVES

The simulated response to simultaneous wind and wave action is shown in Figure 8. Computations of the response to waves alone and to wind alone showed that, under certain circumstances, the wind-induced roll response could exceed the wave induced roll. The reason for this is that the wind turbulence spectrum [10] contains a substantial low frequency content in the vicinity of the roll resonance.

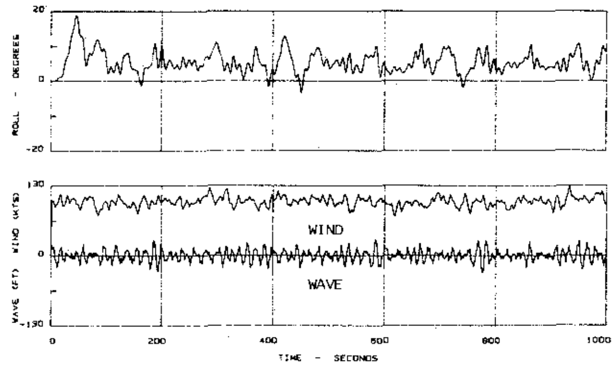

FIG. 8 GENERIC 8-COL SEMI., RESPONSE TO COMBINED WIND AND WAVES

The last two figures, 9 and 10, illustrate the application of the time-domain integration to the response of a damaged platform. Figure 9 gives a comparison of experimental and numerical results and serves as a calibration of the procedure. The wave system was simulated by 44 components whose periods, amplitudes and phases were derived by Fourier analysis of the experimental wave record. Figure 10 depicts a simulated capsize of the damaged platform under simultaneous action of wind and waves. During capsize, the platform rolls through 180 degrees and attains a stable position in this inverted attitude.

CONCLUSIONS

Large amplitude motions leading up to capsize often result from effects not adequately treated by the traditional linear techniques. Nevertheless, by using time-domain integration of the equations of motion coupled with somewhat intuitive and judicious extensions of existing linear hydrodynamic techniques, quite good predictions of large amplitude motions may be carried out for a number of important capsize or large motion scenarios. The principal drawback to time domain simulations is excessive computing time and a lack of generality of results.

Consequently, the method finds greatest application to the study of specific scenarios involving unusual events such as episodic waves, casualties and specific nonlinear motion responses.

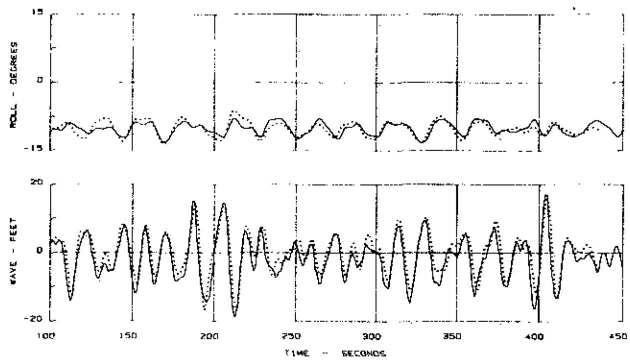

FIG. 9 GENERIC SEMI., DAMAGED CONDITION, EXPT. ——— , SIMULATION .....

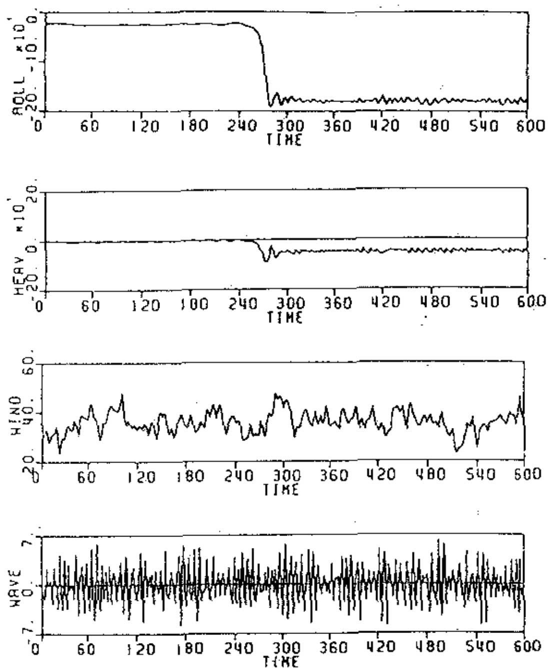

FIG.10 GENERIC SEMI., DAMAGED CONDITION
SIMULATION OF CAPSIZE

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ON THE STABILITY OF ANTISYMMETRIC MOTIONS OF A SHIP EQUIPPED WITH PASSIVE ANTIROLLING TANKS

Alberto FRANCESCUTTO and Vincenzo ARMENIO (*)

ABSTRACT

Until now, the effectiveness of passive antirolling devices, such as tanks, has been checked against the build-up of a large amplitude synchronized rolling motion in monochromatic or narrow band sea. Recently, we have shown that this effectiveness extends also in a general stochastic and short crested sea from any direction. Through the use of a perturbation method, we have also proved that it extends to the so called parametric rolling, i.e. to subharmonic rolling that can be excited as a result of the coupling of heave and roll in longitudinal sea from the stern. The effect of the tanks is to increase the threshold for the onset of subharmonic rolling at least in the first region of instability that was investigated in a second order approximate analysis. Numerical simulation confirms this trend.

Recent approaches to stability assessment seem to pay an increasing importance to the simultaneous occurrence of instability in the antisymmetric motions sway, yaw and roll, and synchronism with the external excitation. The coupling among these motions could explain phenomena such as capsizing, loss of control in waves and, probably, broaching-to.

In this paper, the stability boundaries as regards the antisymmetric motions are computed for a ship considering the eigenvalues of the system of equations describing the coupled sway, yaw, roll, tanks motions. The results indicate that passive tanks properly adjusted to avoid large amplitude rolling in a beam sea, can play an important role also in reducing the range of instability of these motions.

INTRODUCTION

In the last years an increasing attention is being payed to the understanding of the different mechanisms that can lead to ship loss, i.e., in the case of intact ship, to the capsizing and broaching-to phenomena. Whereas the first is connected with large amplitude rolling originated by different mechanisms such as resonance (gradual build-up or jump between different amplitude oscillation states), parametric resonance and loss of stability in waves, the second is the result of quite complex phenomena tied to directional instability, loss of control of control surfaces when surfing, yawing moment generated by waves and loss of stability [1]. Broaching-to is, with respect to capsizing, a less known, feared phenomenon.

Classically, directional stability and stability in the sense of safety from capsizing are considered separate events, the first analysed in terms of stability of the trivial solution of the coupled system of equations governing sway and yaw in linear approach, the second studied by means of methods that, started being fundamentally based on static approaches, later on incorporated more realistic descriptions of the action of the

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meteomarine environment and of the dynamics of nonlinear systems, are becoming of increasing complexity.

This approach, considering uncorrelated the two kinds of instability, is actually considered philosophically unacceptable; at the same time more powerful analytico-numerical tools for the analysis of the qualitative and quantitative behaviour of systems governed by systems of differential equations strongly coupled have been developed.

Modern views on ship safety consider both sources of instability very important and it is physically reasonable to consider a coupling between them. As a consequence, an increasing number of researchers [2-5] is considering with interest the analysis of the system of the equations governing the antisymmetric motions of the ship, i.e. roll, sway and yaw.

What we are actually looking for, are the conditions for the boundedness of the solutions in terms of rolling amplitudes acceptable not only in terms of avoiding capsize, but also in terms of ship operativity, and simultaneously an acceptable level of controllability of the ship that can be used both to avoid ship-wave conditions dangerous for excessive rolling and broaching-to.

To get a useful response to the above questions, only a fully nonlinear model can be sufficiently reliable. Unfortunately, present methods of theoretical and numerical hydrodynamics in general do not allow the statement of a coherent mathematical model sufficiently reliable to describe the behaviour of a strongly nonlinear system when large amplitudes are involved. On the other hand, experiments are often too much correlated to the particular type of ship considered. Moreover, the analysis of such a model can be done analitically in the frequency domain by the use of complex perturbation methods or numerically, by means of time domain simulation. The first approach allows a greater insight into the physics of the phenomena, as it allows a parametric analysis on the separate effect of the different terms and on the dependence on the initial conditions but is generally limited to non very large amplitudes and is quite complicated to use when bifurcations or stochastic excitation are considered [6,7]. The second can be always applied, in the limits of the validity of the model, it is characterized by a high precision, but does not give simple information about the dependence on the parameters and initial conditions. Moreover, it has to be used with caution when a bifurcation scenario is possible.

By the way, we have mentioned the validity of the model, a problem that has to be mentioned when considering large amplitude motions and stochastic excitation. It is, infact, difficult to write down a non ambiguous mathematical model for a strongly nonlinear system, as the coefficients become frequency dependent and it is not very clear what means frequency in presence of an excitation with bandwidth. On the other hand, it is clear that a linearized model is not in general a good one as predictions are often in qualitative disagreement with the effective behaviour.

At the other extreme, a linear model gives a good picture of the system behaviour at very small amplitudes, predicting with good accuracy the possible instabilities of the trivial solution in absence of excitation, that is the intrinsic instability of the system "in the small". Another question is the degree of correlation between these instabilities and the "instability in the large", that is the presence of solutions unbounded or not sufficiently bounded. This question can be solved, with a lot of difficulties and some ambiguities, only by means of a nonlinear approach, so that the position of researchers saying that instability in the linear approach does not mean that capsizing or broaching-to will take place is partly correct. Nevertheless, as in the case of parametric resonance in following sea, here too we can have indications about a possible loss of controllability or loss of transverse stability tied to directional instability. Since these information can be obtained by simple means, it appears of great interest to develop the algoritms that allow this and to compare the parametric indications with the results of casualties at sea. Preliminary results in this sense obtained by other authors [3] seem to be encouraging, as evidence of some correlation between intrinsic linear instability in the system of antisymmetric motions and the effective loss of the ship was found. On the other hand, we have not to forget that correlation between IMO "statistical" stability criterion (worldwide accepted!) is only a statistical one, that is by its own nature "a posteriori", whereas IMO "weather" takes into account only a very simplified description of the action of the environment and thus is far from being satisfactory.

Having in mind the importance of the coupling among the antisymmetric motions on one hand and the importance played by damping in the boundedness of the rolling amplitude on the other, it appears quite natural that the next step should be the matching of the two problems, with particular regard to the effect of devices designed to reduce rolling. This requires upgrading of the system describing the antisymmetric motions with a description of the effect of the antirolling device. In this paper we will devote our attention to the passive antirolling tanks, whose effectiveness, once tuned in regular beam sea, was proved in a stochastic bi- and tridimensional sea and in a regular following sea [8,9]. In this last case, properly adjusted tanks can increase the threshold for the parametric rolling, so reducing the probability of its occurrence.

The reason for this study lies also in the renewed interest towards stabilization systems of the U-tank type due to the versatility obained employing a passive controllable version [10,11]. This device can, in fact, countercact efficiently both rolling and heeling. It is thus useful either during routeing and during loading-unloading and other operation in the port. In particular, it could be used to reduce probability of cargo shifting due to excessive rolling and to control heeling during eventual cargo shifting avoiding dangerous positive feedback leading to structural failure and capsizing.

In practice, a fourth equation, representing the motion of the liquid in the tanks, has to be added to the other three and proper coupling terms have to be inserted in all equations. A first approach to this problem was done by Vasta et al. [12] that considered the system of differential equations describing some motions plus the motion of the fluid in the tanks. The coupling among the different motions was considered only with reference to the terms of mechanical origin involved in the description of the rigid body motion through the Eulero's equations. Here we reconsidered the problem including also the main hydrodynamic coupling terms.

EQUATIONS OF ANTISYMMETRIC MOTIONS

In nondimensional terms, the system of differential equations governing the antisymmetric motions of sway, roll and yaw in the reference frame of body axes, can be written as [4]:

$$(m-Y_{\dot{v}})\dot{v}-Y_{\dot{p}}\dot{p}-Y_{\dot{r}}\dot{r}-Y_{v}v-Y_{p}p+(m-Y_{r})r-Y_{\varphi}\varphi \\ -Y_{vv\varphi}v^2\varphi-Y_{v\varphi\varphi}v\varphi^2-Y_{rr\varphi}r^2\varphi-Y_{r\varphi\varphi}r\varphi^2=Y(t)$$

$$(I_x-K_{\dot{p}})\dot{p}-K_{\dot{v}}\dot{v}-(I_{xz}+K_{\dot{r}})\dot{r}-K_{p}p-K_{v}v-K_{r}r+(\nabla GM/F_n^2 \\ -K_{\varphi})\varphi-K_{vv\varphi}v^2\varphi-K_{v\varphi\varphi}v\varphi^2-K_{rr\varphi}r^2\varphi-K_{r\varphi\varphi}r\varphi^2=K(t)$$

$$(I_z-N_{\dot{r}})\dot{r}-N_{\dot{v}}\dot{v}-(I_{zx}+N_{\dot{p}})\dot{p}-N_{r}r-N_{v}v-N_{p}p-N_{\varphi}\varphi \\ -N_{vv\varphi}v^2\varphi-N_{v\varphi\varphi}v\varphi^2-N_{rr\varphi}r^2\varphi-N_{r\varphi\varphi}r\varphi^2=N(t)$$

This nonlinear non-autonomous approach to antisymmetric motions was used by Son and Nomoto to analyse the coupling of roll and yaw and allows an analysis of the stability of a motion in the large, by means of a perturbation. As a result, they proved that the effect of this coupling, whose origin is to be found in the lift forces and moments generated by the hull in nonsymmetric flow due to a leeway angle and a heeling, is contradictory. In fact, turning ability is improved, whereas course keeping stability and quick response are decreased and heeling is enhanced. In the present analysis, we are interested in the intrinsic stability of the trivial solution (i.e. no yaw, no sway, no roll), so that we can neglect the external forces and moments, included those due to the rudder action, and linearize the system in the neighbohrood of the origin. This way we get the following autonomous system of differential equations (written in matrix form):

$$\begin{vmatrix} m-Y_{\dot v} & -Y_{\dot p} & -Y_{\dot r} \\ -K_{\dot v} & I_x-K_{\dot p} & -(I_{xz}+K_{\dot r}) \\ -N_{\dot v} & -(I_{zx}+N_{\dot p}) & I_z-N_{\dot r} \end{vmatrix} \begin{vmatrix} \dot v \\ \dot p \\ \dot r \end{vmatrix} +$$

$$\begin{vmatrix} -Y_v & -Y_p & m-Y_r \\ -K_v & -K_p & -K_r \\ -N_v & -N_p & -N_r \end{vmatrix} \begin{vmatrix} v \\ p \\ r \end{vmatrix} +$$

$$\begin{vmatrix} 0 & -Y_\varphi & 0 \\ 0 & \nabla GM/Fn^2-K_\varphi & 0 \\ 0 & -N_\varphi & 0 \end{vmatrix} \begin{vmatrix} 0 \\ \varphi \\ 0 \end{vmatrix} = \begin{vmatrix} 0 \\ 0 \\ 0 \end{vmatrix}$$

Considering a perturbed solution of the form

$$v=v_o e^{\lambda t} \quad \varphi=\varphi_o e^{\lambda t} \quad r=r_o e^{\lambda t} \quad (p=p_o\lambda e^{\lambda t})$$

where r_o, v_o, p_o represents the effect of a small perturbation and substituting in the linear system, one has the following system of algebraic equations in the parameter λ:

$$\begin{vmatrix} \lambda(m-Y_{\dot v})-Y_v & -\lambda^2 Y_{\dot p}-\lambda Y_p -Y_\varphi & -\lambda Y_{\dot r}+(m-Y_r) \\ -\lambda K_{\dot v}-K_v & \lambda^2(I_x-K_{\dot p})-\lambda K_p +\nabla GM/Fn^2-K_\varphi & -\lambda(I_{xz}+K_{\dot r})-K_r \\ \lambda N_{\dot v}-N_v & -\lambda^2(I_{zx}+N_{\dot p}) -\lambda N_p-N_\varphi & \lambda(I_z-N_{\dot r})-N_r \end{vmatrix} \begin{vmatrix} v_o \\ \varphi_o \\ r_o \end{vmatrix} =0$$

The condition for the existence of a non trivial solution, i.e. for the instability of the motions, consists in the vanishing of the determinant $\Delta(\lambda)$. The values of λ, called the eigenvalues of the system, that realize this condition, are found as the roots of the following characteristic equation:

$$\beta_o\lambda^4+\beta_1\lambda^3+\beta_2\lambda^2+\beta_3\lambda+\beta_4=0$$

where the β_i's are complicated functions of the hydrodynamic derivatives and of the GM/Fn^2 ratio

The stability of the system of differential equations governing the antisymmetric motions can be analysed determining the locus of the roots of the characteristic equation in the complex plane. For stability it is necessary and sufficient that the roots be either pure imaginary or complex with negative real part. In this case, the solution represented by the initial perturbation is a transient that dies out in time converging asymptotically towards the null solution. A necessary and sufficient condition can be expressed in terms of the Liénard-Chipart criterion (often improperly quoted as Routh-Hurwitz criterion), i.e.requiring that all the coefficients of the characteristic equation, being real, satisfy the simultaneous conditions:

$$\beta_o>0 \quad \beta_1>0 \quad \beta_2>0 \quad \beta_3>0 \quad \beta_4>0$$

and

$$\beta_1(\beta_2\beta_3-\beta_4\beta_1) - \beta_o\beta_3^2 >0 \; .$$

It is not very easy to interpret physically these conditions, except the last one that worths a detailed analysis. One has, infact:

$$\beta_4=(\nabla GM/Fn^2-K_\varphi)[N_rY_v+N_v(m-Y_r)] +Y_\varphi(K_vN_r-N_vK_r)+N_\varphi[Y_vK_r+K_v(m-Y_r)]$$

so that a necessary condition for stability is:

$$GM > \frac{Fn^2}{\nabla}\left[K_\varphi - Y_\varphi\frac{R_1}{R_3} - N_\varphi\frac{R_2}{R_3}\right]$$

with obvious meaning for R_1,R_2 and R_3. In particular, R_3 represents the course-keeping stability (condition is $R_3>0$)

This condition differs from that obtained by Bao-an [5] only for the presence of the hydrodynamic hull lift term K_φ. It represents a coupling between static stability and course keeping stability. When $K_\varphi=Y_\varphi=N_\varphi=0$, the condition reduces to:

$$GM\,[N_rY_v+N_v(m-Y_r)] = 0$$

i.e. to

$$GM > 0 \qquad [N_rY_v+N_v(m-Y_r)] > 0$$

that is the two stability conditions are uncoupled and we recover the model proposed by Bishop, Price and Temarel in the case of the slow motion derivatives. Of course, the coupling between the two aspects is not limited to the term β_4, as we will see later.

Since the three terms neglected to arrive at the last condition represent a contribution due to the hull lift in non symmetric flow due to heeling, it could be expected that they play a relevant (and highly nonlinear) role at high speed and represent a major contribution to limit the stability in the large. In fact, Son and Nomoto [4] found a positive feedback effect in yaw-roll. In the present analysis, attention is devoted to stability in the small, so that we neglect the hydrodynamic derivatives depending on heeling angle.

ANTISYMMETRIC MOTIONS OF A SHIP EQUIPPED WITH PASSIVE U-TANKS

The antisymmetric motions of a ship equipped with passive antirolling U-tanks are governed by the following system of four equations:

$$(m-Y_{\dot v})\dot v-Y_{\dot p}\dot p-Y_{\dot r}\dot r-Y_v v-Y_p p+(m-Y_r)r+d_1\ddot\psi=0$$

$$(I_x-K_{\dot p})\dot p-K_{\dot v}\dot v-(I_{xz}+K_{\dot r})\dot r-K_p p-K_v v-K_r r+\nabla GM/F_n^2\varphi+c_1\ddot\psi+c_3\psi=0$$

$$(I_z-N_{\dot r})\dot r-N_{\dot v}\dot v-(I_{zx}+N_{\dot p})\dot p-N_r r-N_v v-N_p p+f_1\ddot\psi=0$$

$$b_1\ddot\psi+b_2\dot\psi+b_3\psi+c_1\dot p+c_3\varphi+d_1\dot v+f_1\dot r=0$$

The quantities representing ship mass, moments of inertia and metacentric height have been corrected for the presence of the liquid in the tanks, coefficients b_1 to b_3 represent the dynamics of the tanks alone, c_1 and c_3 the usual coupling between fluid motion in the tanks and roll motion. Finally, d_1 and f_1 represent the coupling between fluid motion in the tanks and sway or respectively yaw. They are tied to the accelerations induced in the horizontal tanks channel; the first is always present, whereas the second depends on the longitudinal distance of the tanks from center of gravity. Here too we use nondimensional coefficients.

The equations of the motions have been written following the approach developed in [3,4], neglecting the terms depending on heeling since they are particularly efficient means of energy transfer between different motions at high speed where the passive tanks partly loose effectiveness.

The analysis of the stability of the system can be done in exactly the same way used in the preceding section. To this end, assumed a perturbed solution in the form:

$$v=v_o e^{\lambda t} \quad \varphi=\varphi_o e^{\lambda t} \quad r=r_o e^{\lambda t} \quad \psi=\psi_o e^{\lambda t}$$

and substituting in the system of differential equations, one gets the following characteristic equation for the eigenvalues:

$$\beta_o\lambda^6+\beta_1\lambda^5+\beta_2\lambda^4+\beta_3\lambda^3+\beta_4\lambda^2+\beta_5\lambda+\beta_6=0$$

Applying the Routh-Hurwitz criterion, one has in particular

$$\beta_6=[b_3\nabla GM/F_n^2-c_3^2]\,[N_rY_v+N_v(m-Y_r)]>0$$

This condition is of the same form of that already obtained, with the two contribution to instability uncoupled. The difference is to be attributed to the effect of the liquid in the tanks that changes the metacentric height due to the change in the height of the center of gravity and at the same time contributes a static heeling moment, so that now we have the little more restrictive condition

$$GM>\frac{c_3}{\nabla/F_n^2},$$

i.e. $GM>\dfrac{c_3}{\rho g\nabla}$ in dimensional terms.

RESULTS AND CONCLUSIONS

The analysis of eigenvalues and eigenvectors has been done for the same ship studied in [3], in a loading condition corresponding to GM=0.70 m and a tipical operative speed corresponding to $F_n=0.2$, equipped with passive U-tanks. The optimal condition for the tanks was found as outlined in [8]. The trim, represented by the nondimensional quantity $\gamma=\Delta T/T_m$ was varied to analyse the proposed destabilizing effect of the trim by the bow condition. The effect of the tanks regulation has been studied comparing the properly adjusted in beam sea condition condition (same frequency of the rolling motion and optimal damping in the channel) with an off tuning condition (same frequency but with negligible damping in the channel).

To detect which kind of instability is connected with the appearance of an eigenvalue with real part positive, the corresponding velocities eigenvectors have been computed. As these give the amplitude of the normal modes of the system, it is convenient to obtain them in relative form dividing their modulus by the modulus of that corresponding to yaw motion, except that corresponding to the fluid motion in the tanks that was correlated with the rolling velocity eigenvector. With the positions:

$$\alpha=\frac{|v|}{|r|} \quad \beta=\frac{|p|}{|r|} \cdot \delta=\frac{|p|}{|\dot{\psi}|}$$

one has that the (eventual) instability is sway dominated if $\alpha>1$, roll dominated if $\beta>1$, yaw dominated if $\alpha<1$ and $\beta<1$. A little more complicate is the behaviour when $\alpha \cong \beta \cong 1$.

Let us examine firstly the results relative to the stability of antisymmetric motion of the ship without stabilizing tanks. In Figg. 1 and 2 the eigenvalues and corresponding eigenvectors versus the nondimensional trim coefficient γ are presented. As one can see, two eigenvalues, μ_1 and μ_2 are positive, indicating corresponding ranges of instability. A look at the eigenvectors indicates that the first instability is of the unidirectional diverging type and is yaw dominated, whereas the second is of the oscillatory diverging type and roll dominated.

Considering now the same ship with well regulated tanks, Fig.3 and 4, one can see that no significant variation in the first eigenvalue can be observed, whereas the second is pushed towards lower values, so widening the range of stability. This means that a disturbance to the upright position is reduced to zero, that is the ship is stabilized with respect to rolling motion in a wider range of trim condition by the action of the tanks. In this case, the eigenvalues are six, two couples of which are complex conjugate, since there are two motions that can display oscillatory behaviour, i.e. the ship roll and the fluid in the tanks. The imaginary parts of these eigenvalues represent the frequencies of the normal modes of these two oscillating systems, as one could see by comparison of the eigenvalues with the eigenvalues of the system of the two differential equations governing the ship+tanks without considering the other motions.

The regulation of the towing tanks plays a very important role as one can see comparing preceding results with those, Fig. 5 and 6, corresponding to the ship with tanks with damping in the channel very low with respect to the optimum value. In this case, in the range $-1 < \gamma < 0.6$, i.e. in almost all the investigated conditions, the two complex eigenvalues have positive real part. This means that in this case, the tanks are unable to reduce rolling, acting on the contrary, as roll exciters.

These results, obtained in the linear, autonomous, approach, appear of interest inasmuch as they allow to conclude that the passive tanks, once well adjusted for the maximum roll reduction in a regular beam sea, preserve their effectiveness also in the case of antisymmetric motion, reducing the probability of the onset of dangerous rolling.

Comparing the eigenvalues corresponding to the different situations, one can observe that the real ones do not exhibit marked changes passing from ship without tanks, to well adjusted tanks to off regulation tanks. This means that the tanks have no relevant influence on the stability of the motions of sway and yaw. We suspect that this is due to a weak coupling between roll and the other two antisymmetric motions in this case, so that the system of four equations is actually composed of two subsistems weakly interacting, or at least interacting mainly one way. In fact, we have observed that global instability almost always consists of directional type instability to which, in particular conditions, a roll instability can be superposed. This means that, for this ship, some of the terms $K_{\dot{v}}$, K_v, $K_{\dot{r}}$, K_r can represent an efficient mean of energy transfer from sway-yaw unidirectionally diverging motions to roll,

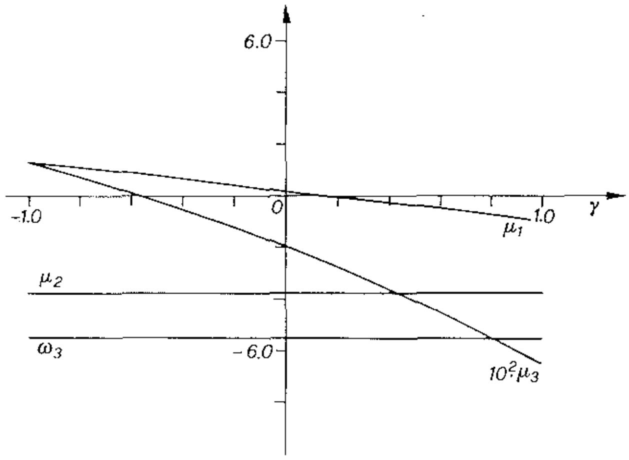

Figure 1. Eigenvalues corresponding to the ship without tanks versus nondimensional trim.

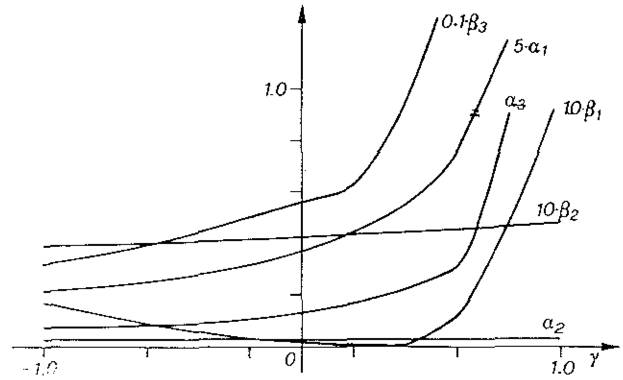

Figure 2. Eigenvectors corresponding to eigenvalues of Figure 1. α represents sway, β roll, yaw is unity.

exciting the instability of the latter when righting ability or damping or other means of dissipating this energy, like tanks, are unable to counteract.

The situation is probably different in a nonlinear approach, when large amplitude motions are involved, since there an efficient feedback bounds yaw and roll. In this case, an efficient action of the tanks could be expected to be exerted on the complex of the antisymmetric motions.

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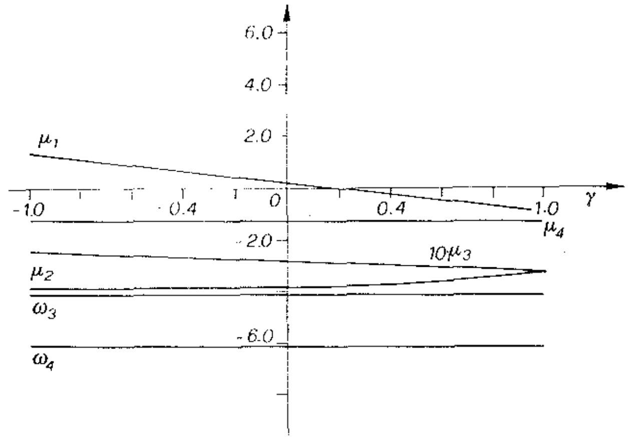

Figure 3. Eigenvalues corresponding to the ship with optimized tanks versus nondimensional trim.

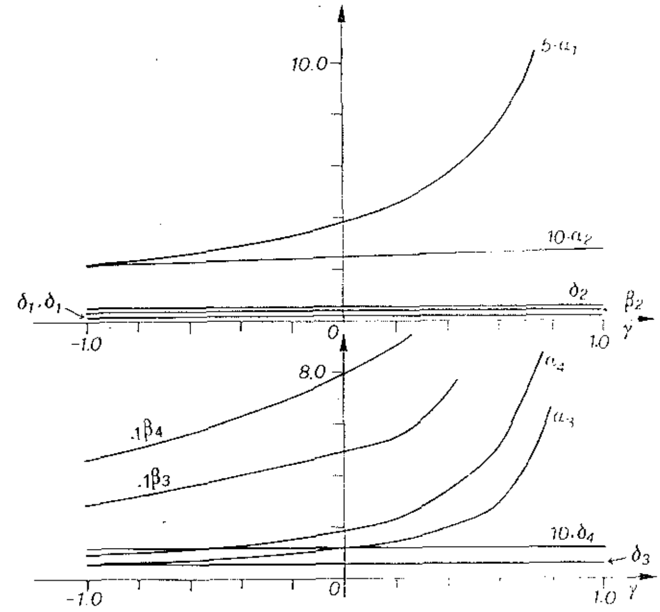

Figure 4. Eigenvectors corresponding to eigenvalues of Figure 2. α represents sway, β roll, yaw is unity, δ represents the fluid motion in the tanks.

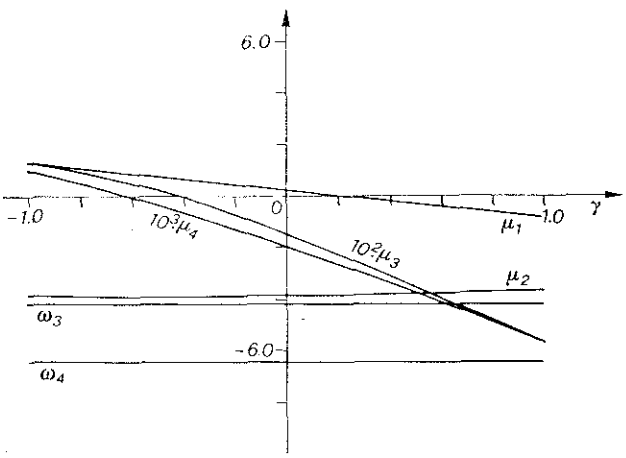

Figure 5. Eigenvalues corresponding to the ship with non optimized tanks versus nondimensional trim.

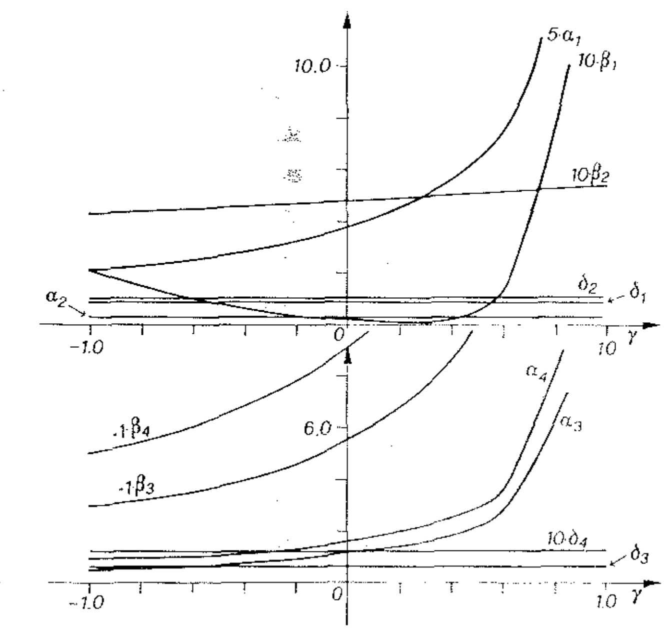

Figure 6. Eigenvectors corresponding to eigenvalues of Figure 5. α represents sway, β roll, yaw is unity, δ represents the fluid motion in the tanks.

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ON THE INFLUENCE OF RESTORING MOMENT CHANGES IN WAVES ON STABILITY ESTIMATIONS

P.Bogdanov [1], S.Dimitrova [2], R.Kishev [3]

ABSTRACT

In this paper a procedure is described for evaluation of the total sum components of the ship restoring moment in waves, developed in the frames of the hydrodynamic theory of ship motion and some results of its numerical realization are given. The validity of the approach is verified for the case of following waves by systematic captive model tests. The influence of the inclination angle, the speed of advance and the frequency of excitations on every restoring moment component is analysed. The stability estimation changes due to the restoring moment variations and the conditions, at which these changes influence considerably the stability assessment process, are examined.

INTRODUCTION

Ship stability in waves is characterized by the hydrodynamic moment, arizing from the counteraction of the exciting and restoring moments, as a function of the angle of inclination. When operating in real seas, the ship motions and sea waves result in alterations of the periodic restoring moment, thus stability estimations in this case could differ significantly from those in still water.

In the practice till now the stability variation has been discussed in a quasistatic formulation, which corresponds to the case of ship motion in following waves at speed close to that of the wave. In fact, in most of the cases this is not like that and it can be expected that the dynamic effects, related to the continuous variation of the waterline form and the wave forces due to ship motions and wave actions, could cause additional changes of the stability characteristics. In this aspect the works of Grim, Wendel, Paulling, Lugovsky, Kastner, etc. are well known and they are generally characterized by the attempt to simplify the formulation of the problem (considering hydrostatic pressure changes only, wallsided ships, small heeling angles, etc.), which leads to qualitative evaluations. Most thoroughly the problem has been formulated and developed by Boroday and Netzvetaev (1), where all components of the hydrodynamic moment arising in waves, are considered. On the basis of this formulation a practical estimation procedure has been developed at BSHC (2), (3), and further below some calculation results are given to demonstrate its applicability and importance. Meanwhile, a large experimental programme has been launched on the problems of ship stability in following waves and some initial results have been used here for comparison purposes, the main body of the results being meant to be published later.

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1. FORMULATION OF THE PROBLEM

In (1) the restoring moment in waves is defined as a moment of the time-dependent hydrodynamic forces acting on the ship hull inclined at fixed heeling angle but free to move otherwise. The potential problem is formulated in the frames of the linear hydrodynamic theory (small excitation and reaction amplitudes). The fluid motion is described in coordinates, as shown on Fig.1. The corresponding linearized velocity potential is presented in (1) in the common form:

$$\phi = \phi_S+\phi_1+\phi_2+\phi_3 \qquad (1)$$

where, ϕ_S - potential of steady motion in still water,

ϕ_1 - incident wave potential,

$\phi_2 = \sum_{j=1}^{6} \dot{X}_j \varphi_j$ - forced motion potential

ϕ_3 - diffraction potential.

The total hydrodynamic moment

$$M_T = - \int_S (p-p_a)(\bar{r} \times \bar{n}) ds \qquad (2)$$

where the pressure is evaluated by the velocity potential of the fluid motion, can be analogously expressed, as follows:

$$M_T = M_S+M_1+M_2+M_3 \qquad (3)$$

where, M_S is caused by pressure field changes when the ship advances in still water,

M_1 is Froude-Krilov component,

$M2$ is caused by ship forced motion in still water at fixed heeling angle,

M_3 is a diffraction component,

and where every component is to be related to the corresponding fraction potentials evaluated at fixed roll assumption and integrated over the correspondingly estimated wetted hull surface (1).

Obviously, the additional moment in waves

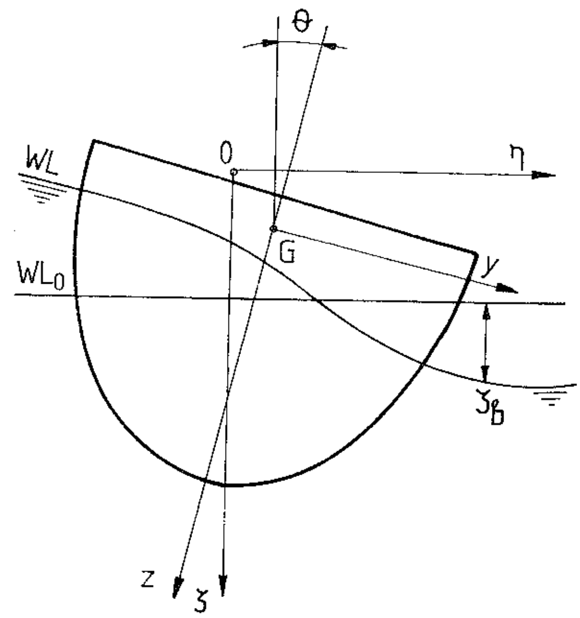

Fig.1. Coordinate Systems

is obtained by

$$M_{AW} = M_T - M_{SW} \qquad (4)$$

and appears as a function of time and the heeling angle.

The motions of the inclined ship restricted in roll can be derived from the general linearized motion equations:

$$\sum_j (M_{jk}+A_{jk})\ddot{X}_k+B_{jk}\dot{X}_k+C_{jk}X_k) = F_j e^{-i\omega t} \qquad (5)$$

considering the existing hull asymmetry and retaining heave and pitch terms only as most significant in forming the fluid motion, i.e. k = 3,5. The corresponding inertia and damping coefficients for inclined cross sections are evaluated by Elis method (4).

2. EVALUATION OF THE MOMENT COMPONENTS

To express the components of the motion - induced additional restoring moment, reference (1) has been strictly followed, where from eqs.(1) and (2) the following general equations can be obtained:

$$M_1 = \gamma\int_{S_\theta} \zeta(y\vec{k}-z\vec{j})ds+\rho\int_{S_\theta} \frac{\delta\phi_1}{\delta t}(z\vec{j}-y\vec{k})ds \qquad (6)$$

$$M_2 = \rho\int_{S_\theta} \frac{\delta\phi_2}{\delta t}(z\vec{j}-y\vec{k})ds \qquad (7)$$

$$M_3 = \rho\int_{S_\theta} \frac{\delta\phi_3}{\delta t}(z\vec{j}-y\vec{k})ds \qquad (8)$$

where, S_θ is the wetted surface at inclined position, and

$\vec{j},\vec{k}$ are components of the unit normal vector.

Further below the components are expressed for the case of following waves:

2.1. Froude-Krilov Components

Changing the integration media and denoting that

$$\Delta V = V_T - V_\theta \qquad (9)$$

where, V_T is the instantaneous displaced volume and

V_θ is the at-rest-volume at inclined position,

M_1 fraction is resolved into three terms:

$$M_1 = M_{10} + M_{11} + M_{12} \qquad (10)$$

where, M_{10} accounts for the steady change of the waterline when the ship advances in still water,

M_{11} gives the influence of the additional liquid volume around the moving hull, and

M_{12} accounts for the pressure distribution changes in the surrounding fluid (Smith effect):

$$M_{11} = \gamma\cos\theta\int_{\Delta V} ydV-\gamma\sin\theta\int_{\Delta V} zdV \qquad (11)$$

$$M_{12} = \zeta_a\rho\omega^2(\cos\theta\int_{V_\theta} I_c ydV-\sin\theta\int_{V_\theta} I_c zdV)\cos\omega_e t+$$

$$+ \zeta_a\rho\omega^2(\cos\theta\int_{V_\theta} I_s ydV - \qquad (12)$$

$$- \sin\theta\int_{V_\theta} I_s zdV)\sin\omega_e t$$

$$I_c = e^{-k\zeta}\cos(kx)$$

$$I_s = e^{-k\zeta}\sin(kx)$$

and where, the heading $\mu = 0°$ is assumed.

2.2. Radiation Component

Retaining heave and pitch only and considering the radiation potential in eq.(7) as

$$\phi_2 = \dot{X}_3\Upsilon_3 + \dot{X}_5\Upsilon_5, \qquad (13)$$

where, $\dot{X}_3 = \dot{\zeta}_g + \upsilon\psi,$

$\dot{X}_5 = \dot{\psi},$

the radiation component can be expressed as:

$$M_2 = M_{21} + M_{22} \qquad (14)$$

where,

$$M_{21} = \ddot{\zeta}_g A_{34}+\dot{\zeta}_g B_{34}+\ddot{\psi}A_{54}+\dot{\psi}B_{54} \qquad (15)$$

$$M_{22} = \upsilon(\dot{\psi}A_{34}+\psi B_{34})$$

and where, M_{22} accounts directly for the speed influence on the moment

2.3. Diffraction Component

The incident potential being

$$\phi_3 = i\zeta_a\frac{g}{\omega}\, e^{-k\zeta+i(\omega_e t-k\xi)}, \qquad (16)$$

the diffraction component (8) for the case of following waves is reduced to:

$$M_3 = -(\ddot{\zeta}_b A_{34}+\dot{\zeta}_B B_{34}+\alpha A_{54}+\dot{\alpha}B_{54}) \qquad (17)$$

where, α is the wave slope, and

ζ_b is the instantaneous wave elevation

One can easily note that at upright position the components M_2 and M_3 turn to zero.

3. PRACTICAL CALCULATIONS

The outlined calculation procedure for estimation of the restoring moment changes in waves has been realized in (2) and (3) for the case of following waves and has been thoroughly checked experimentally on a S-175 containership model subjected to captive towing in following waves.

3.1. Ship Motions

The amplitudes of heave and pitch, ζ_{g_a} and ψ_a, were obtained by solving the set of linear equations (5). The hydrodynamic added masses and damping, symmetrical as well as asymmetric, were calculated by Elis method (4), which was experimentally verified in (2) by forced oscillations of a series of inclined ship cross-sections.

The results of the motion calculations are illustrated on Figs.2 and 3, where they are compared with the corresponding experimental values to prove the correctness of the numerical estimations.

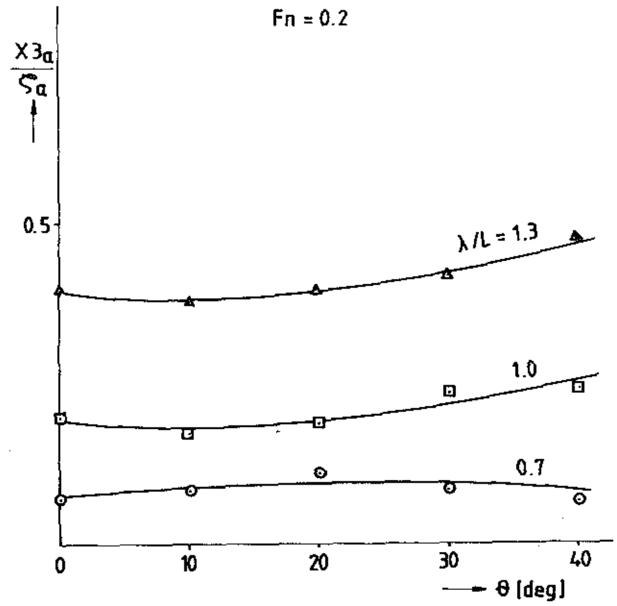

Fig.2. Heave Response in Following Waves

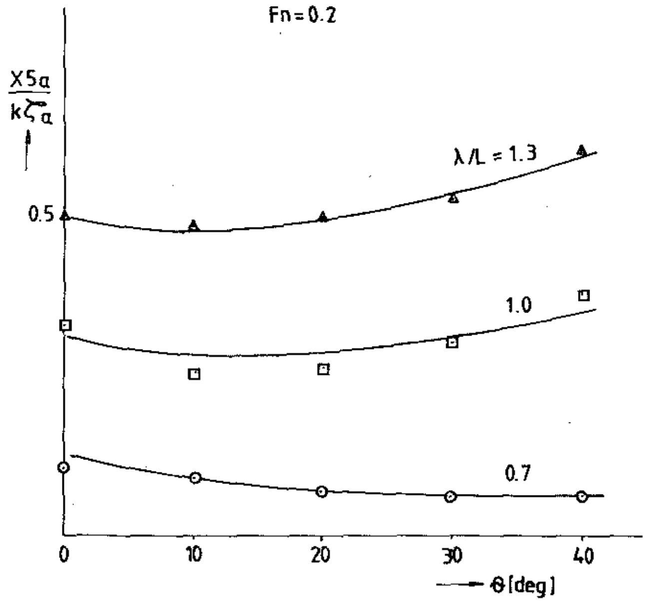

Fig.3. Pitch Reponse in Following Waves

3.2. Restoring Moment Components

The detailed calculations of the restoring moment fractions by eqs.(11), (12), (15) and (17) revealed the unconditional significance of Froude-Krilov term and the negligible importance of the other two, which is illustrated in Table 1.

Fig.4 shows the contribution of every fraction integrated in M_1 term. Obviously, M_1 amplitudes reach their minimum at $\lambda/L=1$, though not strongly expressed. The steady addition to the static moment, M_{10}, is minor and becomes negative at larger inclinations.

Table 1

$\lambda/L = 1.0$						
θ	M_{SW}	M_{AW}	M_{AW}	M_1	M_2	M_3
deg	tm	tm	$\%M_{SW}$	$\%M_{AW}$	$\%M_{AW}$	$\%M_{AW}$
10	4407	1222	27.7	94.6	3.1	1.7
20	10848	4760	43.9	98.7	1.6	0.7
30	21591	6899	32.0	99.9	1.8	0.9
40	28739	4509	15.7	101.5	3.7	2.5

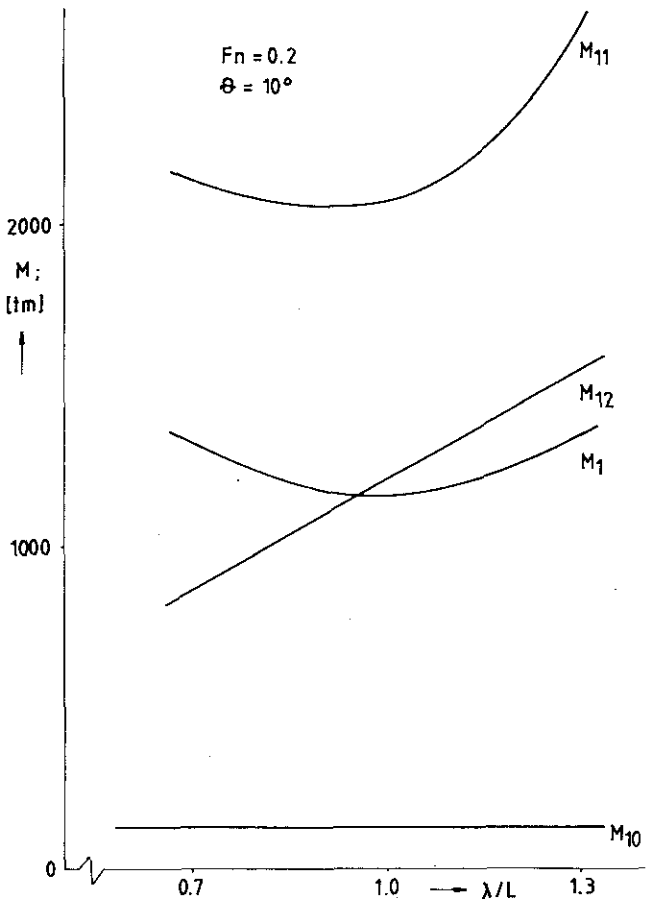

Fig.4. Correlation of Fractions Composing Froude-Krilov Part of the Restoring Moment in Waves

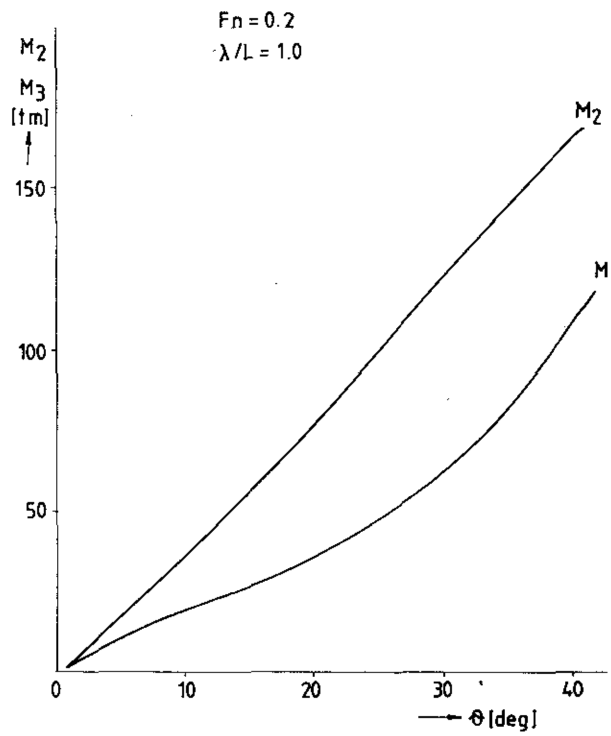

Fig.5. Radiation and Diffraction Moment Components Calculated as a Function of the Heeling Angle

Figs.5 and 6 show the change of the radiation and diffraction components with the change of the frequency and angle of heeling.

The harmonic changes of the restoring moment in waves, imposed over the static moment, leads to relevant fluctuations of the actual lever arm of stability. The calculated moment fluctuations are compared on Fig.7 with the corresponding experimental values obtained by recalculation of the captive towing test results, the good agreement between calculations and experiment being demonstrated. Since the model test program has not been completed yet, only sample results have been cited here.

As it has become clear, the decrease of the restoring moment within one period

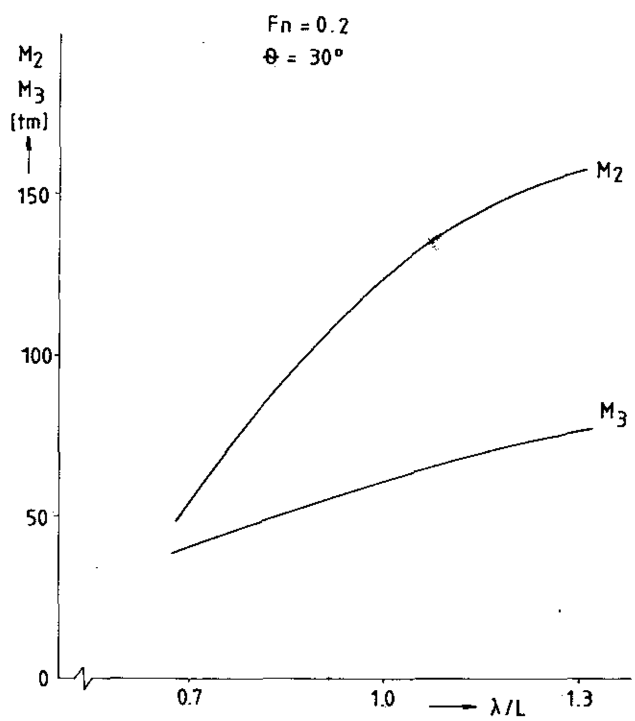

Fig.6. Radiation and Diffraction Moment Components Calculated as a Function of the Relative Wave Length

of motion can reach significant percentage and has to be considered in practical stability evaluations. Systematic calculations show that especially for the following wave case no remarkable speed influence has been detected on ship motions, as well as on the additional restoring moment and the same can be stated for the frequencies of encounter, as they change slightly. The most important operational factor appears to be the phase ratio between waves and inspired heeling (from rolling motion or by wind gust) because as the encounter wave frequency tends to be zero at certain (ω,υ) combinations and the moment fluctuations increase with the increase of the heeling angle, the simultaneous occurrence of large ship inclination and negative restoring moment amplitude could provoke danger.

The above conclusions are based on regular wave considerations but having in mind that the encounter wave spectrum is rather narrow, the same reasonings will do for realistic seas as well.

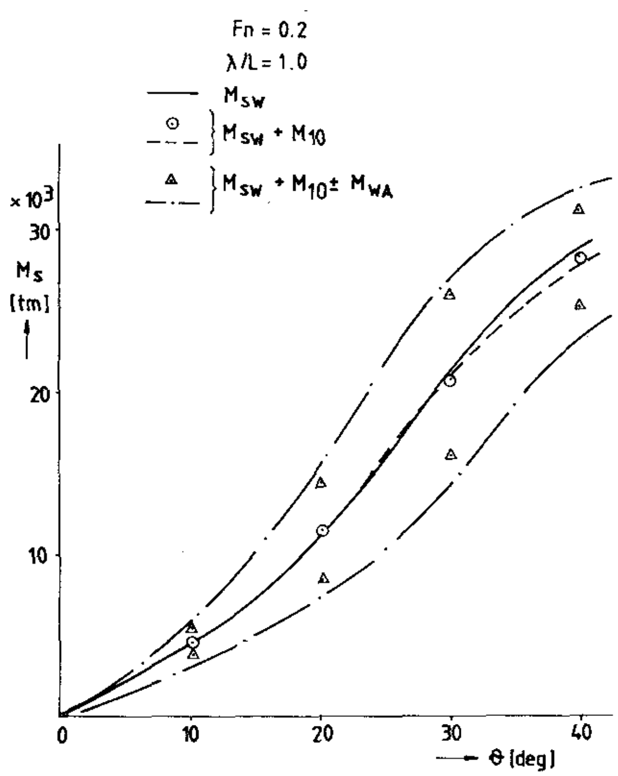

Fig.7. Restoring Moment Fluctuations in Following Waves

NOMENCLATURE

- A_{jk} - Added mass coefficients
- B_{jk} - Damping coefficients
- C_{jk} - Restoring coefficients
- F_j - Exciting forces
- Fn - Froude number
- g - Acceleration of gravity
- k - Wave number
- L - Ship length
- M_{jk} - Generalized masses
- M_{SW} - Restoring moment in still water
- M_T - Total hydrodynamic moment in waves
- M_{AW} - Additional restoring moment in waves
- $M_{1,2,3}$ - Components of the wave-induced restoring moment
- $\vec{n}$ - Unit normal vector to the ship hull
- p - Hydrodynamic pressure
- $\vec{r}$ - Radius-vector of the point on the hull surface
- X_k - Ship motion amplitudes
- x,y,z - Coordinate system fixed to the mean position of the ship at CG
- ξ,η,ζ - Translation coordinate system referred to WL and CL at upright position
- γ - Unit weight of water
- ζ_a - Wave amplitude
- ζ_g - Vertical displacement of CG
- θ - Heeling angle
- λ - Wave length
- ρ - Mass density of water
- φ_j - Fraction radiation potentials
- υ - Speed of advance
- ψ - Pitching angle
- ω - Wave frequency
- ω_e - Frequency of encounter

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ASPECTS OF DAMAGED STABILITY IN THE COMPUTER AUGMENTED DESIGN PROCESS FOR SWATH VESSELS

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ABSTRACT

This paper describes the creation of a design tool which enables the user to quickly and easily assess the damage stability characteristics of Small Waterplane Area Twin Hull (SWATH) ships at the preliminary design stage. The requirement for such a design tool is discussed and the subsequent stages in its development from parametric study through to completed design program are described. The paper also seeks to demonstrate that SWATH vessels possess acceptable damage stability characteristics which are at least equal to if not superior than those of equivalent monohulls.

BACKGROUND

SWATH is an acronym for Small Waterplane Area Twin Hull . This type of vessel , as the name suggests is a form of modified catamaran where the underwater form has been distorted to move the supporting buoyancy well below the surface of the sea and away from the wave action . A typical SWATH vessel consists of two totally submerged torpedo like hulls upon which an above water cross structure or box is supported by means of long streamlined surface piercing struts.The resulting vessels have demonstrated dramatically improved seakeeping performance over conventional monohulls and catamarans at both model and full scale , however the low waterplane area inherent in the concept presents several problems not the least of which is sensitivity to changes in weight or flooding.

The work described in this report was undertaken as part of the first joint Vickers Shipbuilding and Engineering Ltd / Glasgow University 'SWATH Design and Evaluation Project'. This project was initiated with the ultimate aim of enhancing the computer augmented SWATH design capability previously developed at the University(Ref 1,2,4).

INTRODUCTION

Despite the sensitivity to weight changes associated with the low waterplane area and the uncertainty attached to the stability of these vessels when damaged, surprisingly little work has been published in the field (Ref 3). This paper is directed towards changing that situation and seeks to reassure potential SWATH ship operators that survivability for SWATH vessels is at least comparable to that for monohulls .

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It is obviously desirable to include consideration of a vessels ability to survive damage when evaluating design proposals. Conventional damage stability software packages capable of handling the novel geometry of the SWATH form are available however all existing programs require detailed design information and are both time and labour intensive.Since the Naval Architect is often faced with the task of evaluating a large number of alternative design proposals for a given vessel , the value of a tool providing fast , first estimates of damage stability becomes clear. Ideally , such a tool should be quick to use and require only preliminary design data.

A workplan was therefore formulated with the ultimate aim of extending the existing "DESIN" suite of SWATH design programs to include damage stability considerations in the design process.

APPROACH ADOPTED

The Parametric Study

In order to provide such a capability the links between design geometry and survivability must be explored and relationships between the two established. To this end , a parametric study was selected as the most suitable vehicle for the first part of the work. Results from this study were analysed and mathematically defined to allow the construction of a program which predicts damaged behaviour at the earliest stages of the design process.

The first stage was to create a 'family' of SWATH vessels , that is , vessels whose principal dimensions and geometrical proportions were closely related. These vessels are not geosims in the true sense but share the same basic proportions for the main design variables : for instance , hull/strut length ratios , strut setback , nose and tail run-in , run-out etc. The computer program 'DESIN' was used to synthesize this family of ships for five displacements in the range 1000 to 5000 tonnes. It is felt that this displacement range covers most likely SWATH newbuildings in the foreseeable future.

Simple circular hulls with elliptical noses and paraboloidal tails were chosen for all five designs. All designs had "short" struts (80% of hull length) supporting a standard cross structure of depth equal to one deck plus structure. The length of the cross structure was equal to that of the struts without any overhang forward or aft. A linear sheer was incorporated into the wet deck (the underside of the box) over the forward 25% of its length. Fig 1 shows a typical bodyplan for a vessel in the study. The resulting designs are the most basic SWATH forms likely to be considered in practice. Their main attribute is simplicity of construction , and the coincidence of LCB/LCF afforded by the short single strut arrangement . This is desirable in reducing coupled heave and pitch motions. The final reason for their selection was to maintain continuity with other work utilizing the same hull forms (Ref 6).

For each displacement , vessels were created with one of two different box clearance values and one of three different compartment lengths. The values of box clearance chosen were selected to correspond with values proposed by Lamb in 1987(Ref 5) for contouring and platforming modes of operation for SWATH vessels. These values form the upper and lower ends of the range of feasible wet deck/waterline clearances. All vessels were idealised to have uniform bulkhead spacing and therefore equal compartment spacing throughout their length. This simplifying assumption , whilst clearly unrealistic was made in order to reveal trends and patterns in the results which might otherwise have remained hidden.

Bulkhead spacings of 6.25% , 8.33% and 12.5% of the vessels length were selected for all designs. These values were selected after careful consideration of current SWATH design subdivision practice. The percentage values chosen gave compartments of length approximating to the upper , lower and intermediate values of compartment length currently considered suitable by contemporary SWATH designers (Ref 4).

The only remaining 'ship' variable considered was operating draught . For each design displacement flooded stability calculations were carried out at three draughts , corresponding to design displacement and design displacement +/- 5%. The resulting range of 10 % design displacement , whilst low for conventional vessels , was considered sufficiently large to cover the operating envelope of most SWATH vessels.

It is recognised that many other parameters have a significant effect on the damaged stability of SWATH vessels. However , it must be appreciated that for reasons of sheer logistics , the number of variables must be kept low since in a study of this kind , each additional variable has a multiplying effect on the size of the study.

Table 1 shows the main particulars of vessels used in the study while Fig 2 illustrates the study plan.

Having determined the variables associated with the ship attention was then focussed on suitable damage scenarios. For each bulkhead spacing , compartments were successively flooded singly and in pairs , fore , aft and amidships , port and starboard around the vessel. It is felt that the resulting ten flooding conditions represent most foreseeable damage conditions which a vessel may reasonably be expected to survive.

The variables initially selected were chosen because they were considered to be the most fundamental. They also provide a sound foundation around which the study can be later expanded to consider the effects of variations in many other parameters.

The main parametric study was thus established with five variables and a final total of some 900 permutations.

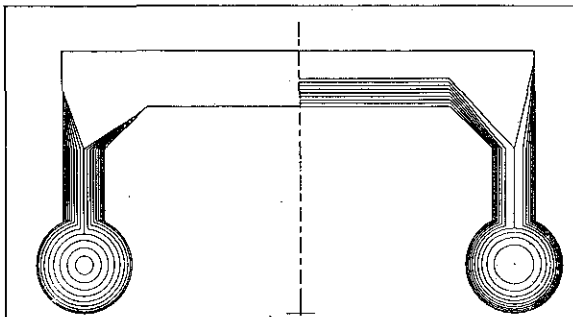


Fig 1. Typical Bodyplan Drawing for a Vessel in the Study

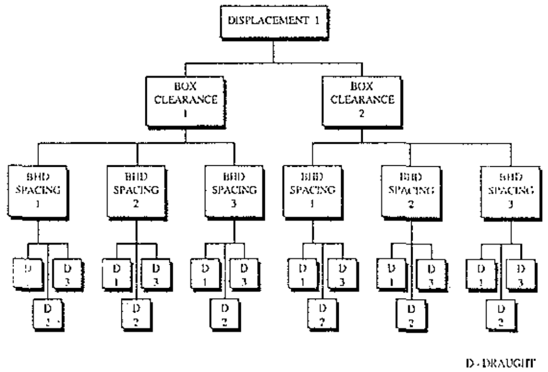
Calculation and Analysis Procedure

Commercially available damage stability software was used to calculate the effect of the ten different flooding scenarios on each of the ninety combinations of vessel design features. The resulting mass of 'raw' damage stability data was processed and analysed exhaustively in an effort to define and isolate suitable indicators of a vessels ability to survive damage.

After careful study of a 'testcase' vessel the five quantities; heel , trim , maximum GZ and area under the GZ curve for the two regions 0-45 degrees and 0-20 degrees were selected to represent a vessels response to flooding. Area under the GZ curve was calculated for two regions after study of the preliminary results indicated contrasting behaviour in the two regions for some flooding conditions.

For every combination of design parameter and flooding scenario , plots of these five values against flooding extent were prepared. The resulting curves were then mathematically defined using regression routines and the polynomial coefficients of the equations thus produced were stored.

FIG 2. Outline for the Parametric Study



Design Variables for Each Displacement

TABLE 1

DESIGN DATA FOR PARAMETRIC STUDY VESSELS

DESIGN DISPLACEMENT tonnes	1000	2000	3000	4000	5000
HULL + STRUT GEOMETRY					
Hull Length	71.747	82	83.558	86.97	89.79
Ellipsoidal Nose Length	21.524	24.6	25.067	26.091	26.934
Paraboloidal Tail Length	17.937	20.5	20.889	21.743	22.445
Hull Diameter	3.116	4	4.897	5.5	6.016
Strut Length	57.398	65.6	66.846	69.58	71.824
Elliptical Nose Length	20.089	22.96	23.396	24.353	25.138
Parabolic Tail Length	20.089	22.96	23.396	24.353	25.138
Strut Thickness	0.917	1.5	1.964	2.399	2.798
Design Draught	4.674	6	7.346	8.25	9.024
Hull Centreline Spacing	21.244	22	23.92	24.4	25.484
BOX CLEARANCE VALUES					
Upper	3.4	4.27	4.91	5.38	5.94
Lower	2.21	2.77	3.17	3.48	3.88
COMPARTMENT LENGTHS					
6.25 % Hull Length	4.49	4.94	5.22	5.44	5.61
8.33 % Hull Length	5.98	6.58	6.96	7.24	7.48
12.5 % Hull Length	8.96	9.68	10.44	10.88	11.22
INTACT GM VALUES	2.124	2.268	2.392	2.44	2.548
VCG VALUES	9.417	10.066	10.863	11.193	11.986

All Dimensions in Metres Unless Otherwise Stated

DISCUSSION OF RESULTS

The test ship selected had a design displacement of 4000 tonnes and a wet deck/waterline clearance of 3.48 m corresponding to the lower bound for a contouring mode of operation. This ship was selected arbitrarily for no other reason than that its combination of geometrical parameters combined to produce a vessel of fairly realistic proportions. Some results from the analysis of this vessel are presented here together with some brief general conclusions on the trends exhibited.

Effect of Increasing Flooding Extent

Effect on GZ curves : For asymmetric flooding the GZ curve is shifted "fwd and down" as expected - Fig 3. However , for symmetrical flooding resulting in trim alone we find that the righting lever GZ opposing forced heeling actually increases with flooding for initial heel angles. This is due to early immersion of the haunch and cross deck structure resulting from the flooding induced trim. The subsequent rise in waterplane area increases stability and hence raises GZ. Above 25 degrees heel this immersion is relatively constant for all cases regardless of initial trim , GZ therefore reduces as expected- Fig 4.

Effect on Heel + Trim : Heel and trim increase almost linearly with flooding extent for all flooding cases. Flooding forward results in values of heel and trim which are slightly higher than those resulting from equivalent damage aft. This is due to the increased volume of the forward compartments and the presence of sheer on the wet deck reducing restoring forces for a given inclination. Fig 5 and 6 illustrate this.

Effect on Max GZ : Increasing the extent of flooding reduces the maximum value of the righting lever GZ possessed by the damaged vessels. This is most noticeable for asymmetric flooding amidships when the reduction is almost linear with increased flooding. When damage occurs towards the ends of the ship the onset of the reduction is delayed.This is due to immersion of the cross structure caused by trim-Fig 7.

Effect on Area under the GZ Curve : The energy required to heel the damaged vessel to a given angle is represented by the area under the GZ curve. This area was found to decrease with increased flooding as anticipated. As for maximum GZ the reduction was again greatest for asymmetric damage amidships , while trim induced immersion of the cross structure delayed the onset of the reduction where damage occurred at the vessels extremities. This immersion was found to be particularly significant for cases involving symmetrical flooding. Indeed it was discovered that the area under the GZ curve in the region 0-20 degrees was actually increased rather than decreased for these cases- Fig 8 and 9.

Effect of Increasing Design Displacement

Heel and trim resulting from damage both increase with increasing vessel size. This phenomenon is due to the volume of flooding and hence the heeling/triming moment increasing at a faster rate than does the restoring moment.Since the vessels in the study were not true geosims the ratio of waterplane area x beam to enclosed volume did not remain constant with increasing size. Simple calculations verify this explanation while Fig 10 illustrates the behaviour observed.

Maximum GZ and area under the GZ curve are both relatively unaffected by changes in design displacement.

Effect of Reducing Box Clearance

Reduction of box clearance results in earlier immersion of the cross structure , this effectively reduces heel and trim for a given flooding condition and raises the area under the GZ curve correspondingly.

For vessels with a low box clearance small amounts of flooding immerse the cross structure resulting in large maximum values of GZ when the vessel is forcibly inclined. For higher box clearance designs these amounts of flooding may not immerse the cross structure , the maximum values of GZ opposing forced heeling are therefore less.

For more extensive flooding the equilibrium angle of heel is sufficient to immerse the cross structure for both high and low box clearance designs. For these cases vessels designed with high clearances ultimately demonstrate the greatest resistance to heeling. The damage extent at which this change occurs reduces with increasing design displacement and the rise in heel associated with increasing design displacement described above.

Whilst many of these results were intuitively anticipated their verification is not without value. Similarly the value of demonstrating probable magnitudes for heel trim etc after flooding should not be underestimated.

Some representative flooded stability results for a 4000 tonnes SWATH vessel are presented in Table 2. The values shown in this table confirm the excellent "survivability" of the SWATH concept while Fig 11 illustrates some damaged waterlines in an effort to demonstrate the physical significance of the values given.

Current US Navy stability and buoyancy criteria for advanced marine vehicles (Ref 7) identify the principal constraints on SWATH survivability to be:

1. maximum initial heel after flooding of not more than 20 degrees ,

2. the main deck edge remains above water at all points.

The vessel analysed was found to satisfy the first criteria for all flooding scenarios studied , however the large trim caused by extreme flooding at the vessels ends was found to slightly immerse the main deck as illustrated in Fig 11. This immersion was however only slight and the results should perhaps be put in perspective by stating that they resulted from an extreme flooding condition :- 25% of the vessels length flooded with 95% permeability . Such a extreme condition is highly unlikely ever to occur in service.

Indeed it appears that the greatest risk to the survivability of a damaged SWATH vessel is damage to structure and superstructure caused by green sea loads. Careful consideration should be given to the possibility of these loads when designing the cross structure and superstructure for SWATH vessels. Other logical priorities for designers in the field include the development of fast counter flooding techniques in an attempt to combat initial heel and trim.

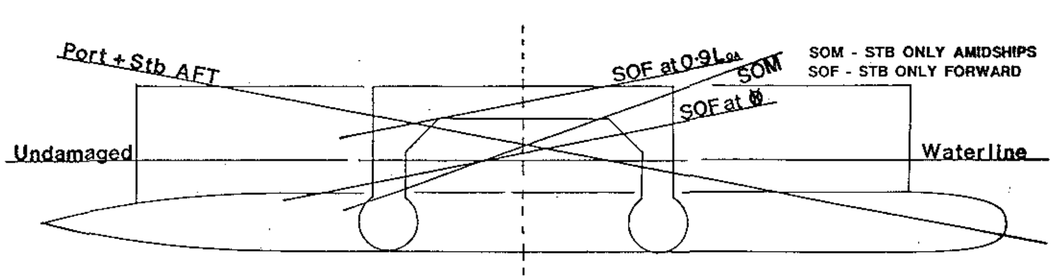
Whilst the above results are valuable , undoubtedly the greatest benefit of the study stems from the provision of a large database of SWATH damage stability information. It is this database which subsequently allowed the construction of the Flooded Stability Estimation Program "FSEP1".

TABLE 2
SAMPLE FLOODED STABILITY RESULTS FOR 4000 tonne VESSEL

HEEL Degrees	Port + Stb Aft	Port + Stb Fwd	Stb Only Aft	Stb Only Fwd	Stb Ammidships
Flooding Extent					
6.25%	0	0	0.665	1.536	3.74
8.33%	0	0	1.37	2.291	[illegible]
12.50%	0	0	3.75	5.344	[illegible]
16.66%	0	0	6.19	7.331	[illegible]
25%	0	0	10.71	11.508	20.309

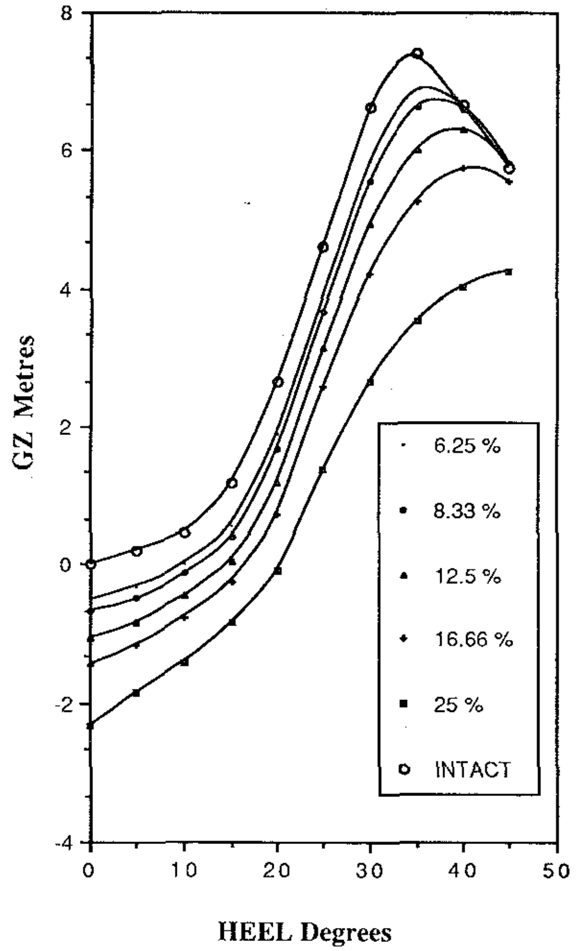
TRIM Degrees	Port + Stb Aft	Port + Stb Fwd	Stb Only Aft	Stb Only Fwd	Stb Ammidships
Flooding Extent					
6.25%	0.946	-2.183	0.475	-1.087	0.049
8.33%	1.873	-3.526	0.936	-1.798	0.012
12.50%	4.527	-6.432	2.485	-3.551	-0.095
16.66%	6.911	-8.481	4.132	-4.899	-0.141
25%	10.826	-12.39	6.17	-7.167	-0.301

MAX GZ Metres	Port + Stb Aft	Port + Stb Fwd	Stb Only Aft	Stb Only Fwd	Stb Ammidships
Flooding Extent					
6.25%	7.421	7.342	7.42	7.342	6.847
8.33%	7.434	7.295	7.434	7.294	6.637
12.50%	7.387	6.944	7.388	6.913	6.313
16.66%	6.86	6.27	6.825	6.199	5.741
25%	4.862	3.808	4.854	4.101	4.25



SOM - STB ONLY AMIDSHIPS
SOF - STB ONLY FORWARD

FIG 11. ILLUSTRATION SHOWING DAMAGED WATERLINES CORRESPONDING TO 25% FLOODING EXTENT AT THE LOCATIONS INDICATED



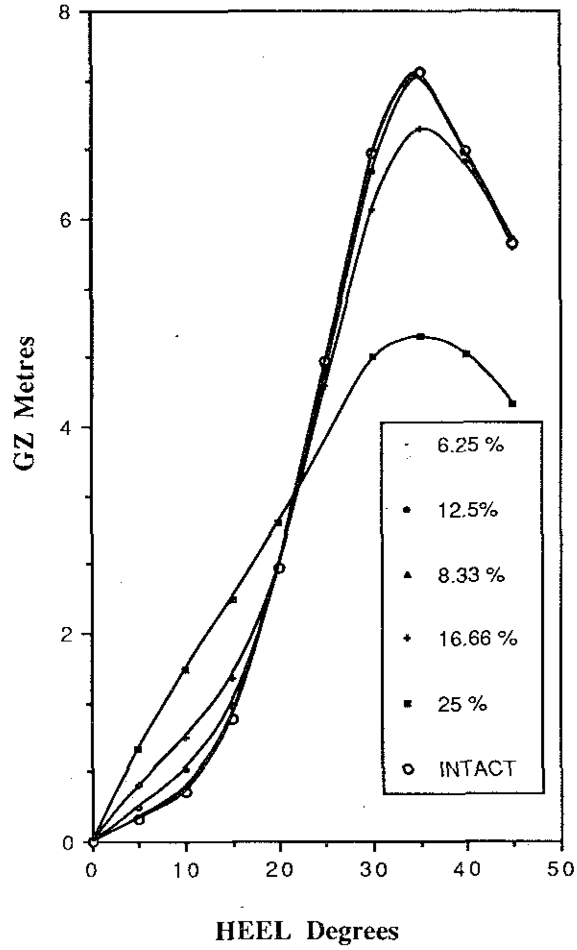


FIG 3. Variation of GZ with Flooding Extent Flooding Stb Ammidships.

FIG 4. Variation of GZ with Flooding Extent Flooding Port and Stb Aft.

FIG 5. HEEL ANGLE / FLOODING EXTENT
4000 Tonne Initial displacement

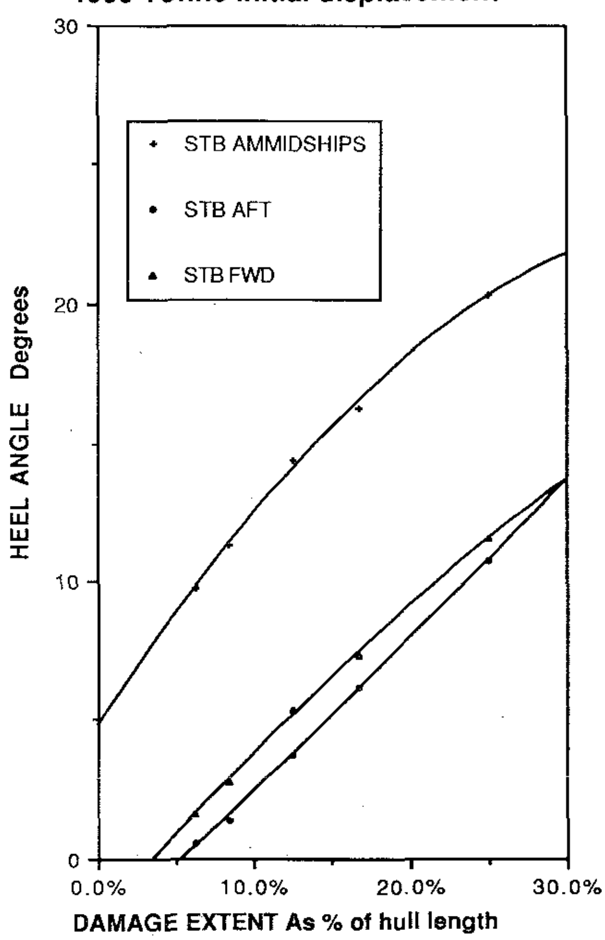
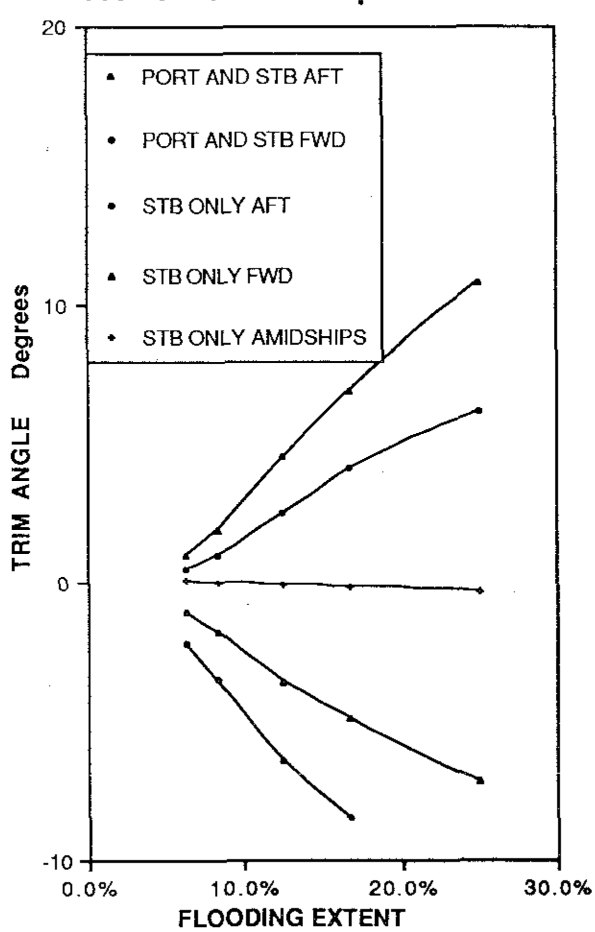


FIG 6. TRIM ANGLE / FLOODING EXTENT
4000 Tonne Initial Displacement



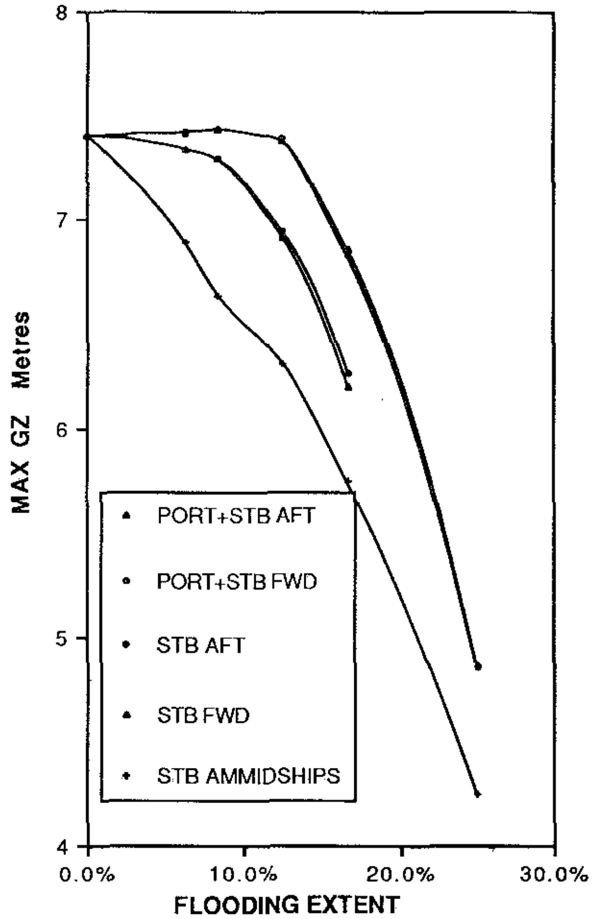
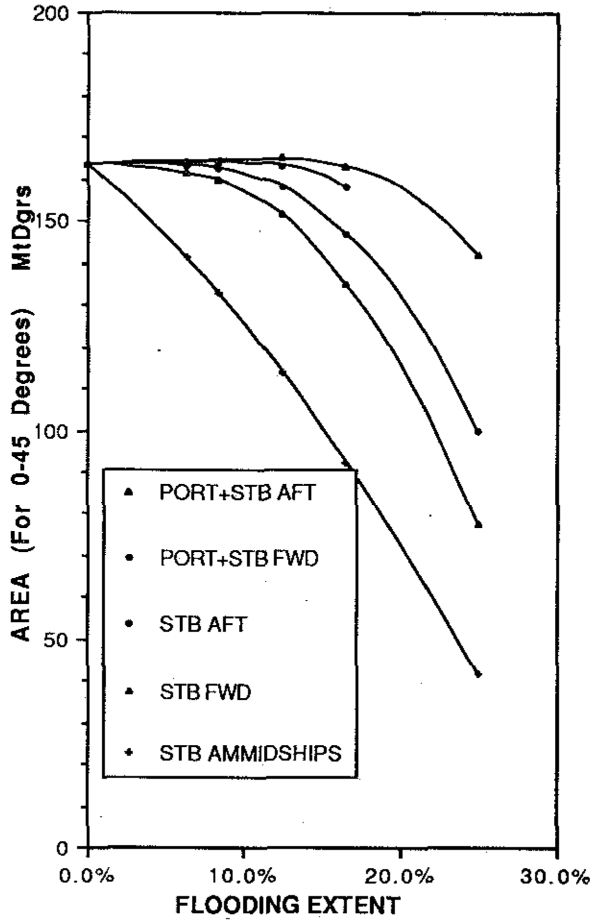


FIG 7. MAXIMUM GZ / FLOODING EXTENT
4000 Tonnes Initial Displacement



AREA UNDER GZ CURVE / FLOODING EXTENT
4000 Tonnes Initial Displacement FIG 8.

AREA UNDER GZ CURVE / FLOODING EXTENT
4000 Tonnes Initial Displacement FIG 9.

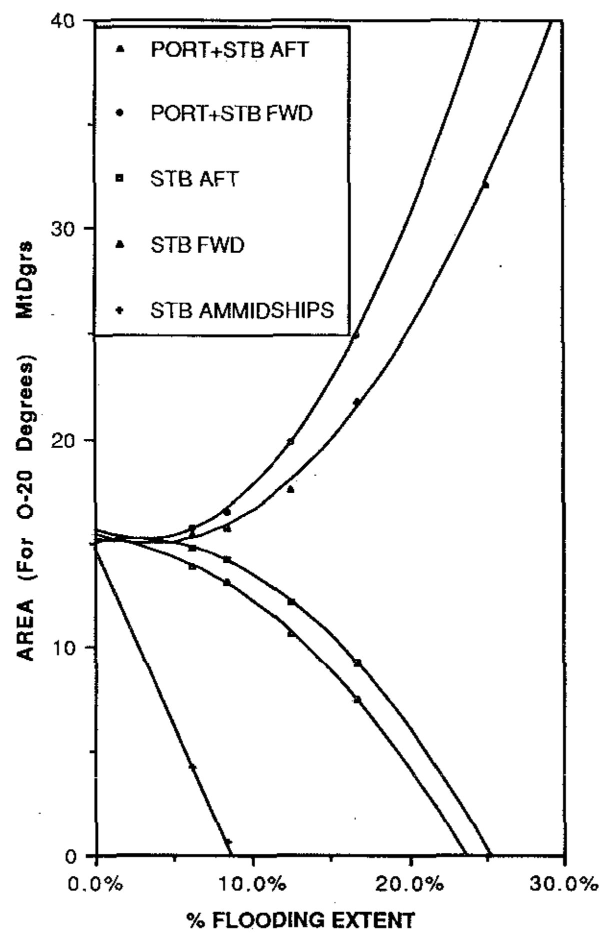
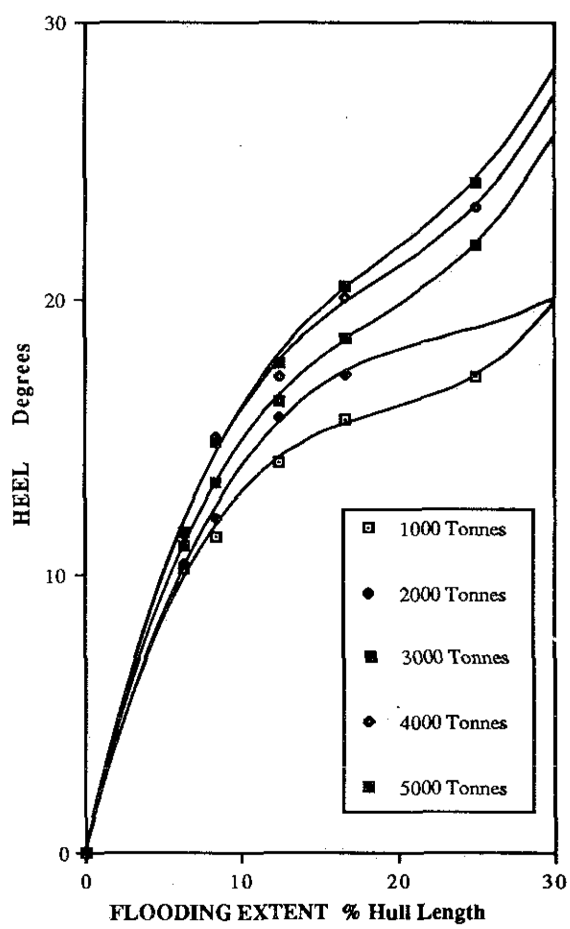


Fig 10. Heel / Flooding Extent
Flooding Stb Amidships



FORMULATION OF 'FSEP1'

"FSEP1" is the first stage of a program which allows estimation of a SWATH vessel's flooded stability at the preliminary design stage. That is only basic geometry details are used in the evaluation.

The program requires the user to input for his design , operating displacement , wet deck/waterline clearance , location and extent of flooding as a percentage of vessel length.

Given this information "FSEP1" will estimate the likely angles of heel and trim after flooding and produce probable values of Max GZ and the area under the GZ curve for the two regions 0-45 and 0-20 degrees.

Essentially "FSEP1" relies on an iterative interpolation technique to produce results. The program searches an extensive database for values bounding the required input condition. Using the polynomial coefficients contained in this database the program calculates values for the bounding conditions and then interpolates between these to find values for the design condition. This process is repeated in a 'nested' fashion until finally output is produced for the required input condition.

Extension of the program to consider additional design information should be readily possible leading to the development of a sophisticated design tool.

VALIDATION OF 'FSEP1'

Results from the program have been checked against actual flooded stability data at three levels.

Level 1 - For the first stage of the validation process flooded stability calculations were performed for ship files which were already defined , that is using designs which were utilised in the construction of the "FSEP1" database. This effectively fixed box clearance and limited operating displacement to within +/- 5% . Flooding extent was of course fully variable within the 0-25 % program range.

Level 2 - The second level of validation again utilised existing ship definitions which this time were uniformly 'distorted' within the computer to give vessels of intermediate displacements whilst still retaining 'family' proportions for the main dimensions. This allowed investigation of larger changes in displacement whilst still retaining a relatively fixed box clearance (either an upper or a lower bound value).

Level 3 - The existing ship definitions were distorted uniformly in the horizontal and longitudinal directions but not in the vertical. The influence of varying box clearance on the accuracy of "FSEP1" 's predictions could then be assessed. At this level it is possible to investigate fairly large changes in all input parameters whilst still remaining loosely within the envelope of "family" proportions.

Some of the results from these comparisons are presented in Table 3.

At the first level of validation the predictions made by the program match closely the values calculated by the commercial software. Maximum errors are of the order of 0.5 degrees for heel and trim and 0.1 metres in the estimation of maximum GZ . Areas under the GZ curve were calculated to within 0.125 MetreRadians (MtRads) in all cases.

Comparison of calculated and estimated values at the second level of validation show similar good agreement. Maximum errors experienced were 1.5 degrees in heel and 0.6 degrees in trim. Maximum GZ was estimated to within 0.45 metres in all cases.

As expected the errors experienced at the third validation level were slightly larger. Maximum errors were however still only of the order of 2 degrees for both heel and trim values whilst the maximum error in predicting MaxGZ was 0.7 metres.

Overall the figures produced are encouraging , however it must be remembered that all three validation levels utilized ships from the same "family" of designs. Once outside the envelope of 'family' proportions it can be expected that the error figures will rise substantially. Despite this it is anticipated that with a little flexibility on the part of the user , the program will give meaningful results for vessels of geometry quite far removed from the 'family' tested here.

Results from the program will almost certainly be sufficiently accurate for use at the intended preliminary design stage .

FUTURE WORK

With "FSEP1", the foundations for a valuable computer aided damage stability estimation tool have been laid. The program although self contained should more properly , be regarded as the first stage in the development of computer assisted damage stability estimation for SWATH vessels.

"FSEP1" has been written in such a way as to be readily expandable to consider variations in many other parameters. Work is already underway to include the effect of changes in the vertical centre of gravity (KG). Further studies to include the effect of beam , strut flare and internal subdivision would all be extremely beneficial adding greatly to the value of "FSEP1".

TABLE 3

SELECTED VALIDATION POINTS FOR "FSEP 1" PROGRAM

DISPLACEMENT	BOX CLEARANCE	DMGE CONDITION	HEEL	TRIM	MAX GZ	GZ 0-45	GZ 0-20	
Tonnes	Metres		Degrees	Degrees	Metres	MtRads	MtRads	
LEVEL 1								
5125	3.88	4% Port + Stb Aft	0	0.298	7.591	2.624	0.194	CALCULATED
			0	-0.004	7.569	2.623	0.188	"FSEP 1"
			0	**0.302**	**0.022**	**0.001**	**0.006**	DIFFERENCE
		4% Stb Only Fwd	0.708	-0.492	7.582	2.606	0.185	CALCULATED
			0.714	-0.485	7.565	2.608	0.185	"FSEP 1"
			-0.006	**-0.01**	**0.017**	**-0.002**	**0**	DIFFERENCE
LEVEL 2								
2600	4.567 ACTUAL 4.27 USED	19% Stb Only Aft	7.357	4.887	6.656	2.482	0.168	CALCULATED
			7.435	4.878	6.857	2.554	0.772	"FSEP 1"
			-0.078	**0.009**	**-0.2**	**-0.072**	**-0.604**	DIFFERENCE
		19% Stb Midships	18.865	0.002	6.297	1.888	0.006	CALCULATED
			17.931	-0.019	6.403	2.007	0.002	"FSEP 1"
			0.934	**0.021**	**-0.11**	**-0.119**	**0.004**	DIFFERENCE
LEVEL 3								
3600	3.967	13% Stb Midships	15.207	0.077	6.648	2.277	0.037	CALCULATED
			15.366	0.033	6.692	2.204	0.039	"FSEP 1"
			-0.159	**0.044**	**-0.04**	**0.073**	**-0.002**	DIFFERENCE
		13% Port+Stb Fwd	0	-6.375	7.112	2.946	0.384	CALCULATED
			0	-6.403	7.105	2.904	0.366	"FSEP 1"
			0	**0.028**	**0.007**	**0.042**	**0.018**	DIFFERENCE

CONCLUSIONS

Through an extensive parametric study , the damage stability characteristics of SWATH vessels have been investigated. Damage stability calculations have been performed for a large number of combinations of initial ship condition and flooding scenario.

The results from these calculations illustrate the dominance of the cross structure effects on SWATH damage stability. Overall the results confirmed what was intuitively expected , initial flooding leads to rapid heeling/triming which eases upon the immersion of the cross structure and the subsequent massive rise in waterplane area and hence stability. From the data collected to date it appears that the SWATH vessel possesses acceptable damaged stability characteristics , and indeed survivability which is likely to be ultimately superior to that of an equivalent monohull. It should be noted that the maximum angle of heel attained by any of the vessels flooded in the study was only just greater than 20 degrees. This corresponded to asymmetric flooding amidships of extent equal to 25 % of the vessels length. Clearly this is an extreme damage condition and one which very few conventional monohull vessels could hope to survive.

The capability to estimate at the preliminary design stage , a vessels ability to survive in the event of it sustaining damage leading to partial flooding , now exists within the 'DESIN' suite of programs. With the creation of "FSEP1" the foundations of a potentially extremely valuable computer aided damage stability estimation tool have been established.

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SYSTEM-CYBERNETIC APPROACH TO THE SHIP'S STABILITY PROBLEM

Janusz T. Stasiak *)

The subject-matter of the paper is a problem of the proper way of ensuring of sea-going ship's stability. It is assumed the solutions which have been used and developed so far are erroneous for many reasons. First of all they do not comply with structure of the man-ship-environment object. They ignore a main role of a man as the operator of the object as well as technic and economic aspects of the ship's operation.
The approach presented here identifying the ship's stability with roll motion safety makes it possible avoiding these inadequacies. It is based on principles of a theory of control system safety and takes fully into account the system-cybernetic propriety of the ship's working. Since the stability is ensured through the decision making process, an importance of information is a distinguishing mark of the approach. Some remark about the role of the administrative and legal infrastructure of marine safety on the correctness of ship's stability solutions are also placed.

1. INTRODUCTION

The situation in the field of ships' stability can be characterized as follows:

1. A great research activity is concentrated upon the problem.
2. Stability is, at least nominally, the object of continuous interest of national and international organizations engaged in the activities of navigational safety.
3. These organizations play a dominant role in formulating and enforcing utilitarian stability solutions.
4. The stability standards being in force are surprisingly primitive.
5. The number of stability accidents is not so remarkable as to intensify special anxiety.

In view of the above mutually contradictory features it is possible to draw two conclusions.
It may be assumed that for practical reasons the problem of ship's stability is trivial and therefore there is no substantial need to revise its existing criteria. To take up and carry out investigations related to stability are only reasonable if intended to satisfy the needs of pure scientific cognition. The second reasoning resolves itself that although stability is a complex problem but simultaneously such one in which:

- the role of existing criteria is very limited or even neglected,
- the effect of other factors, not only hydromechanical, is remarkable and efficiently controled in ship's operation. However, we are not able to describe this effect by a simple and useful mathematical model.

Both the diagnoses can be very embarrassing for the researchers of the ship's stability problem. It is evident that the very important aspect of ship's safety is practically solved without their share. This frustrational situation is aggravated by the fact that one still does not know what strategy should be undertaken in further investigations. The range of propositions is very wide, beginning with polishing of the criteria in use to the search of new ones based on the so-called "rational" premises, and almost all of them have numerous unsolved substantial problems. Thus, up to now such fundamental questions related to stability as: what is the problem, why does it exist, and what are its attributes remain still open.

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2. SOME METHODOLOGICAL REMARKS ABOUT THE PROBLEM

2.1. General remarks

Kotarbinski, the famous Polish praxiologist, characterizing the state of affairs of practical (technical) sciences formulated the following correctness:

"... it is only possible for us to justify completely such a description that we have been unable to realize previously, but we are able to realize such a description that it has been impossible for us to justify in full before" /[1]/.

This rule very well explains the situation on the ship's stability field but first of all indicates that the inherent attribute of engineering is risk. It is the risk that in principle differs the solutions of engineering from the ones of physical or exact sciences.

The physical sciences, whose only purpose is to satisfy the needs of human cognition, aiming at making possibly accurate models, can afford to idealize the problem. In consequence the models are a theoretical description of such a fragment of reality the existence of which in only a priori postulated.

However, the task of engineering is to create artifacts whose proper functioning should be guaranted, not only postulated. Thus, the object of interest of the practical specialities (sciences) must be defined part of reality treated necessarily as a whole, as a system whose structure is made up of both the respective elements and the couplings between the elements and the environment.

However, dealing in general with the practical problems is simultaneously the main reason of the risk (uncertainty) of the created artifact's functioning. In other words, risk is a price we have to pay for the complex treatment of a problem. The necessity creates anti-needs - it is also an engineering correctness.

Designing - the major work of all engineering /[1]/- cannot ignore the necessity of applying system solution to a given problem, and avoid the assessment and minimalization of the risk to follow. Solving the complex and complicated practical problems the system approach and safety investigation are therefore inseparable notions.

One can even say that the system solution of safety is the fundamental task for designing of engineering objects. It is such a task then, when the safety is understood as an accepted compromise of permissible risk and usability of the created object.

2.2. Remarks about ship's stability

The stability of seagoing ship is a very important and practical problem of naval architecture and navigation. Together with buoyancy it is the fundamental ship's propriety which decides about the essence and the usability of this means of transport. At the same time it is a multi-aspect and "badly" organized - diffusion problem. The significance of stability as well as all its complicated nature has one and the same source. It is sea-waving.

Under conditions of calm-water the threat to stability is unnoticeable, there is an univocal measure of stability in form of easy to determine the GZ-curve and above all there are no substantial couplings within the system: environment-ship-cargo-ship's command. However, sea wave integrate the ship into a system whose the most important propriety is just the stability. Sea-wave is simultaneously an essential random input of the system.

For ship's stability understood like this, it is impossible to determine entirely true, and at the same time simple functional relationships occurring in physics as laws.

The search for "rational" models, or better "rational" criteria of stability is a chimera which should by all means be rejected since in this way either it is impossible to find a solution or a problem we deal with is frequently formulated in an elegant form which is rather far from the initial one. "Rational", that is particular approach to stability proper from the cognitive viewpoint creates only illusion to obtain solutions of utilitarian importance. Paraphrazing words of Bishop quoted in [2] one may say that all attempts aimed at finding out solutions to stability based on a physically correct model of appropriate hydrodynamical phenomena, are a waste of time and money.

However, these attempts are definitely non-pragmatic and unpraxeological procedure.

The only right method of solving the ship's stability problem is the system approach. This concept has lately gained more and more supportes which is proved, for instance, by statements at STAB'86 discussion /[3]/ and in Kobylinski's paper [4].

However, the system approach remains a meaningless slogan, if we confine ourselves only to a nominal indication of the necessity to consider various factors responsible for stability problem. It will be another "dead street" if the major stress is put, first of all, on the necessity of identifying "rational" models of par-

ticular elements of the system. The results of great investigation programs (SAFEPROJECT and Norwegian programs e.g) seem to prove entirely the latter thesis.
The success of a system approach heavily depends on the availability and the accuracy of the supporting data. In compliance with the principle at the solution of stability a task more important than concentration on absolute accuracy of the disorganized parts of the problem should be to take into consideration the complete complexity of the even relatively much simplified system elements. In this context it is worth quoting the words of Quade :" .. it is better to be roughly right than exactly wrong" [5].
Taking it into account the stability system solutions maintaining sufficiently correct the physics of the problem ought to:

- pay complete respect to functional structure of seagoing ship, and
- concern also the administrative and legislative aspects of navigation and shipbuilding.

It is, however, impossible to solve properly the substantial problems of stability by adapting them to the arbitrarily determined structures of an appropriate formal system. This question seems to be of significance since so far it has not yet been raised.

3. FEATURES OF A SAFETY QUESTION

Stability is one of the most important aspects of seagoing ship safety. For this reason its solutions should correspond to the rules of the theory of safety. In particular the solutions ought to comply with the principles of safety ensurence of steerable technical objects which, primarily, must be usable i.e. both reliable and economically effective.

The problem of safety appers only then when the character of the phenomenon it refers to is random.
Thus, probability is therefore, a natural measure of safety, although in practice not necessarily the only one. The notion of "safety" is often erroneously identified with a complete certainty of lack of undesired phenomena (e.g. accidents). In fact, however, whenever we talk about safety, we do it because we are not sure about it. The degree of safety of the engineering objects, if they are to exist at all, and are to be useful, is always subject to restrictions ($p_S<1$) which simultaneously means that there is always some risk of an accident ($p_F>0$). This happens regardless of the fact whether real world is determined or random. The point lies, first of all, in our limited perception of the phenomena.

The restrictions can be both objective and subjective- purposeful and conscious. In technical solutions both the factors are important but the latter seems to be even more significant. It results mainly from the fact that technical means must be economically effective. Since we pay both for accidents and safety there is accepted only such a safety level which guarantees, in general, positive economical result. The method of risk factored investment analysis appropriate to such an approach illustrates Fig.1. taken from [6].

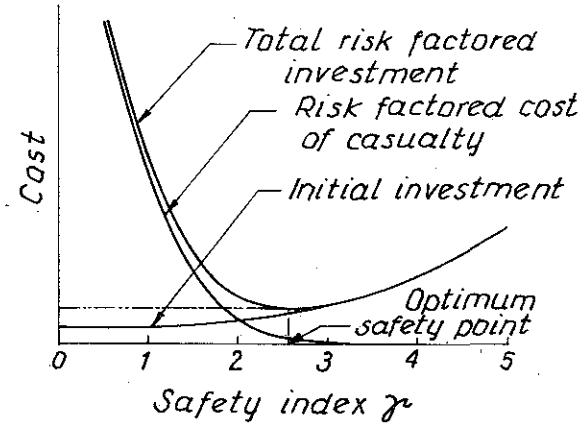

Fig.1. Risk factored investment analysis to determine desirable level of safety.

The subjective randomnes of an object depends on remoteness from which properties of the object we must or want to design. The point is that together with an increase of prognosis distance capacity- **C** as well as demand - **D** of a designed object the variances σ_C^2 and σ_D^2 also rise evidently.

An effect of this phenomenon upon safety shows the idealized*) relationship:

$$p_S \sim \gamma = \frac{\bar{C} - \bar{D}}{(\sigma_C^2+\sigma_D^2)^{1/2}} \tag{1}$$

It is not difficult to notice that the greater the range of the designing prognosis the lasser is the safety or greater is the cost (measured in terms

*) assuming that the random variables **C** and **D** are independent and have normal distributions.

of difference $\bar{C}-\bar{D}$ of the mean values of C and D) that must be born to satisfy the required safety level of the object.

The problem is very important. However, it may turn out that redimensioning of the object and its low usability are not justified. That is an expense paid for subjective and needless complication of the design problem. Thus the range of the designing prognosis should be so limited as possible. In other words each significant feature of designing object that can be determined by short-term prediction way should thus be determined.

The safety states of each system **C-D** are determined by the condition:

$$C > D \qquad \text{(Fig.2a)} \qquad (2)$$

If **f(C,D)** is the joint density distribution of capacity and demand, then the maximum probability of the safety states of system **C-D** is:

$$p_S=P(C > D)=\iint\limits_{C>D} f(C,D)dCdD \qquad (3)$$

The attained in practice safety level depends on the adaptability of a system, i.e. on the system capabilities to monitoring and control (regulation) of the **C** and **D** proprieties:

a) if the system has unlimited adaptability, which means that condition (2) can always be fulfilled, it will function with maximum usability. The global safety level of the system can then be equal to value p_S described by formula (3);

b) if the system adaptability is limited, the safe functioning of system is confined to states determined by condition:

$$\left.\begin{aligned} &C > C_N = Q \\ \text{and}\quad &D < D_N = Q \qquad \text{(Fig.2b)} \end{aligned}\right. \qquad (4)$$

and thereby the usability of the system is limited. The degree of the guaranteed safety of the system is now:

$$p_{SQ}=P(C>Q \wedge D<Q)= \iint\limits_{\substack{C>Q\\ D<Q}} f(C,D)dCdD \qquad (5)$$

which means that there is a global margin of safety equal to p_S-p_{SQ};

c) if limited monitoring and control are possible only with regard to propriety **C**, then it is admissible for the system to function in states:

$$C > C_p \qquad \text{(Fig.2c)} \qquad (6)$$

With such a condition, the system can nominally function both in states for which there exists a margin of safety, and in states really dangerous. Probability of failure of the nominally safe system is:

$$p_{Fp}=P(C>C_p \wedge D>C)= \iint\limits_{\substack{C>C_p\\ D>C}} f(C,D)dCdD \qquad (7)$$

while the global margin of safety is:

$$\Delta p_{Sp}=P(C<C_p \wedge C>D)= \iint\limits_{\substack{C<C_p\\ D<C}} f(C,D)dCdD \qquad (8)$$

Significant in cases b) and c) the boundary values $Q = C_N = D_N$ and C_p must be determined by long-term prognosis.

It is evident that a need or even necessity of a long-term safety prediction is the bigger the smaller is the system adaptability, and vice versa.

Thus, a sine qua non condition to obtain both the proper safety solution and optimum usability is a complete recognition of the adaptability of a specified system.

In view of the above remarks there seem to arise the following suggestions:

- the methods of safety solutions should not be " automatically" transferred from one branch of engineering activity to another;

- various approaches can be required for different aspects of safety even in the case of the same technical object.

In particular, to solve the ship stability problem it is impossible to apply either methods related to safety of stationary objects of civil and marine engineering, or methods appropriate to ship structure reliability or to ship subdivision. In other words, the problem of ship's stability can be solved neither by the GZ-curve criteria nor by probability criteria determined according to the concepts of Firsov /[7]/ or Krappinger [8] e.g.

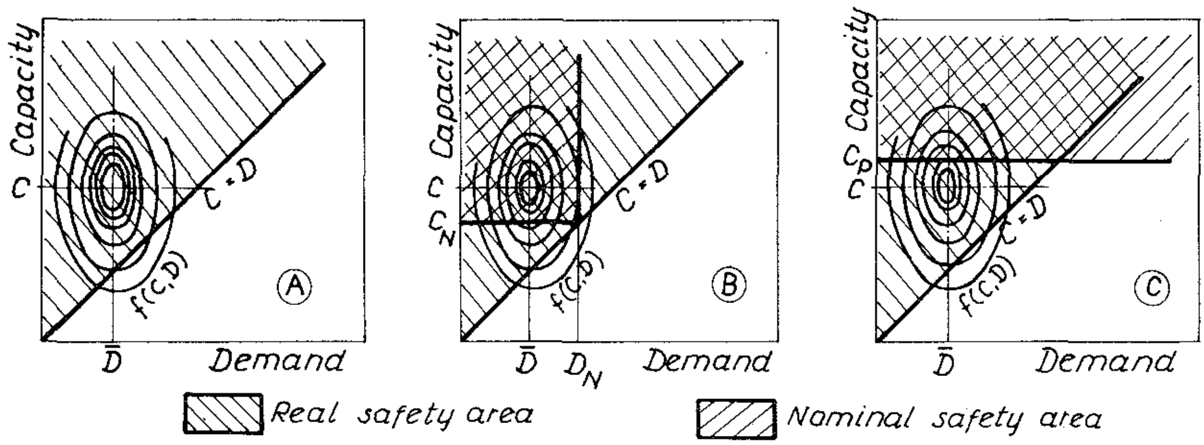

Fig.2. Real and nominal safety states of C–D system.

4. SHIP'S STABILITY SYSTEM

4.1. Definition of ship's stability

From hydromechanical viewpoint, ship's stability is identified with ship's safety against capsizing. At the same time the capsizing or loss of stability is understood as bringing the ship to the upside-down position. A more proper definition, especially when taking into account practical purposes, seems to be the one which treates the loss of ship's stability "... as exceeding the angle of roll at which situation dangerous to the ship occurs which makes further ship's operation impossible" /[9] [2]/.
This definition includes not only the physical capsizing but also any excessive heeling that leads to flooding, shifting of cargo, etc. These phenomena themselves are marine accidents and, what more, can become a direct cause of capsizing.

Taking the above into consideration, stability can be defined as the ship's capability of not exceeding certain amplitudes of roll angles.
In other words, a ship will be stable if the roll amplitudes ϕ_A satisfy the inequality:

$$\phi_A < \phi_{max} = \phi_i \qquad (9)$$

where:

$\phi_i = \min\{\phi_C, \phi_L, \phi_F, \dots\}$,
ϕ_L – the angle determining a boundary of the effectiveness of cargo securing,
ϕ_F – the angle of ship's flooding,
$\phi_C \approx \phi_V$ – the angle of capsizing.

Such a definition identifying the ship's stability with roll motion safety has, for ensuring the safety, a very useful meaning. First of all, it is a complex solution which involves most of the significant threats to stability, and, at the same time, it enables a correct and effective compatibility of reliable functioning of the ship with its profitability. It is trefore a solution par excellence practical. It is also a pragmatic solution. The roll angles are such proprieties of ship which are simple to be monitored as well as controled. The determination of the criteria O_i is equally simple.
Finally it is a natural solution; it harmonizes completely with the natural structure of ship's control.

4.2. Ship's roll safety control process

A scheme of the process of roll control is illustrated in Fig.3.

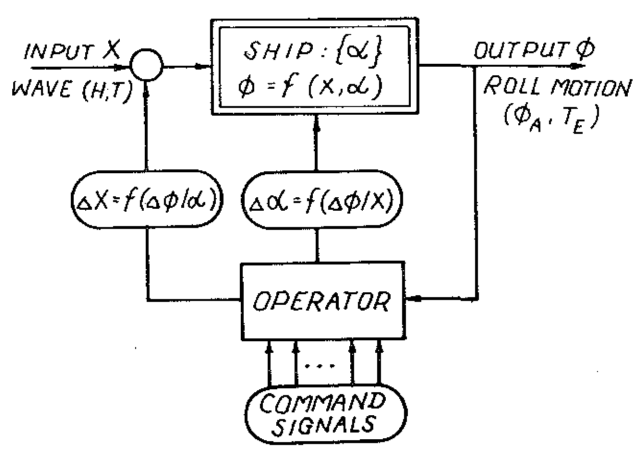

Fig.3. Flow diagram of the cybernetic system of the ship roll control.

One can see, the roll control process takes place as a monitored control system. The essential elements of this system are as follows:

- the main path in which the wave input $X(H,T_E)$ (H,T_E- wave height, and wave encounter period respectively) is transmitted on the ship roll motion $\phi(\phi_A,T)$ (ϕ_A,T- amplitude and period of roll) in accordance with a real mechanism of regulation:

$$\phi = f(X,\alpha) \qquad (10)$$

where:

α = {L, B, d, GZ, V, ...} denotes a vector of ship properties;

- the feedback path in which the ship properties α as well as the input signal X are steered in order to satisfy the safety condition [ineq.(9)]. An effectiveness of such steering is determined by an attainability of the steering rules i.e. the models of inverse, in respect to (10), functions:

$$X = f(\phi/\alpha) \quad \text{and} \quad \alpha = f(\phi/X) \qquad (11)$$

- the command signals which supports an operator of the steering process. This is a set of information regarding the conditions as well as the possibilities of the steering. Particulary, the command signals can include information about:

    - the safety criteria i.e.: angles of ϕ_C, ϕ_L, ϕ_F, etc.,
    - a range of the possible input values X,
    - a range of the admissible modification of ship properties α,
    - an essential requirements about navigation or trade-service,
    - anothers individual factors which have an essential meaning for proper roll control.

The ship's roll control is a hierarchic decision making process in which the following three levels can be described:

- a strategic level which is nothing else than ship's designing. Here the long-term steering of the ship properties denotes such a choice of the body shape for which the roll motion is a minimum for all-live ship service conditions;
- a tactical level in which the following activity are realized:
    - a stowage and cargo securing and thus determining such ship properties as the draught and the GZ-curve as well as safety criteria,
    - an initial choice of ship weather routeing based on the forecast.

One should notice that for proper safety control the both activities should be coupled;
- a navigation level in which the steering of the ship properties relies on choosing the ship velocity i.e. choosing the ship advanced speed as well as the heading angle for a given seaway.

4.3. Features of the ship's stability system

Basing on the above characteristics it is easy to notice that the ship's stability identyfied with the roll motion safety is entirely a system-cybernetic problem. Man - both the ship's designer and the master - acts as a homeostat, that is to say as a searcher for the optimum ship's safety at each service conditions. The significance of his decision rises from level to level, and therefore it may be assumed, that the ship with regard to stability is a subject to continual designing; it is also designed during its exploitation.
The design process of the ship's stability consists of:

- initial, statical determination of ship's properties, and

- dynamical adaptation of these properties to the current environmental conditions.

At the tactic level both the elements appear together. At the strategic level the ship is subject to exclusive statical designing, whereas at the navigation level the dynamical adaptation dominates. For this reason there is a possibility for a passage from a coarse stability control (as described ad c-item of chapter 3) at the strategic level to a fine one (a-item of chapter 3) at the navigation level. Increasing from level to level precision of identification of ship and environment properties is conducive to this possibility. However, the possibility can essentialy be reduced when the master knowledge about stability control mechanizms as well as a bulit-in ship's stability are too low, and it is rather a typical situation. This results both from too poor seakeeping characteristics of modern energy-saving ships and from rapid progress of shipping technology these days; there is no possibility for seamanship settlement. (It is rather usual that seakeeping is not a object of ship designing).

Thus, for the ship adaptative process to be most effective at the strategic level it is necessary to work out:

- a body shape having as good sea-keeping characteristics as possible,
- an appropriate for this ship stability *manual containing, first* of all, the dependences of type (10) or (11) and, moreover, some information about the proper way of identifying all essential command signals.

On can say, that the essence of the system-cybernetic approach to ship's stability is based on equivalent treatment of the hardware ship solution,which suits to bulit-in ship's stability, and on the software solution, which determines the operation stability i.e. the ship's adaptability.

The final form of subject-matter of the stability manual requires further work. Anyway, two following questions should necessarily be taken into consideration: cargo stability problem and relationship between ship's stability and ship's velocity. Cargo shifting has been the cause of many recent capsizing accidents and, moreover, the problems of cargo stability and ship's stability are substantially coupled; they must be solved jointly. The sea-wave, in comparison to the calm-water condition, is ananisotropic environment in respect to the ship proprieties. Hence, the adaptability of ship is, first of all, a question of the appropriate choice of ship velocity.

Thus, the system approach to the ship's stability problem is a natural and pragmatic solution. It takes into account all the fundamental principles of seagoing ship functioning, that is those which result both from hydromechanical laws and the shipping technology requirements. By inclusion within the stability problem such aspects of the ship's working as flooding and above all the cargo stability the ship's service can in fact be safe as well as profitable.Finally, in system approach framework many serious obstacles relating to pure mathematical solutions of the stability problem can be overcome, and, at same time, the existing seakeeping methods can be better utilizated. Taking above into consideration one can to state that there is no option about the system approach to the ship's stability problem. " It has to be used. It is a necessity !" - as Jenkins pointed /[10]/.

5. A ROLE OF AN ADMINISTRATIVE AND LEGAL INFRASTRUCTURE OF SHIPBUILDING AND NAVIGATION

Designing, treated as a discipline in the field of practical sciences, is intended to constantly raise the instrumental rationality of human activities to attain the more and more effective goals. The more rationality is appreciated, and socially important are the functions of the object of designing, the greater is the chance to institutionalize the design process.

The institutionalization of the design and operation process of large and complex engineering systems is a necessity. However, this necessity can easily change into an anti-need, and become a source of artifical tensions and difficulties. The point is that every formal system, and administrative one in particular, is inclined to generate its own needs which, in general, do not agree with the real purposes of the designing object. Unfortunately, the administrative and legal solutions of shipbuilding and navigation are not free from this phenomenon, and also due to the ship's stability solutions are not the best.

Functioning of the "measured course" institution is a distinguish mark in this question. Ship owners normally specify the requirements for calm-water performance of the ship to be bulit such as the maximum cruising speed and maneuvering capabilities be achieved. Hence, the seakeeping characteristics are neglected during designing although these proprieties directly afect the safety of ship and cargo as well as the real ship's speed. Finally, the ship designed for ensuring of the measured course conditions can be neither safe nor profitable.

The necessity for institutionalizing the ship's and navigation safety solutions results mainly from the fear that functioning of the ship, according to the particular interests of its owner, can disagree with the major social reasons. In particular, the point here is to give a guarantee to protect human life and the natural environment, to secure the interests of third parties.

There is obvious need for international ship's safety rules but simultaneously there is ineffective democratic way of elaborating and passing these rules. Equality of rights in the establishement of ship's safety rules indicates nearly always a check of development in merits of this questions.

Moreover, the IMO activities and Conventions' effect weakens significantly

the activities of national maritime administrations in the field of elaboration and improving the standard of their own safety regulations.
The habit of waiting for the appearance of international safety regulations is far too common. What more, those countries which were busily engaged in preparing new international rules, do not introduce these regulations to their codes even after they have been agreed upon at an appropriate Convention.

The fact that certain maritime administrations transfer their authorities in ship's safety to classification societies seems to be rather disadvantageous in question of a regulation correctness. This fact, makes, among other things, it possible that there is still such a great "adherence" to the designing stability criteria and particular, to GZ-curve criteria. Due to this reason, the on board stability information in current use is of insignificant exploitational applicability - its subject-matter provides, first of all, a classification character.
The commercial aims of the classification societies have also some effect upon the quality of their stability requirements. To tell the truth, the adequacy of these requirements is to a great extent subordinated by fiscal interests of the classification societies, and in fact the ship's safety is here at most a secondary stimulator.

The purpose of the above remarks is to give some hints that in process of modifying the quality of the stability solutions the problem of rationalizing the administrative and legal infrastructure of shipbuilding and navigation is also of primary importance. It is hard to image that the infrastructure can be hic et nunc subject to variations, but one should not assume that its present solutions are indefensible and that substantial solutions of ship stability should submited to them.

6. CONCLUSIONS

"Primum not nocere"- it is well known, the fundamental precept of medicine formulated by Hipocrates. Unfortunately, at the ship's stability question, this precept remain still to be satisfied. There still dominates artifical and unnecessarly tendency to complicate the problem. However, natural solutions accounting inherent ship's adaptability have been not in favour so far. The system-cybernetic approach presented here emphasizes this. Its main points are as follows:

1. A ship is too complicated engineering object to describe its features, inclusive stability, in sense and in the way as it is done in physical sciences.
The so-called "rational" criteria of stability are an illusion that is to be rejected. It is not intended, either, to search for designing criteria of stability the accomplishment of which will provide, in fact, a global (for all exploitational conditions) ship's stability. However, it is of primary importance that the ship should be designed in view of the optimum seakeeping proprieties insted in view of the calm-water resistance and propulsive proprieties.

2. Exploitation is a further step of ship's designing process, and the decision undertaken there, are of great priority for real stability. On the base of the available knowledge, in particular on the seakeeping knowledge, it is necessary to work out a useful, for the master, and a specific one for a particular ship, stability manual.

3. Attemps should be made to estabilish more correct administrative and legislative solutions referring to ship's stability.
The problem concerns mainly the elimination of these bureaucratic mechanizms which are responsible for the essential arguments to be "pushed aside" in competition with particular interests of respective institutions. The point is also to eliminate the mechanizms which stimulate the psychological resistance phenomenon against all innovations in the sphere of the ship's safety.

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AN EXPERIMENTAL INVESTIGATION
INTO THE STABILITY AND MOTIONS
OF A DAMAGED SWATH MODEL

Bruce C. Nehrling[1]

ABSTRACT

The objective of this experimental work was to observe and partially quantify the stability and seakeeping characteristics of a dead-in-the-water scale model of a SWATH (Small WAterplane Twin Hull) both before and after simulated flooding had occurred.

First, a conventional inclining experiment was conducted in order to establish the model's intact displacement and center of gravity. In addition, the model was suspended in air and swung as a compound pendulum in order to determine its mass pitch and roll gyradii.

Second, an intact and two damaged conditions were modeled. In each condition, the untethered model was repeatedly subjected to both moderate and severe, irregular, long crested seas. In each sea state the model was positioned to experience head seas, following seas and beam seas. Roll and pitch motions were measured with a gyroscope.

This SWATH, even though dead-in-the-water, had sufficient stability to survive the specific intact and damaged conditions and sea states which were modeled.

INTRODUCTION

The purpose of this experimental work was to observe and partially quantify the stability and seakeeping characteristics of a dead-in-the-water scale model of a SWATH (Small WAterplane Twin Hull) both before and after simulated flooding had occurred [1].

MODEL DESCRIPTION

A scale model of a SWATH hull, which had previously been used for a series of seakeeping experiments and structural loading studies, was used for these stability experiments. The geometric characteristics of this model are listed in Table 1. This fiberglass, aluminum, and wood model was outfitted with a pair of dihedral canards, a set of dihedral rudders, and all around propeller protection rings. There were no propellers. The canards and rudders were locked in place. The model was not self-propelled nor were there any active control surfaces. Figure 1 is a schematic drawing of this model. For the intact stability experiments, the model was ballasted to the displacement, center of gravity location, and gyradii shown in Table 1.

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MODEL PREPARATION

In order to confirm the ballasted model's displacement it was weighed in air from a load cell. The model was ballasted to the required center of gravity by repositioning small onboard weights while the hull was suspended in air on a "knife" edge. For the longitudinal center of gravity (LCG) and the transverse center of gravity(TCG) this "knife" edge was an inverted angle iron properly positioned under a pair of longitudinal aluminum tubes located in the model's cross structure. To obtain the vertical center of gravity (VCG), reinforced "L" shaped brackets, which had been counter balanced to have the required centroid, were attached at the bow and stern to aluminum blocks which extended through the corners of the deck. By using these brackets, the model's VCG could be set to the required height while it was hanging sideways on the "knife" edge of a heavy piece of inverted angle iron.

A conventional inclining experiment was then conducted to confirm both the expected draft and the position of the vertical center of gravity.

Once the model was ballasted to the desired displacement and center of gravity it was suspended in air and swung as a compound pendulum in order to determine its mass pitch and roll gyradii. In a trial and error process, onboard weights were

TABLE 1

MODEL
GEOMETRY AND MASS
CHARACTERISTICS

INTACT CONDITION

LOA	3.220	m
LBP	2.632	m
Beam, max @ DWL	1.108	m
Beam, extreme	1.295	m
Freeboard @ amidships	0.360	m
DWL draft @ amidships	0.343	m
Trim	0	deg
Heel	0	deg
DWL area	0.526	m^2
KB	0.141	m
LCB	0.044	m
LCF	0.042	m
Displacement @ DWL	322.1	kg
KG	0.418	m
LCG	0.044	m
Transverse GM	0.124	m
Longitudinal GM	0.366	m
Roll gyradius	0.510	m
Pitch gyradius	0.840	m
Natural roll period	3.93	sec
Natural pitch period	3.58	sec
Natural heave period	2.14	sec

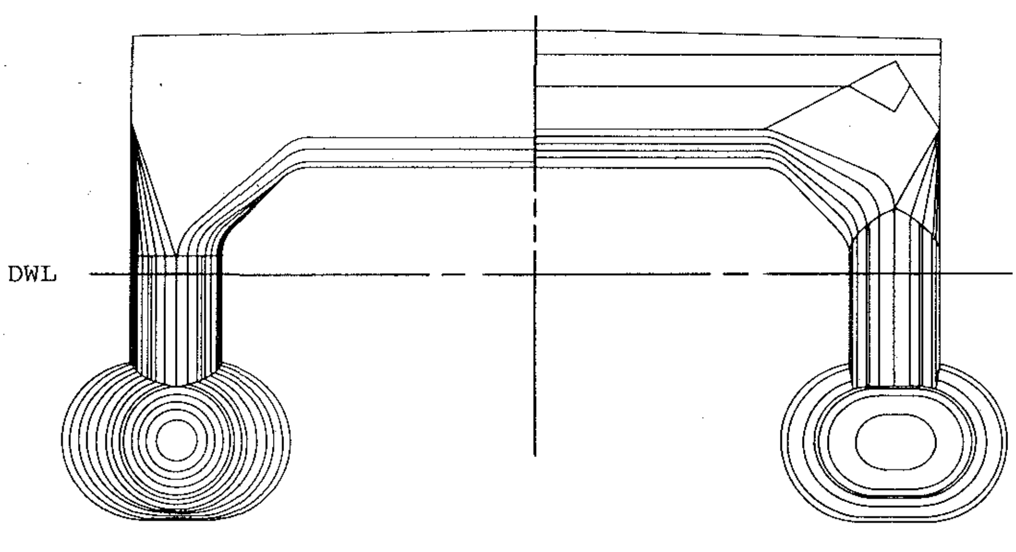
NOTE: Longitudinal dimensions are from amidships (+ forward); vertical dimensions are from the baseline (+ upward); displacement conversions are based on the model being in fresh water at 20.6°C.

shifted in order to obtain the desired values for the gyradii. These weight shifts were done in such a way that the center of gravity did not change. That is, equivalent weights were moved symmetrically fore and aft to alter the pitch gyradius and inward and outward to modify the roll gyradius. To swing the model, an "A" frame assembly was built which would allow the model to be suspended from a forklift. Prior to attaching the model the "A" frame was weighed and its gyradius determined by swinging it and measuring the period of oscillation. With the gyradius and the mass known, the "A" frame's mass moment of inertia was computed. After the model was connected to the "A" frame this process was repeated for the composite system. The parallel axis theorem was then used to subtract out the effect of the "A" frame and thus derive the model's pitch moment of inertia about its own center of gravity. The model's pitch gyradius was then calculated. The roll gyradius was found by rotating the "A" frame assembly 90° and then swinging the composite system. As before, the effect of the "A" frame was then computed out in order to obtain the model's roll moment of inertia. Finally, the model's roll gyradius was calculated.

Two conditions of damage were modeled. The first condition simulated flooding of the starboard strut and demi-hull near the stern. This "flooding" of the model was

FIGURE 1

HULL FORM SCHEMATIC



accomplished by adding fixed ballast until the model had the required overall displacement and center of gravity. After this "flooding" the model had a measured trim of 5.2° by the stern and a list of 9.15° to starboard. Table 2 contains a more complete description of the model in this damaged condition.

TABLE 2

MODEL CHARACTERISTICS

DAMAGE IS STARBOARD AND AFT

Change in displacement	23.85	kg
LCG of damaged water	- 0.805	m
KG of damaged water	0.270	m
TCG of damaged water	0.485	m
Displacement (Damaged)	345.9	kg
LCG (Damaged)	- 0.014	m
KG (Damaged)	0.407	m
TCG (Damaged)	0.033	m
Damaged Trim (by stern)	5.20	deg
Damaged Heel (to strbd)	9.15	deg

NOTE: Longitudinal dimensions are from amidships (+ forward); vertical dimensions are from the baseline (+ upward); transverse dimensions are from the centerline (+ strbd); displacement conversions are based on the model being in fresh water at 20.6°C.

The second damage condition simulated flooding of the starboard strut and demi-hull toward the bow. As a result of this "flooding," the model developed a trim of 3.0° by the bow and a list of 11.0° to starboard. This second damaged condition is summarized in Table 3.

TABLE 3

MODEL CHARACTERISTICS

DAMAGE IS STARBOARD AND FORWARD

Change in displacement	25.00	kg
LCG of damaged water	0.418	m
KG of damaged water	0.229	m
TCG of damaged water	0.489	m
Displacement (Damaged)	347.1	kg
LCG (Damaged)	0.071	m
KG (Damaged)	0.404	m
TCG (Damaged)	0.035	m
Damaged Trim (by bow)	3.00	deg
Damaged Heel (to strbd)	11.00	deg

NOTE: Longitudinal dimensions are from amidships (+ forward); vertical dimensions are from the baseline (+ upward); transverse dimensions are from the centerline (+ strbd); displacement conversions are based on the model being in fresh water at 20.6°C.

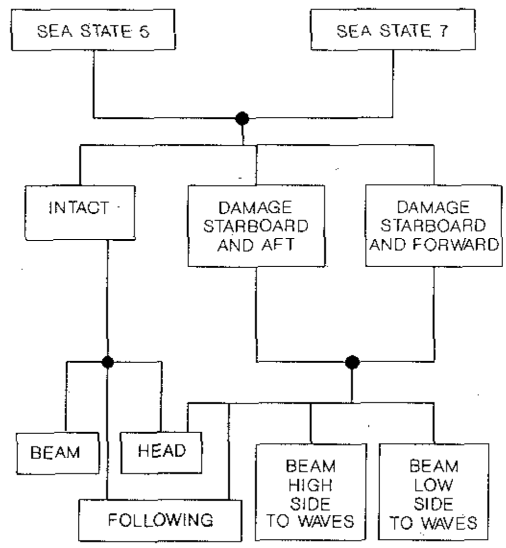
It should be noted that these two conditions of damage were modeled as individual events and not as simultaneous or sequential occurrences. Also, the model's mass pitch and roll gyradii were not recalculated for either of the damaged conditions. Furthermore, since solid weights were used to represent the flooding water, the detrimental effect that any free surfaces would have on the model's overall stability was not measured.

EXPERIMENTAL PROCEDURES

Sea Conditions

The SWATH model was tested untethered, in two different irregular, long crested seaways, at zero forward speed, in three stability conditions, and in four orientations relative to the propagating wave. This test matrix is delineated in Figure 2.

FIGURE 2 TEST MATRIX



Note: High side means that the model's high freeboard side (port side) was facing the direction from which the waves propagated. Low side means that the model's low freeboard side (starboard side) was facing the direction from which the waves propagated.

The waves used in this experimental program were generated by a servo-electro-hydraulically driven dual flap wavemaker acting under computer control. These experiments were conducted in the United States Naval Academy's Hydromechanics Laboratory's 116 meter long towing tank. A description of this facility is given in reference [2].

The wave energy spectra for the two seaways were based on the modified ITTC spectral formulation which is defined by significant wave height and modal period. Significant wave heights and modal periods for the two seaways were obtained from the summary of North Atlantic wave buoy data presented in [3]. They are representative of fully developed seaways in the open ocean.

Model scale sea state 5, which had a significant wave height of 0.146 meters and a modal period of 2.08 seconds, was considered to be representative of a moderate operational environment. Model scale sea state 7, which had a significant wave height of 0.354 meters and a modal period of 3.23 seconds, was selected since it would represent a severe operating condition for this vessel. These two repeatable sea states are summarized in Table 4.

TABLE 4

SEA STATE SUMMARY

MODEL SCALE

Sea State	5	
Modal frequency	0.48	
Modal period	2.08	sec
$H_{1/3}$ #	0.146	m
$H_{1/10}$ #	0.186	m
Highest wave *	0.229	m
Sea State	7	
Modal frequency	0.31	
Modal period	3.23	sec
$H_{1/3}$ #	0.354	m
$H_{1/10}$ #	0.448	m
Highest wave *	0.536	m

Significant peak-to-trough wave heights are based on an assumed Rayleigh distribution of the wave's energy spectrum.

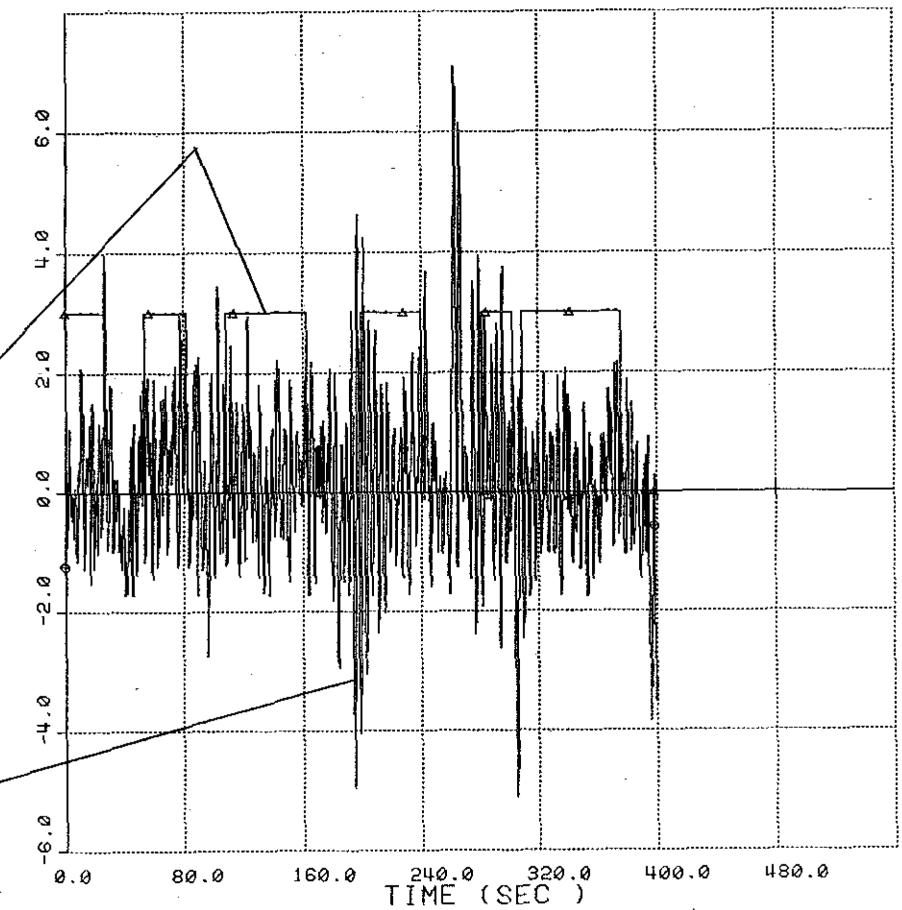
* Greatest wave height which was observed during the generation of this wave sample.

<u>Testing</u>

As mentioned previously, the model was positioned in the desired orientation and then, with the handling lines slack, it was allowed to be buffeted by the waves. Since the model was free to drift it did not always encounter the same wave at the same time in the 400 seconds of data acquired per test. Periodically the model had to be man-handled back to its original orientation and location in the towing tank. A binary switch was used to identify within the computer records those time periods during which this repositioning was taking place. Figure 3 shows a typical plot of the raw data. In this example, the rolling time history for the intact model in moderate

FIGURE 3

TYPICAL PLOT OF RAW DATA



seas is plotted as a function of time. The step function plotted in this figure represents the status of the binary switch (on or off). When the switch was on the model was unconstrained. When the model had to be repositioned this switch was set to off. The status of this switch enabled the erroneous data which were gathered during those times that the model was being repositioned to be identified and deleted from any subsequent analysis.

Roll and pitch motions were measured relative to an initial condition of zero heel and zero trim with a dual axis electrically powered gyroscope which was mounted on the weather deck. The cable for the gyroscope was suspended from an overhead hoist which was moved as needed to remain over the model at all times. The Hydromechanics Laboratory's computer system was used to gather, store, edit and analyze these roll and pitch time histories.

ANALYSIS OF EXPERIMENTAL DATA

In the analysis of the acquired data the H 1/3 and H 1/10 significant peak-to-trough roll and pitch amplitudes were calculated assuming a Rayleigh distribution of these events. Thus the significant double amplitude values were obtained by multiplying the standard deviation about the sample's mean by 4.0 for the H 1/3 values and by 5.1 for the H 1/10 values. During most of these tests the model's heaving, pitch and roll motions produced a wave system which eventually began to reflect off of the towing tank's walls. Consequently, after a period of time, the irregular wave which the model experienced was somewhat different from the wave which was being generated by the wavemaker. While heave was not measured, its magnitude and frequency were, at times, quite noticeable. A SWATH hull form appears to provide very little heave damping. In addition, the heave which did develop quickly became coupled with the other motions - particularly roll and pitch. During this limited number of tests, the model, whether intact or damaged, never exhibited any signs of becoming unstable. Finally, given the box like profile of this SWATH, it is important to realize that the model was not being simultaneously subjected to any wind loads.

Intact Condition

The intact model's H 1/3 and H 1/10 significant peak-to-trough roll and pitch motions, as a function of two sea states and four orientations, are shown in Figures 4 and 5 respectively. The intact model would tend to hold its position in head seas, surge with the waves in following seas, and, while rolling noticeably, move laterally and quickly across the tank in beam seas. While in this orientation, the model's largest angle of roll, which occurred during sea state 7, was 18.4° to starboard. Solid water never swept over the weather deck.

Damage Starboard and Aft

Figures 6 and 7 show the model's H 1/3 and H 1/10 significant peak-to-trough roll and pitch motions, as a function of two sea states and four headings, after it had been ballasted to simulate flooding to the aft starboard side. The damaged model, though pitching, would tend to maintain its direction and position when subjected to head seas. In following seas it would maintain its heading but pitch and surge as it drifted with the waves. The largest observed single amplitude pitch angle was 11.6° and occurred during sea state 7 while the model was encountering head seas. When starting out in beam seas, with either the starboard (or low) side facing the waves or with the port (or high) side facing the waves, the model would tend to pivot about the damaged "corner" and then, with the bow turning away from the waves, begin to move diagonally and quickly across the tank. During these limited tests, solid water never swept over the weather deck. However, rolling, wave splash, and wave slap were very noticeable when the port (or high) side was facing the waves. Whenever a substantial wave would hit the model, a vertical sheet of water would shoot up the side of the hull to a height well above the weather deck. In this orientation, the model's largest angle of roll, which occurred during sea state 7, was 23.4° to starboard.

Damage Starboard and Forward

Figures 8 and 9 illustrate the model's H 1/3 and H 1/10 significant peak-to-trough roll and pitch motions as a function of two sea states and four orientations after it had been ballasted to simulate flooding near the bow on the starboard side. The damaged model would tend to maintain its direction when subjected to either head seas or following seas. However, in either orientation, it would drift with the waves while pitching noticeably. The largest observed single amplitude pitch angle was 12.7° and occurred in sea state 7 while the model was encountering head seas. A nearly equivalent maximum pitch angle was observed

FIGURE 4: SWATH ROLL MOTIONS
INTACT CONDITION

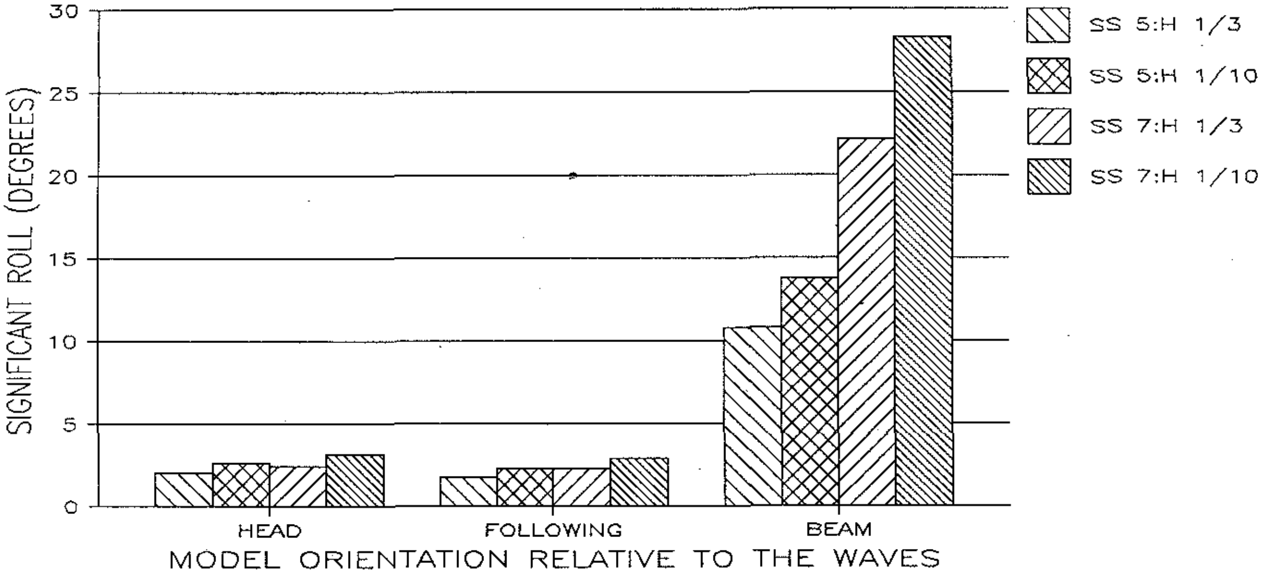


FIGURE 5: SWATH PITCH MOTIONS
INTACT CONDITION

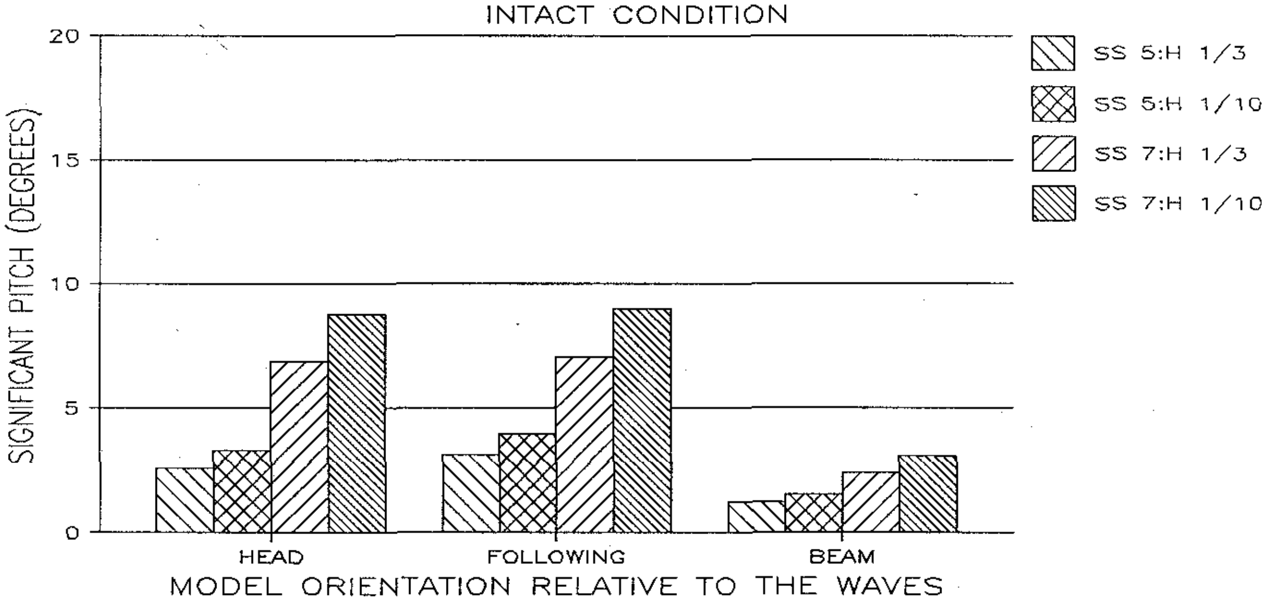


FIGURE 6: SWATH ROLL MOTIONS
DAMAGE STARBOARD AND AFT

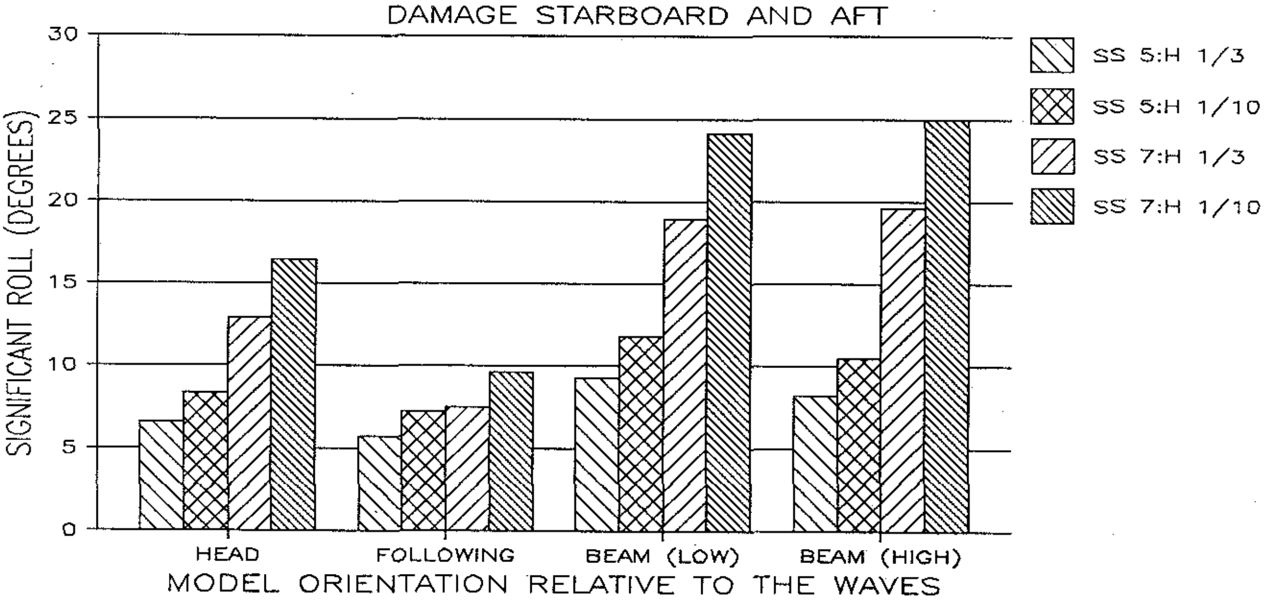


FIGURE 7: SWATH PITCH MOTIONS
DAMAGE STARBOARD AND AFT

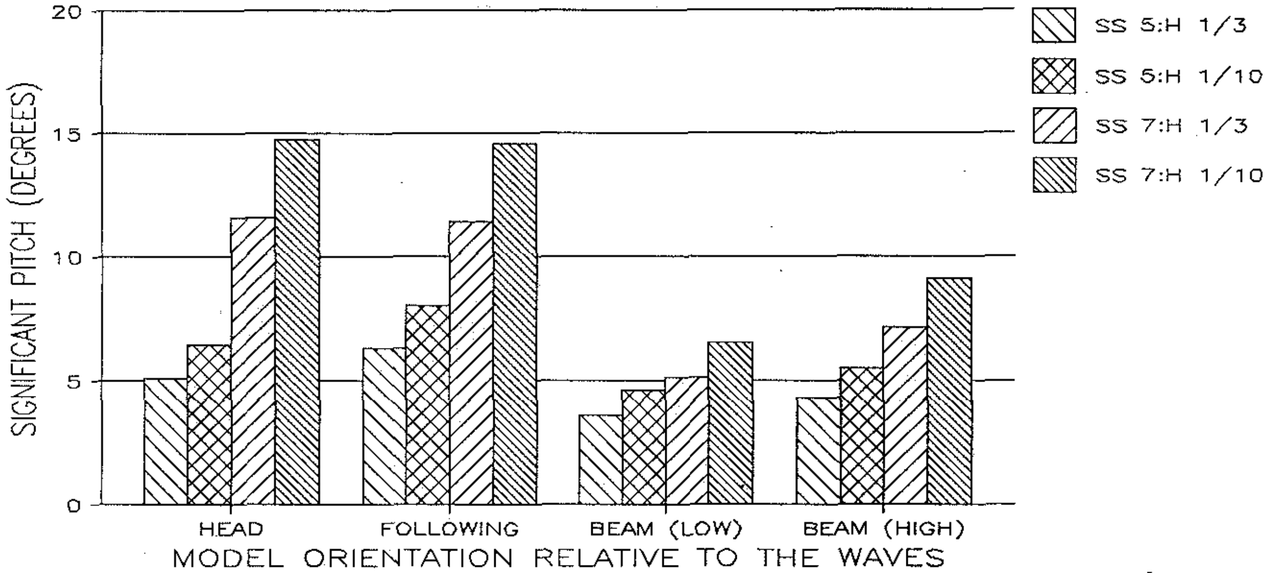


FIGURE 8: SWATH ROLL MOTIONS
DAMAGE STARBOARD AND FORWARD

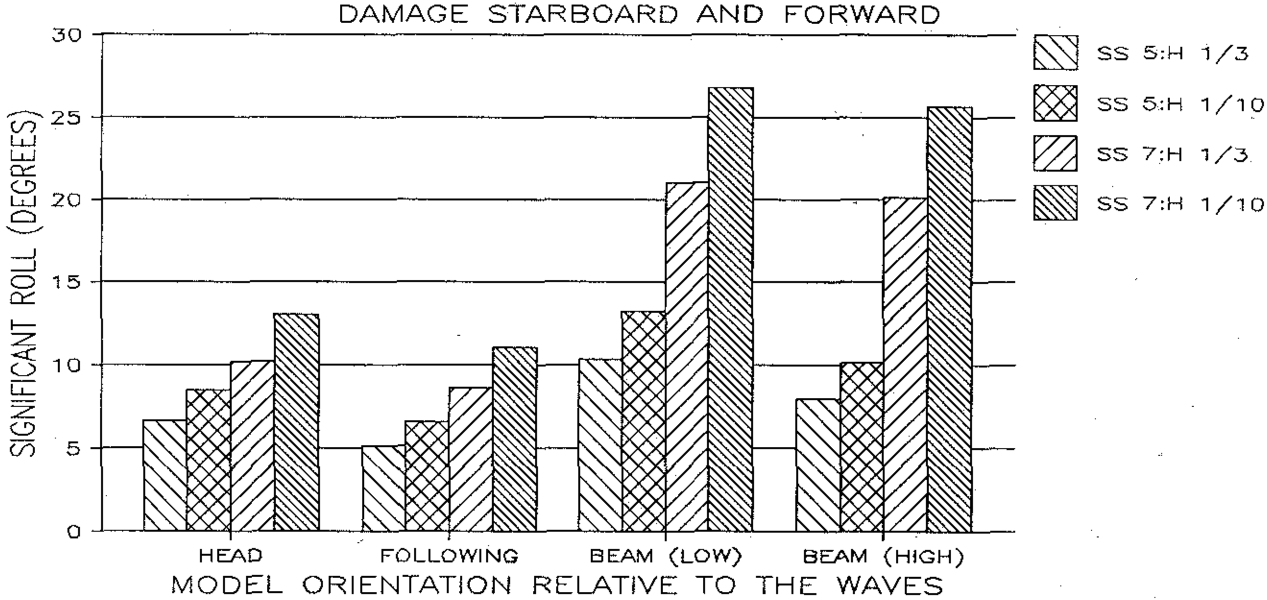
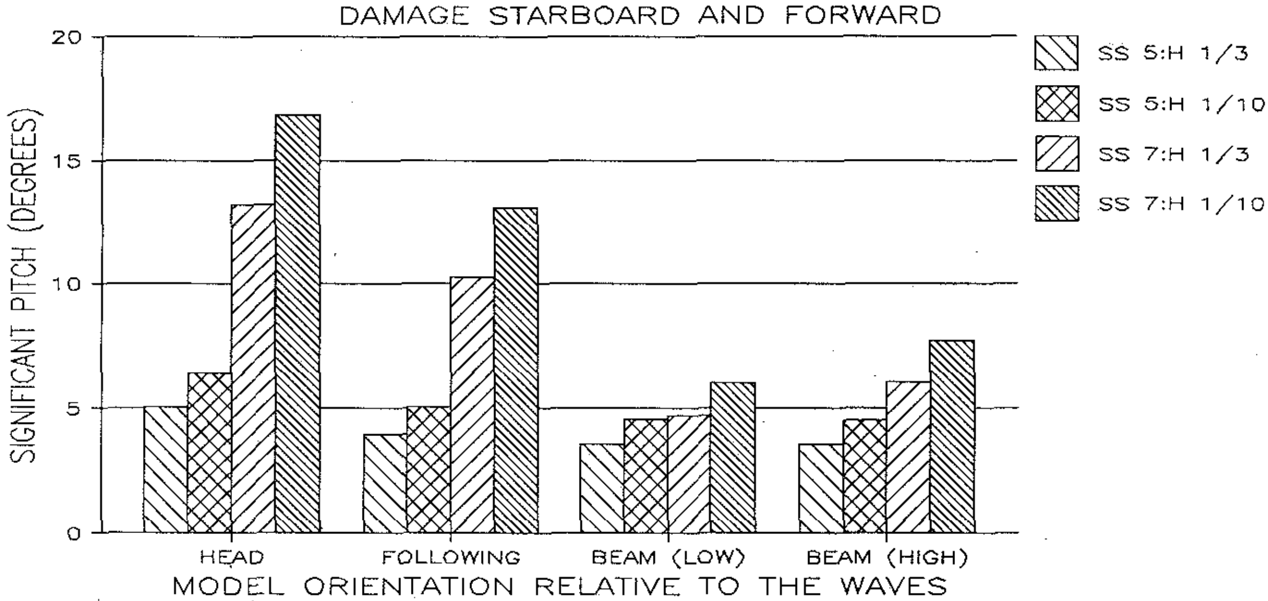


FIGURE 9: SWATH PITCH MOTIONS
DAMAGE STARBOARD AND FORWARD



when the model was in following seas When starting out in beam seas, with either the starboard (or low) side facing the waves or with the port (or high) side facing the waves, the model would tend to pivot about the damaged "corner" and then, with the bow turning away from the waves, begin to drift diagonally across the tank. During these limited tests, solid water never swept completely over the weather deck. However, rolling, wave splash, and wave slap were most noticeable when the port (or high) side was facing the waves. Whenever a substantial wave would hit the model a vertical sheet of water would shoot up the side of the hull to a height well above the weather deck. During sea state 7 and while in this orientation, the model experienced a maximum roll angle of 26.6° to starboard.

CONCLUSIONS

The results of these experiments indicate that this SWATH model had sufficient stability to survive the damaged conditions and sea states which were modeled. However, the full scale vessel's structural integrity, operational performance and maneuvering ability would most likely be adversely effected by the severe motions that the hull would experienced. For example, the propeller ring on the port side was only partially submerged when the simulated damage was on the starboard side near the bow. In both damaged conditions, the propeller rings would submerge and emerge as the model rolled and pitched. Finally, it should be noted that while these limited tests did not produce a capsizing, there is absolutely no guarantee that this hull would be able to survive all possible combinations of waves, wind conditions, displacement and center of gravity variations, and damage scenarios.

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SHIP STABILITY IN WAVES: ON THE PROBLEM OF RIGHTING MOMENT ESTIMATIONS FOR SHIPS IN OBLIQUE WAVES

by I.K.Boroday *

ABSTRACT

An analysis is performed on the structure of the righting moment of a ship moving at specified speed on an arbitrary course in waves. Shown are the components determined by hydrodynamic forces of different nature: the Krylov force, diffraction force and the one induced by ship motions. Expressions are given which may be used for calculation of the righting moment value for an arbitrary angle of heel in wave conditions.

The adverse effect of the following seas resulting in a reduction of the ship's transverse stability and a situation dangerous from the viewpoint of capsizing is well-known. This mode of ship's motion may by accompanied by the development of intensive rolling oscillations of a subharmonic (parametric) nature. However, of no less danger are the modes of ship's motion on courses close to the following ones when stability changes are accompanied by the ship's rolling, which in the general case represents a set of subharmonic and regular "forced" oscillations. A correct assessment of the righting moment under complex sea conditions is of paramount importance for the prediction of ship's safety in waves.

The transverse stability of the ship moving at an arbitrary heading angle χ to the direction of two-dimensional waves in calm water is characterized by the righting moment representing a supporting--force moment due to athwartship inclinations about the central longitudinal axis of the ship's mass.

As applied to the ship, let us introduce the term "righting moment in waves" M_8 which may be considered as generalization of the moment in calm water [2].

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The righting moment in waves is defined as a moment about the longitudinal central axis of hydrodynamic forces acting on the ship which is inclined at a constant angle of heel, with all other degrees of freedom remaining unrestricted. For the ship under consideration the righting moment in waves will be the function not only of the heel but also of the time. In the absence of waves, the moment M_8 is identical to the righting moment in calm water in the conventional sense. It follows from the above that the moment M_8 includes the exciting moment as defined in the theory of ship rolling motions. In this respect, the term "righting moment in waves" introduced in this paper does not essentially differ from that used in a number of papers, beginning from Froude's work [8], i.e. from the term "supporting--force moment" which includes the action of hydrostatic restoring forces of the inclined ship in calm water and the excitation effect due to waves. A similar idea of the righting moment in waves which had been defined, however, taking into account only hydrostatic pressure due to free surface curvature, was used for instance by K.Wendel, B. Arndt, S.Kastner, S.Roden [7, 9].

FORMULATION OF THE PROBLEM

The analysis of the righting moment for finite angles of heel will be based on the assumptions that the fluid movement is potential, the ship inclined at a preset angle experiences heave, pitch and sway, the amplitudes of all modes of ship's oscillations and waves induced thereby are the values of the first order of smallness, the solution is restricted to application of the small amplitude wave theory.

Let us consider the ship moving on a preset course in two-dimensional waves. The fixed coordinate system $\xi_1 \eta_1 \zeta_1$ is so selected that the $O_1\xi_1$ - axis is coincident with the direction of the wave propagation, and the plane $\xi_1 O_1 \eta_1$, with the undisturbed surface of water (Fig.1). The rectangular coordinate system $\xi \eta \zeta$ is moving at the ship's average speed v, with the vector directed along the $O\xi$ -axis. The plane $\xi O \zeta$ of this system coincides with the centerplane of the upright ship in calm water, and the plane $\xi O \eta$ with the undisturbed water surface.

The ship's heading is determined by the angle χ between the $O_1\xi_1$ - and $O\xi$ - axes. For the ship at zero speed the origins of the coordinate systems $\xi \eta \zeta$ and $\xi_1 \eta_1 \zeta_1$ coincide. The origin of the xyz - axes fixed in the ship will be at its center of gravity marked by point G, and the plane xGz will be coincident with the ship centerplane. Let us suppose that the longitudinal GX - axis is the main axis of inertia. Besides, let us introduce axes $x_2 y_2 z_2$ half-related to the ship, which remain parallel to $\xi \eta \zeta$ - axes when the ship oscillates in waves. The origin of this system is in the ship's center of gravity. At zero angles of yaw and trim the plane $z_2 G y_2$ is parallel to the middle section.

Fig.1 shows the waterline $W_1 L_1$ for the ship's inclination at equal volume in calm water and the instantaneous wave waterline WL.

The positive direction of the heeling angle Θ will be taken for the ship's inclination to starboard. The pitch motions of the inclined ship will be defined as oscillations about Gy_2 - axis, with the trim by the stern, usually corresponding to the positive angle of pitch.

In accordance with the existing tradition the rule of signs for the moment M_B will be assumed as opposite to the rule of signs for the angles of heel. Thus, with $\Theta > 0$ the moment $M_B > 0$, if it shows the tendency to turn the ship from starboard to port. However in theoretical calculations this rule of signs for M_B is not always convenient. Therefore, let us denote the main moment of hydrodynamic forces by M_h, and the rule of signs for this moment will be the same as for the angles of heel:

$$\bar{M}_B = - \bar{M}_h / _{\theta = Const} \quad ,$$

where the bar over the letter is the vector symbol.

Let us at first determine the moment $\bar{M}_h$, as applied to the ship whose behaviour in waves is described by equations of roll, pitch, heave and sway motions, excluding yaw and surge ($\varphi(t) = 0, \xi_g(t) = 0$), and then take account of the $\bar{M}_h$ variation following the restriction of the ship's free movement due to the heeling.

$$M_h = \int_{\Omega_t} P' \left[z \cos(\hat{ny}) - y \cos(\hat{nz}) \right] d\Omega \quad , \qquad (1)$$

where
- P' = excessive pressure applied to the hull as compared to the pressure on the free water surface;
- $d\Omega$ = component of the ship surface;
- Ω_t = instantaneous wetted surface of the ship;
- $\bar{n}$ = outer normal to the surface;
- y, z = coordinates of the surface.

It should be noted that the moment M_h determines the fluid response to terminal rolling oscillations of the ship.

Let the wave motion of the fluid be so slow that the pressure can be defined by the linearized Cauchy integral; in this case the potential Φ of the absolute fluid movement is obtained from the solution of a nonlinear hydrodynamic problem of the rolling motions of terminal amplitude, when the oscillations of different modes are small.

A successive solution of this problem using the small parameter method is reported in Ref. [4].

Using the results of that work as the base, the potential Φ will be given by the following sum:

$$\Phi = \Phi_v + \Phi_1 + \Phi_2 + \Phi_3 \quad . \qquad (2)$$

Here
- Φ_v = potential of the fluid movement due to ship running in calm water;
- Φ_1 = potential of free oncoming waves;
- Φ_2 = potential of fluid movement induced by the ship's motions in calm water;
- Φ_3 = potential of the diffracted wave motion.

It should be remembered that the potentials Φ_v, Φ_1, Φ_2, Φ_3 in the problem under consideration are dependent upon the angle of ship's inclination.

To pass from the general expression (1) to the case we are interested in, account must be taken of the condition for fixation of the ship's heeling angle.

Firstly, the potential Φ_2 shall be calculated on the assumption that no rolling motions occur (let it be designated here as Φ_2^*); secondly, relative changes should be introduced in the wetted surface over which the integration is performed

$$\Omega_{t/\theta=const} = \Omega^* .$$

Proceeding from the above, the moment M_θ is given as the following sum

$$M_\theta = M_v + M_1 + M_2 + M_3 \;, \qquad (3)$$

where

$$M_v = \rho \int_{\Omega^*} \frac{\partial_a \Phi_v}{\partial t} \left[z \cos(\hat{ny}) - y \cos(\hat{nz}) \right] d\Omega, \quad (4)$$

$$M_1 = \gamma \int_{\Omega^*} \zeta \left[y \cos(\hat{nz}) - z \cos(\hat{ny}) \right] d\Omega +$$

$$+ \rho \int_{\Omega^*} \frac{\partial_a \Phi_1}{\partial t} \left[z \cos(\hat{ny}) - y \cos(\hat{nz}) \right] d\Omega, \quad (5)$$

$$M_2 = \rho \int_{\Omega^*} \frac{\partial_a \Phi_2^*}{\partial t} \left[z \cos(\hat{ny}) - y \cos(\hat{nz}) \right] d\Omega, \quad (6)$$

$$M_3 = \rho \int_{\Omega^*} \frac{\partial_a \Phi_3}{\partial t} \left[z \cos(\hat{ny}) - y \cos(\hat{nz}) \right] d\Omega. \quad (7)$$

The term M_1 in formula (3) determines the component of the moment M_θ based on the hypothesis of A.N.Krylov. This component may be treated as the main part of the righting moment.

The value of M_2 depends upon the ship's motions at a given angle of heel in calm water. From the viewpoint of physics it is evident that this component is associated with the inertial and damping properties of the fluid.

The term M_3 allows for disturbance of the fluid motion due to the presence of the inclined motionless ship in the way of wave propagation.

The term M_v is the result of changes in the pressure field of the fluid due to disturbed motion induced by the ship running in calm water, the ship itself being the reason of this motion. However, the following considerations must be kept in mind. Let us suppose that experimental assessment is made for the difference between the righting moment of the ship (model) running or standing in calm water with a given angle of heel. In the written expressions this difference is not assessed by the term M_v only. In addition, account must be taken of the change in the hydrostatic component of the term M_1 due to the difference between the wetted surface of the ship running or standing.

THE MAIN PART OF THE RIGHTING MOMENT

Converting from surface to volume integrals in formula (3) and introducing kinematic characteristics of the wave particle, the moment M_1 can be expressed as

$$M_1 = M_\theta^\circ + M_{11} + M_{12} + M_{1v} \qquad (8)$$

Here

$$M_\theta^\circ = \gamma \cos\theta \int_{V^\circ} y \, dV - \gamma \sin\theta \int_{V^\circ} z \, dV \qquad (9)$$

is the righting moment of a ship standing in calm water

$$M_{11} = \gamma \cos\theta \int_{v_t} y \, dv - \gamma \sin\theta \int_{v_t} z \, dv, \qquad (10)$$

$$M_{12} = \rho \sin\theta \int_{V^\circ} \ddot{\eta}_\theta \, y \, dV - \rho \cos\theta \int_{V^\circ} \ddot{\zeta}_\theta \, y \, dV +$$

$$+ \rho \cos\theta \int_{V^\circ} \ddot{\eta}_\theta \, z \, dV + \rho \sin\theta \int_{V^\circ} \ddot{\zeta}_\theta \, z \, dV, \qquad (11)$$

where

V° = volume of the submerged part of the ship at a given angle of heel in calm water;

$\ddot{\eta}_\theta, \ddot{\zeta}_\theta$ = components of acceleration of fluid particles in the coordinate system $\xi \eta \zeta$.

The values above the first order of smallness are not included in formula (8). In this case, it has been taken into account that the difference of volumes

$$v_t = V_t - V^\circ - V_v \;, \qquad (12)$$

represents an additional time-varying volume which is determined by the position of the wave profile in relation to the oscillating ship inclined at angle θ ; this value has the first order of smallness. In formula (12) V_t is the instantaneous volume of the submerfed hull, and V_v is the volume of the hull corresponding to the difference between the wetted surfaces of the ship running or standing in calm water.

It should be emphasized that the superposition of independent volumes according to (12) suggests, as in the Ref. [4] that the results of the theory of small amplitude waves are valid for the problem under consideration.

The characteristics of the moment M_g° become the subject of ship statics. M_{1v} is the hydrostatic component of the righting moment taking into account the change in the waterline form for the ship running in calm water.

The term M_{11} is determined by integration of hydrostatic pressures over the additional volume causing the ship's motions and the oncoming free waves.

The moment M_{12} considers the peculiarity of pressure distribution in the disturbed fluid. In literature this distribution is sometimes interpreted as the so called Smith effect.

For the case of plane progressive waves the potential Φ_1 is derived from the formula

$$\Phi_1(\xi,\eta,\zeta,t)=-\tau\frac{g}{\omega}e^{-K\zeta}\sin(K_1\xi + K_2\eta - \omega_K t). \quad (13)$$

Then, the component

$$\begin{aligned}M_{12}=&\tau\rho\omega^2\Big[\cos\theta\int_{V^\circ} e^{-K\zeta}\cos(K_1X+K_2y_1)\,y\,dV - \\ &-\sin\theta\sin\chi\int_{V^\circ} e^{-K\zeta}\sin(K_1X+K_2y_1)\,y\,dV - \\ &-\cos\theta\sin\chi\int_{V^\circ} e^{-K\zeta}\sin(K_1X+K_2y_1)\,z\,dV - \\ &-\sin\theta\int_{V^\circ} e^{-K\zeta}\cos(K_1X+K_2y_1)\,z\,dV\Big]\cos\omega_K t + \\ &+\tau\rho\omega^2\Big[\sin\theta\sin\chi\int_{V^\circ} e^{-K\zeta}\cos(K_1X+K_2y_1)\,y\,dV + \\ &+\cos\theta\int_{V^\circ} e^{-K\zeta}\sin(K_1X+K_2y_1)\,y\,dV + \\ &+\cos\theta\sin\chi\int_{V^\circ} e^{-K\zeta}\cos(K_1X+K_2y_1)\,z\,dV - \\ &-\sin\theta\int_{V^\circ} e^{-K\zeta}\sin(K_1X+K_2y_1)\,z\,dV\Big]\sin\omega_K t. \end{aligned} \quad (14)$$

Here the following designations are used:

τ = wave amplitude

K = wave form frequency

$K_1 = K\cos\chi$,

$K_2 = K\sin\chi$,

y_1 = ordinate of the elementary volume of dV in the coordinate system $X_1 y_1 z_1$ fixed to the ship, in which the plane $y_1 O_1^* z_1$, passes through the ship's C.G., and the plane $X_1 O_1^* z_1$ coincides with the plane $\xi O \zeta$ when no ship's motions occur (Fig.1).

Owing to smallness of pitch and trim angles of the inclined ship in calm water, it is assumed for derivation of relation (14) that

$$\xi = X_1 = X .$$

Let us analyse the moments M_{11} and M_{12} for small angles of heel corresponding to the values of the ship's inclinations considered in the linear theory of rolling. With an accuracy to the first order of smallness

$$M_{11} = \gamma\int_{V_t} y\,dV . \quad (15)$$

At the heading angle χ = 0 corresponding to the ship running in following waves, we shall obtain M_{12} = 0 with the same accuracy, and besides that, in this case for the wall-sided vessel M_{11} = 0 due to the hull symmetry about the centerplane.

Therefore, if the wall-sided vessel is positioned in the beam seas, the change in the main part of the righting moment at small angles of heel will be of the second order of smallness. The same conclusion may be derived from the results obtained in Ref. [4].

It may be easily found that at large angles of heel a change in the main part of the righting moment in waves will be of the first order of smallness, as are the amplitudes of the oncoming waves.

While the ship is running beam-on to regular seas ($\chi = \pi/2$), the sum of expressions (14) and (15) will give the value of the main part of the exciting moment calculated on the basis of the linear theory of rolling motions [1].

THE RIGHTING MOMENT COMPONENT DUE TO SHIP MOTIONS

Turning to the analysis of the component M_2 let us first for convenience find the hydrodynamic moment $\bar{M}_{2h} = -\bar{M}_2$.

Let us use the following representation of the hydrodynamic moment [3]

$$M_{2h} = -\frac{\partial I_x^\circ}{\partial t} - \omega_y I_z^\circ + \omega_z I_y^\circ - \\ - u_y B_z + u_z B_y \ , \qquad (16)$$

where $I_x^\circ, I_y^\circ, I_z^\circ$ - projections on to respective axes of the vector of the main moment of pressure impulses about the origin of the ship fixed axes,

B_y , B_z - projections of the main vector of pressure impulses,

ω_y , ω_z - projections of the angular velocity vector of the ship fixed axes,

u_y , u_z - projections of the linear velocity vector at the origin of the ship fixed axes.

Taking into account that the angle between axes G_x and G_{x_2} is small in this case the projections of the vectors mentioned above will be referred to axes $x_2 y_2 z_2$ and the velocity projections will be equal:

$$u_x = u_1 = v \ , \ u_y = u_2 = \dot{\eta}_g \ , \\ u_z = u_3 = \dot{\zeta}_g + v\psi , \ \omega_x = u_4 = \dot{\theta} = 0 , \\ \omega_y = u_5 = \dot{\psi} \ , \ \omega_z = u_6 = \dot{\varphi} \ , \qquad (17)$$

where $\dot{\varphi}$ - yaw rate.

If the potential Φ_2^* is represented as a sum taken over radiation functions

$$\Phi_2^* = \sum_{j=1}^{j=6} u_j \varphi_j \ , \qquad (18)$$

the following relations obtained with an accuracy of the first order of smallness become valid

$$\frac{\partial I_x^\circ}{\partial t} = -\rho \Big[\ddot{\eta}_g \int_{\Omega^\circ} \varphi_2 \frac{\partial \varphi_4}{\partial n} \partial\Omega + \\ + (\ddot{\zeta}_g + v\psi) \varphi_3 \frac{\partial \varphi_4}{\partial n} \ d\Omega + \\ + \ddot{\psi} \int_{\Omega^\circ} \varphi_5 \frac{\partial \varphi_4}{\partial n} \ d\Omega + \ddot{\varphi} \int_{\Omega^\circ} \varphi_6 \frac{\partial \varphi_4}{\partial n} \ d\Omega \Big] , \qquad (19)$$

$$\omega_y I_z^\circ = -\rho \dot{\psi} v \int_{\Omega^\circ} \varphi_1 \frac{\partial \varphi_6}{\partial n} d\Omega \ , \qquad (20)$$

$$\omega_z I_y^\circ = -\rho \dot{\varphi} v \int_{\Omega^\circ} \varphi_1 \frac{\partial \varphi_5}{\partial n} d\Omega \ , \qquad (21)$$

$$u_y B_z = -\rho \dot{\eta}_g v \int_{\Omega^\circ} \varphi_1 \frac{\partial \varphi_3}{\partial n} d\Omega \ , \qquad (22)$$

$$u_z B_y = -\rho (\dot{\zeta}_g + v\psi) v \int_{\Omega^\circ} \frac{\partial \varphi_2}{\partial n} d\Omega \ . \qquad (23)$$

In regular waves we have

$u_j = u_{j_0} e^{i\omega_k t}$, j = 2,3,5,6

where u_{j_0} - amplitude of oscillations.

Then by introducing a complex quantity dependent on the frequency of ship oscillations and, generally speaking, on the ship speed

$$\mu_{j\ell} - \frac{i}{\omega_k} \lambda_{j\ell} = \\ = -\rho \int_{\Omega^\circ} \varphi_j \frac{\partial \varphi_\ell}{\partial n} \ d\Omega \ , \qquad (24)$$

where $\mu_{j\ell}$ - added mass of sea water

$\lambda_{j\ell}$ - damping coefficient to ship oscillations

we obtain an expression for the hydrodynamic moment M_{2h} :

$$M_{2h} = -(\ddot{\eta}_g \mu_{24} + \dot{\eta}_g \lambda_{24} + \ddot{\zeta}_g \mu_{34} + \\ + \dot{\zeta}_g \lambda_{34} + v\dot{\psi}\mu_{34} + v\psi\lambda_{34} + \ddot{\psi}\mu_{54} + \\ + \dot{\psi}\lambda_{54} + \ddot{\varphi}\mu_{64} + \dot{\varphi}\lambda_{64}) - v\big[\dot{\psi}\mu_{16} - \\ - \dot{\varphi}\mu_{15} + \dot{\eta}_g \mu_{13} - (\dot{\zeta}_g + v\psi)\mu_{12}\big] . \qquad (25)$$

Relation (25) takes into accou't translational motion of the ship along the longitudinal horizontal axis GX_2 . From formula (24) it is clear that in general the numerical values of the added mass $\mu_{j\ell}$ and damping coefficient $\lambda_{j\ell}$ vary with the heeling angle.

Passing on to the component M_2 and assuming that no ship yaw exists, we obtain

$$M_2 = M_{21} + M_{22} \ , \qquad (26)$$

where

$$M_{21}=\ddot{\zeta}_g\mu_{24}+\dot{\zeta}_g\lambda_{24}+\ddot{\zeta}_g\mu_{34}+$$

$$+\dot{\zeta}_g\lambda_{34}+\ddot{\psi}\mu_{54}+\dot{\psi}\lambda_{54}\,, \tag{27}$$

$$M_{22}=v\left[\dot{\psi}(\mu_{34}+\mu_{16})+\psi\lambda_{34}+\right.$$

$$\left.+\dot{\zeta}_g\mu_{13}-(\dot{\zeta}_g+v\psi)\mu_{12}\right]. \tag{28}$$

The moment M_{22} explicitely takes into account the effect of the forward ship speed on the value of M_2.

It should be noted that statical moments (27) and added mass moments (28) as well as damping coefficients are related to the $X_2\,Y_2\,Z_2$ axes. If small heeling angles are considered, it should be assumed basing on the linear theory of ship motions that

$$\mu_{12}=0\,,\quad \mu_{13}=0\,,\quad \mu_{16}=0\,,$$
$$\mu_{34}=0\,,\quad \mu_{54}=0\,,\quad \lambda_{34}=0, \lambda_{54}=0$$

because of the symmetry of the ship hull about the centerline plane.

In this case, considering the values of the first order of smallness, we shall have

$$M_2=M_{21}=\ddot{\zeta}_g\mu_{24}+\dot{\zeta}_g\lambda_{24}\,. \tag{29}$$

The sum (29) accounts for inertial and damping effects of the fluid arising during transverse movement of the ship. In the linear theory of ship motions this sum is included in the roll equation [5].

Considering relations (27), (28) it can be concluded that the hydrodynamic characteristics play unequal parts in these expressions. In view of the lengthened hull form of the ship the terms with μ_{16}, μ_{12}, μ_{13} will be apparently small. By neglecting them the component under consideration can be approximated by

$$M_2=M_{21}+v\left[\dot{\psi}\mu_{34}+\psi\lambda_{34}\right]$$

DIFFRACTION COMPONENT

Analyzing this component it is convenient from the methodical point of view to neglect at first the effect of the ship speed on the amplitude, introducing appropriate corrections after the structure of the moment M_3 is found. Naturally, the period of the moment variation in regular waves will be expressed using the encounter frequency ω_κ in both cases.

Based on the results obtained in Ref. [6] the moment M_3 may be written as

$$M_3=-\rho g z \kappa e^{i\omega_\kappa t}\int_{\Omega^\circ}\varphi^*\varphi_4\Big(-i\cos\chi\frac{\partial\varphi_1}{\partial n}- \tag{30}$$

$$-i\sin\chi\;\frac{\partial\varphi_2}{\partial n}\;-\;\frac{\partial\varphi_3}{\partial n}\Big)\,d\Omega\,,$$

where

$$\varphi^*=e^{-\kappa\zeta-i\kappa(\xi\cos\chi+\eta\sin\chi)}.$$

Complex representation in (30) is used for convenience. Later on only the real part of the complex quantities should be taken into account.

First, let us consider a case when the ship breadth and length are small relative to the wavelength. Assuming

$$\kappa\zeta\to 0\,,\;\; \kappa\xi\to 0\,,\; \kappa\eta\to 0\,, \tag{31}$$

we obtain

$$M_3=-z\omega^2 e^{i\omega\kappa t}\Big[i\cos\chi\big(\mu_{14}-\frac{i}{\omega}\lambda_{14}\big)+$$

$$+i\sin\chi\big(\mu_{24}-\frac{i}{\omega}\lambda_{24}\big)+\big(\mu_{34}-\frac{i}{\omega}\lambda_{34}\big)\Big]\,, \tag{32}$$

where (24) is taken into consideration.

By writing the potential of oncoming progressive waves in the complex form (13) including relation (31) and designating the moment M_3, for a small ship, as M_{31}, we may obtain

$$M_{31}=-\ddot{\xi}^\circ_{1g}\mu_{14}\cos\chi-\dot{\xi}^\circ_{1g}\lambda_{14}\cos\chi-$$

$$-\ddot{\xi}^\circ_{1g}\mu_{24}\sin\chi-\dot{\xi}^\circ_{1g}\lambda_{24}\sin\chi- \tag{33}$$

$$-\ddot{\zeta}^\circ_{1g}\mu_{34}\;-\;\dot{\zeta}^\circ_{1g}\cdot\lambda_{34}\;,$$

where $\dot{\xi}^\circ_{1g}$, $\ddot{\xi}^\circ_{1g}$, $\dot{\zeta}^\circ_{1g}$, $\ddot{\zeta}^\circ_{1g}$ =absolute velocities and accelerations of wave particles at the origin of coordinates ξ,η,ζ.

From (33) we may obtain a relationship between the moment M_{31} and linear components of wave motion velocities and accelerations.

In order to take account of commensurability of the wave length and ship size to a first approximation, we should represent the function φ^* as a series limiting the expansion terms to the first power

$$\varphi^* = 1 - K\zeta - iK(\xi\cos\chi + \eta\sin\chi). \quad (34)$$

Further transformations will give the following relation for the moment M_3

$$M_3 = M_{31} + M_{32} + M_{33}, \quad (35)$$

Here

$$M_{32} = -\ddot{\alpha}\mu_{54}\cos\chi - \dot{\alpha}\lambda_{54}\cos\chi -$$

$$-\ddot{\alpha}\mu_{44}\sin\chi - \dot{\alpha}\lambda_{44}\sin\chi - \quad (36)$$

$$-\ddot{\alpha}\mu_{64}\cos\chi\sin\chi - \dot{\alpha}\lambda_{64}\cos\chi\sin\chi,$$

$\dot{\alpha}, \ddot{\alpha}$ = absolute velocity and acceleration of the surface angle of wave slope at the origin of coordinates ξ, η, ζ.

The component M_{33} is unwieldy and may be given as the product of factor $(-\rho\omega^2\alpha_0 e^{i\omega_K t})$, where α_0 is amplitude of the angle α, by the sum containing the integrals in the form

$$\int_{\Omega^0} y_1 \varphi_4 \frac{\partial \varphi_3}{\partial n} d\Omega. \quad (37)$$

Strictly speaking, the relations (33), (36) correspond to the case when the ship is at standstill. It has been known [5] that the influence of the ship speed will result in that the amplitude values of the inertial part of the exciting forces defining the hydrodynamic components are proportional to the product $\omega\omega_K$; then the amplitudes of the speed term is determined from true frequency. This condition shall be taken into account in calculation of the diffraction component of the righting moment for the ship underway.

The structure of expression (36) shows that in the general case consideration of the ship's dimensions leads to the terms proportional to the wave slope angle.

Assuming in (33) and (36) that the angles of heel are small, and therefore the ship's hull is symmetric about the centerplane, we shall have

$$M_{31} = -\ddot{\xi}^0_{1G}\mu_{24}\sin\chi - \dot{\xi}_{1G}\lambda_{24}\sin\chi, \quad (38)$$

$$M_{32} = -\ddot{\alpha}\mu_{44}\sin\chi - \dot{\alpha}\lambda_{44}\sin\chi -$$

$$-\ddot{\alpha}\mu_{64}\cos\chi\sin\chi - \quad (39)$$

$$-\dot{\alpha}\lambda_{64}\cos\chi\sin\chi.$$

In case of the ship travelling across the crests ($\chi = 0$) it follows from (38), (39) that $M_{31} = 0$, $M_{32} = 0$, and the moment M_{33}, also turns out to be zero.

If the heading angle $\chi = \pi/2$ (beam to the sea), we shall obtain from (38) and (39)

$$M_{31} = -\ddot{\xi}^0_{1G}\mu_{24} - \dot{\xi}^0_{1G}\lambda_{24}, \quad (40)$$

$$M_{32} = -\ddot{\alpha}\mu_{44} - \dot{\alpha}\lambda_{44}. \quad (41)$$

Besides, for this case

$$M_{33} = -2i\tau\rho\omega^2 e^{i\omega t}\int_{\Omega^0} y_1\varphi_4\frac{\partial\varphi_3}{\partial n}d\Omega. \quad (42)$$

The sum $M_{31} + M_{32} + M_{33}$ the terms of which are determined from formulae (40) - (42) is equal to the expression for the hydrodynamic component of the exciting moment obtained in Ref. [4] when solving the problem of small amplitude of rolling.

CALCULATION OF DIFFRACTION COMPONENT

This component of the righting moment for an arbitrary relationship between the ship's dimensions and regular wave length based on (30) may be expressed as:

$$M_{33} = \gamma\alpha_0\sqrt{I_I^2 + I_{II}^2}\cos(\omega_K t - \delta M_3), \quad (43)$$

where $\delta M_3 = \operatorname{arc\,tg} \frac{I_{II}}{I_I}$.

Here the following designations are used

$$I_I = I_3 + I_4 \cos\chi + I_5 \sin\chi \,, \qquad (44)$$

$$I_{II} = -I_1 \cos\chi - I_2 \sin\chi + I_6 \,, \qquad (45)$$

$$I_1 = \int_{\Omega^\circ} e^{-KZ_1} \varphi_4 \cos\beta \frac{\partial \varphi_1}{\partial n} \, d\Omega \,,$$

$$I_2 = \int_{\Omega^\circ} e^{-KZ_1} \varphi_4 \cos\beta \frac{\partial \varphi_2}{\partial n} \, d\Omega \,,$$

$$I_3 = \int_{\Omega^\circ} e^{-KZ_1} \varphi_4 \cos\beta \frac{\partial \varphi_3}{\partial n} \, d\Omega \,,$$

$$I_4 = \int_{\Omega^\circ} e^{-KZ_1} \varphi_4 \sin\beta \frac{\partial \varphi_1}{\partial n} \, d\Omega \,,$$

$$I_5 = \int_{\Omega^\circ} e^{-KZ_1} \varphi_4 \sin\beta \frac{\partial \varphi_2}{\partial n} \, d\Omega \,,$$

$$I_6 = \int_{\Omega^\circ} e^{-KZ_1} \varphi_4 \sin\beta \frac{\partial \varphi_3}{\partial n} \, d\Omega \,,$$

$$\beta = K_1 X_1 + K_2 y_1 \,. \qquad (46)$$

On the basis of the linear wave theory relation (43) defines the value of the moment M_θ component conditioned by wave diffraction near the inclined ship representing a motionless obstacle. The integrals I_j can be expressed in a first approximation in terms of the elements of lines drawing, if the recommendations of [6] are used. In accordance with these recommendations the function φ_i is determined as the velocity potential on the surface of a three-axis ellipsoid in infinite fluid. Therefore, this method makes it possible to consider the inertia forces only. It should be noted that the potential φ_i in (43) refers to the fixed-in-the-ship coordinate system $X_2 Y_2 Z_2$ (Fig.1). For the functions φ_i and potentials φ_i' relating to the coordinate system $X_1 Y_1 Z_1$ the following relations are valid: $\varphi_1 = \varphi_1'$, $\varphi_2 = \varphi_2'$, $\varphi_3 = \varphi_3'$, $\varphi_4 = \varphi_4' - Z_P \varphi_2$; in this case Z_P is the center of gravity above the plane $X_1 O_1^* Y_1$.

In the theory of ship's motions when the calculation procedure for exciting forces is being developed extensive use is made of the method based on introduction of the reducing coefficients taking into account the relations between the regular wave length and ship's dimensions. Since the value of M_3 for small ships is expressed as relation (33), we can apply this method and write the diffraction moment as

$$M_3 = \ddot{\xi}^\circ_{16}(-\mu_{24} \varkappa_1 \sin\chi - \mu_{14} \varkappa_2 \cos\chi) - \ddot{\zeta}^\circ_{16} \mu_{34} \varkappa_3 + \dot{\xi}^\circ_{16}(-\lambda_{24} \varkappa_1^\lambda \sin\chi - \lambda_{14} \varkappa_2^\lambda \cos\chi) - \dot{\zeta}^\circ_{16} \lambda_{34} \varkappa_3^\lambda \,, \qquad (47)$$

where $\varkappa_1$, $\varkappa_2$, $\varkappa_3$, $\varkappa_1^\lambda$, $\varkappa_2^\lambda$, $\varkappa_3^\lambda$ = reducing coefficients defined by relations of typical hull dimensions and wave lengths.

When solving the problem, let us consider the means of calculation of coefficients $\varkappa_1$, $\varkappa_2$, $\varkappa_3$, i.e. let us restrict ourselves to consideration of the inertial properties of fluid only. In this case it should be noted that, in general, these reducing coefficients may also be used when estimating the damping components, just as it was done in Ref. [5] .

Comparison of expressions (43) and (47) will give the following relations of reducing coefficients

$$\varkappa_1 = -\frac{\rho}{\mu_{24}} \left(I_2 - \frac{I_6}{\sin\chi} \right) , \qquad (48)$$

$$\varkappa_2 = -\frac{\rho}{\mu_{14}} \, I_1 \,, \qquad (49)$$

$$\varkappa_3 = -\frac{\rho(I_4 \cos\chi + I_5 \sin\chi + I_3)}{\mu_{34}} \,. \qquad (50)$$

It may be easily seen that for small ships the coefficients $\varkappa_1$, $\varkappa_2$, $\varkappa_3$ will be equal to unity.

It should be borne in mind that due to elongated hull form the values of μ_{14} and λ_{14} will apparently be small and, in the first approximation, the terms containing these values may be ignored. In this case the formula for the moment M_3 takes a simpler form:

$$M_3 = -\ddot{\xi}^\circ_{16} \mu_{24} \varkappa_1 \sin\chi - \ddot{\zeta}^\circ_{16} \mu_{34} \varkappa_3 - \dot{\xi}^\circ_{16} \lambda_{24} \varkappa_1^\lambda \sin\chi - \dot{\zeta}^\circ_{16} \lambda_{34} \varkappa_3^\lambda \,. \qquad (51)$$

For example, an expression for the reducing coefficient $\varkappa_1$ is introduced.

The integrals I_2 and I_6 may be converted from surface to volume integrals. As to the domain of integration, let us restrict it to the volume of a parallelepiped, taking into account the approximate definition of the reducing coefficients and the fact that the values of these coefficients are mainly dependent on the relationship between the ship's linear dimensions and the wave length. The length of the parallelepiped L is taken to be equal to the ship's length on waterline at equilibrium in calm water without heel. The calculation shows, that the breadth B^* of the parallelepiped which is equivalent for the purpose of this study to the inclined ship may be taken as

$$B^* = \frac{1}{3}\left(B_5^* + B_{10}^* + B_{15}^*\right),$$

where B_5^*, B_{10}^*, B_{15}^* = values equal to the breadth of the inclined ship in the middle of the draft of frames 5, 10 and 15.

It is advisable to determine the draft of the parallelepiped T^* in a similar manner

$$T^* = \frac{1}{3}\left(T_5^* + T_{10}^* + T_{15}^*\right),$$

with the draft on each frame to be calculated from the formula similar to

$$T_5^* = \frac{B_{15}}{B_5^*}\, T_{15}\, \beta_{15} \quad , \tag{52}$$

where B_{15}, T_{15}, β_{15} are breadth on the waterline, draft and coefficient of fineness of the inclined frame, respectively.

The parallelepiped should be considered symmetric about the vertical plane $X_1 O_1^* Z_1$.

Integrating in the domain of the parallelepiped and assuming that $Z_P = 0$, we obtain the reducing coefficient in the simplest form:

$$\varkappa_1 = \frac{2}{B^* L T^{*2}} \sin\frac{K_1 L}{2}\left[2\frac{B^*}{K_1}\frac{1 - e^{-KT^*}(1+KT^*)}{K^2} \times \cos\frac{K_2 B^*}{2} - \frac{T^*}{\sin\chi} e^{-KT^*} \frac{4}{K_2^2 K_1}\left(\sin\frac{K_2 B^*}{2} - \frac{K_2 B^*}{2}\cos\frac{K_2 B^*}{2}\right)\right]. \tag{53}$$

In formula (53) the need for a limiting transition has been taken into account

$$\lim \varkappa_1\big|_{\omega\to 0} = 1.$$

For a ship without heel the inertia component of the moment M_3 represents a certain item in the hydrodynamic part of the roll-excitation moment which is treated in the linear formulation. It is seen from formula (47) that in this case

$$M_{3in} = -\ddot{\xi}_{18}^{\circ}\, \mu_{24}\, \varkappa_1 \sin\chi \; . \tag{54}$$

It is obvious that the proposed method of assigning the values of B^* and T^* should be applied to the ship in the upright position as well.

Shown in Fig.2 are calculated data on the amplitude of the exciting moment M_θ which is acting on a cylindrical model of the Lewis frame section. The experimental points are also shown. The moment M_{3in} has been calculated for $\chi = \pi/2$ from formula (54). As is seen, the resulting amplitude of the exciting moment is in good agreement with the experimental data.

Now let us turn to derivation of a formula for the reducing coefficient $\varkappa_3$ based on relation (50). Make in the formulae defining the integrals I_3 , I_4 , I_5 a substitution of the ellipsoid potential for function φ_4 . As regards the integration domain, it cannot be defined by the parallelepiped volume, as was assumed in the derivation of the coefficient $\varkappa_1$. This comes from the fact that the term $\ddot{\zeta}_{18}^{\circ}, \mu_{34}, \varkappa_3$ of formula (47) goes identically to zero for a body symmetric about the plane $X_1 O_1^* Z_1$. However, for simplicity, the integration may be carried out over the volume of the cylindrical body of length L inclined at a given angle with the frames coincident with the ship's midsection. Assuming this simplified scheme, the expression for the reducing coefficient can be written in the form

$$\varkappa_3 = \varkappa_3^{\circ}(P_1\cos\chi + P_2\sin\chi + P_3). \tag{55}$$

In this formula the following symbols are used

$$P_1 = \frac{2}{K_2^2}\sin\left(\frac{K_1 L}{2}\right) e^{-K\zeta_c}\int_0^{T_1} Z_1\left[\left(\cos K_2 a_n - \cos K_2 a_\Lambda\right) + K_2\left(a_n \sin K_2 a_n - a_\Lambda \sin K_2 a_\Lambda\right)\right] dZ_1 ,$$

$$P_2 = \frac{2}{K_1} \sin\left(\frac{K_1 L}{2}\right) e^{-K\zeta_c} \int_0^{T_1} z_1 (a_n \sin K_2 a_n - a_\wedge \sin K_2 a_\wedge)\, dz_1 ,$$

$$P_3 = \frac{1}{K_1} \left\{ -KP_1 + \frac{2}{K_2^2} \sin\left(\frac{K_1 L}{2}\right) e^{-K\zeta_c} \int_0^{T_1} \left[\cos K_2 a_n - \cos K_2 a_\wedge) + K_2 (a_n \sin K_2 a_n - a_\wedge \sin K_2 a_\wedge) \right] dz_1 \right\},$$

$$\text{æ}_3^\circ = \left[P_1(0) \cos \chi + P_2(0) \sin \chi + P_3(0) \right]^{-1},$$

$$P_1(0) = K_1 L \int_0^{T_1} z_1 (a_n^2 - a_\wedge^2)\, dz_1 ,$$

$$P_2(0) = tg\,\chi\; P_1(0),$$

$$P_3(0) = \frac{1}{K_1} \left[-KP_1(0) + K_1 L \int_0^{T_1} (a_n^2 - a_\wedge^2)\, dz \right].$$

For simplification of this formulae the exponent is removed from the integral, and the coordinate z_1 is substituted by the value of ζ_c equal to the depth of submergence of CB under the free surface of a ship inclined at a given angle in calm water. Besides, $a_\wedge$, a_n are the distances of the points on port and starboard surfaces of the above inclined cylindrical body, measured in the real water plane, from the plane $X_1 O_1^* z_1$ (Fig.1).

When the ship is positioned across the crests, $\chi = 0$ and the reducing coefficient is expressed as the following elementary telationship

$$\text{æ}_3 = \frac{2}{KL} \sin \frac{KL}{2} e^{-K\zeta_c} \qquad (56)$$

It should be noted that for such a position of the ship the term containing μ_{24} goes to zero, and if the added static moment μ_{14} is neglected, the inertia component of the moment M_θ is determined by the product $\ddot{\zeta}_{1\theta}^\circ$, μ_{34} æ_3 .

The reducing coefficient in the form of the relationship (56) has been used in the calculation of the righting moment for models of fishing vessels tested in the following seas. As an illustration, Fig.3 shows the data on the extreme values of the righting moment calculated with consideration for the inertia component. As is seen, the introduction of this component into the calculation generally improves the convergence with the experimental data. Unfortunately, individual calculations, similar to the one mentioned above, do not allow on a quantitative evaluation of the diffraction component in the righting moment. In order to make this kind of evaluation with confidence, it is necessary to have systematical hydrodynamic characteristics of the inclined frames.

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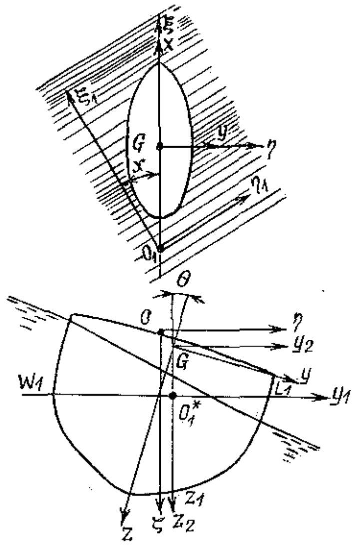

Fig.1 Coordinate systems

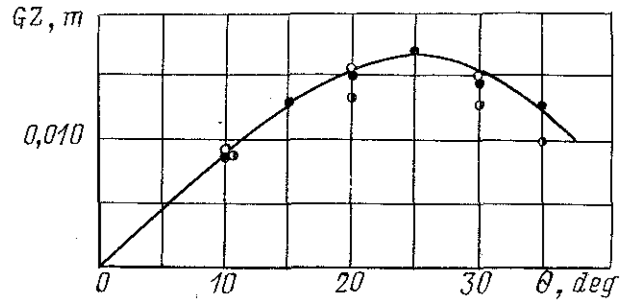

Fig.3 Curve of statical stability of model on wave top

◐ – main part; o - with consideration for inertia component;
● - experiment.

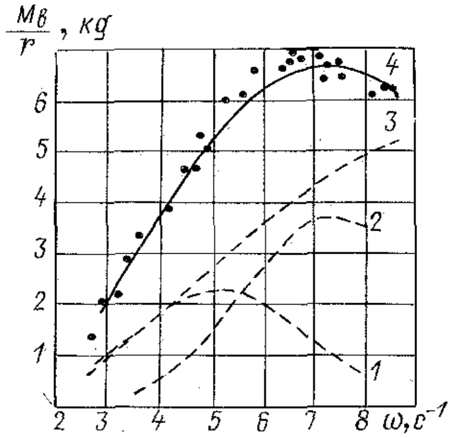

Fig.2 Exciting moment amplitudes

1 - moment of inertia forces;

2 - moment of resisting forces;

3 - main part;

4 - resulting moment;

● - experiment

ON THE INFLUENCE OF THE VARIATION OF RIGHTING LEVERS IN WAVES ON STABILITY REQUIREMENTS

Peter Blume [1]

ABSTRACT

Two extreme models designed to experience large and small variations of righting levers in waves were tested in following irregular waves. The test procedure and sea states used in the tests were the same as in earlier tests series. So the results also can be compared with results for other hull forms. At the limit between safe and unsafe the common stability parameters derived from the smooth water curve are significantly higher for the model with large righting lever variations in waves than for the other model.

The hull form factor introduced some years ago in a proposal for new stabilty criteria serves in a simple way for stability demands depending from individual dimensions. The factor contains with C_B/C_W already a part which should take care for righting lever variations in waves. Due to the small variations of this part within the models tested earlier however it was not possible to find a better correlation by variation of the power. The new results indicate that e.g. a power 2 would lead to a better fit. Therefore a modification of the hull form factor is proposed.

INTRODUCTION

There are well-founded doubts that existing recommendations regarding stability requirements are sufficient to keep the safety level with modern designs as existing in ealier years. In comprehensive tests it has been shown that stability requirements should depend on hull form characteristics. Introducing the hull form factor a practicable method was proposed and also presented to IMO which serves for a better judgement of safety against capsizing for a wide variety of hull forms [1, 2].

The investigation reported here should clarify a special aspect, namely the relation between the variation of righting levers in waves and stability requirements. Therefore two extreme hullforms were designed named Model F and G in continuation of our series. The first one experiences large variations and the second one very small variations of righting levers in waves. Both models were tested in the same manner as the models used before.

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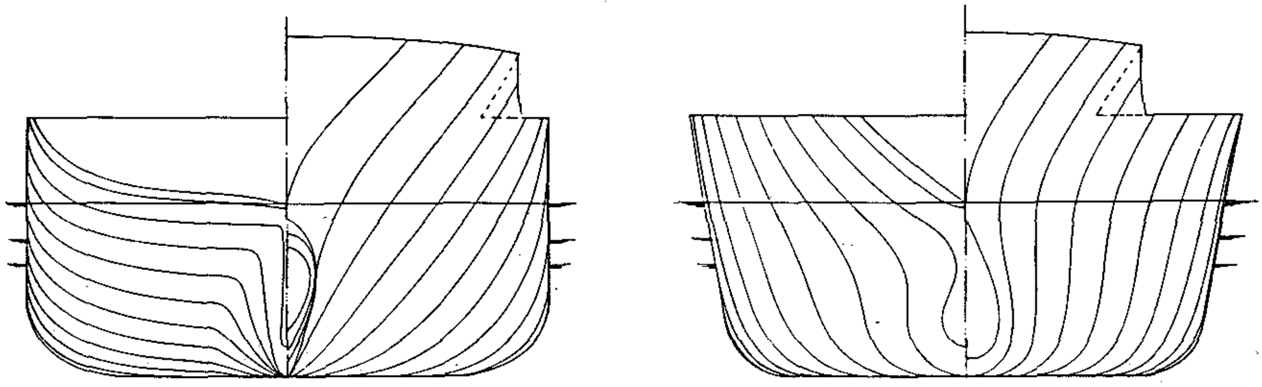
DESCRIPTION OF MODELS AND TESTS

Modern ship designs have very often wide transoms with flat afterbody sections due to the trend to large deck areas. Such hull forms are disposed to larger variations of righting levers in waves because of the large variation of the transverse moment of inertia of the water plane. Therefore at first a hullform named Model F was designed having extrem flat afterbody sections and a vee-shaped forebody.

Also more moderate hullforms experience a loss in transverse moment of inertia at the fore- and afterbody on a wave crest. This can be compensated by an increasing breadth with draft in the midship section. Burcher showed this effect for slender frigate hull forms [3] which have nearly any variation of righting levers in waves. Following these principles the lines of the second Model G were designed under consideration of some restrictions given by the Model F. Length, depth, breadth in CWL, the lateral view and the hatches are the ~~same for~~ both models.

Table 1 contains the main parameters of the models for three draughts used in this investigation. The Figure 1 shows the cross section of the hull forms.

For both models extensive hydrostatic calculations of righting levers in calm water and in waves were executed. The different behaviour of both models in waves can be seen from the



Model F   Model G

Figure 1 Cross sections of the models

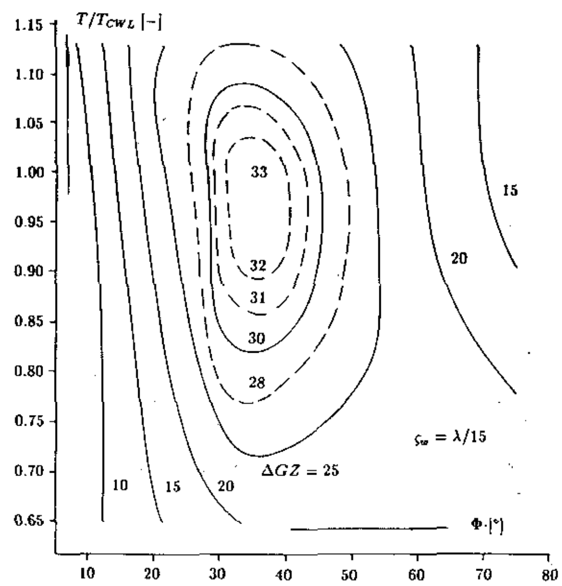


Figure 2 Loss of righting lever on a wave crest for Model F

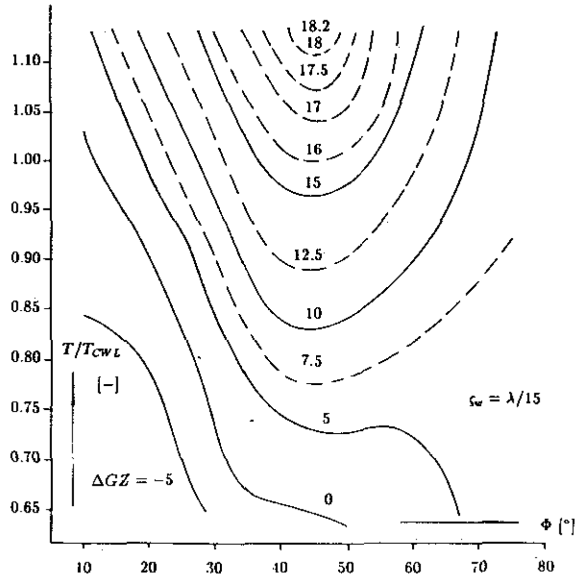


Figure 3 Loss of righting lever on a wave crest for Model G

TABLE 1
Main Dimensions of the Models

Model		F			G		
Model No.		3350			3351		
L_{pp}	$[m]$	5.000			5.000		
L_{oa}	$[m]$	5.280			5.280		
B_{CWL}	$[m]$	0.852			0.852		
T_{CWL}	$[m]$	0.285			0.285		
D	$[m]$	0.426			0.426		
L_{pp} / B_{CWL}	$[-]$	5.87			5.87		
B_{CWL} / D	$[-]$	2.00			2.00		
T	$[m]$	0.285	0.222	0.185	0.285	0.222	0.185
B_{WL}	$[m]$	0.852	0.852	0.852	0.852	0.830	0.817
B_{WL} / T	$[-]$	2.99	3.84	4.61	2.99	3.74	4.42
D / T	$[-]$	1.49	1.92	2.30	1.49	1.92	2.30
V	$[m^3]$	0.703	0.495	0.389	0.722	0.544	0.444
C_B	$[-]$	0.579	0.523	0.489	0.594	0.590	0.588
C_P	$[-]$	0.601	0.548	0.517	0.640	0.630	0.626
C_W	$[-]$	0.830	0.725	0.671	0.689	0.658	0.650

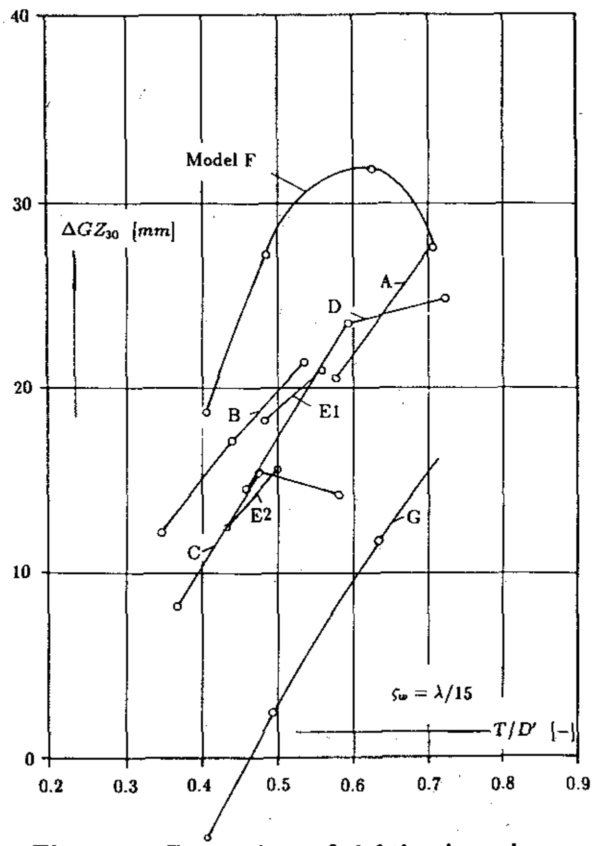


Figure 4 Comparison of righting lever losses at 30° heeling on a wave crest

Figures 2 to 4. The first two figures show the loss of righting lever on a wave crest calculated for a wave of ship length and a wave hight of length divided by 15. There are drawn lines of constant righting lever loss as function of heeling angle and the draft ratio.

Model F has a pronounced maximum of 33 mm loss at CWL and 35° heeling. At Model G the maximum seems to be shiffed to a larger draft. The values are much lower. At smaller heeling angles there are already ranges with negative losses, this is an increase of righting levers on a crest compared to the calm water condition. Figure 4 shows the righting lever loss at 30° heeling as function of the draft related to the weighted depth (which includes a correction for the hatches) for both models in comparison to the models tested earlier. From this figure it is evident that the Models F and G represent extreme cases with regard to the variation of righting levers in waves. All other models are between these extremes and the differences in their behaviour are small.

All tests were performed in the same approved manner as the tests with the earlier models with the intention to get comparable results. The free running models were steered through the tank mainly on a desired relative course of about 30° to the direction of wave propagation. Due to the restricted width of the tank thereby a zig-zag course had to be steered. But usually only the first leg was of importance because in most cases the models there experienced the largest heeling in a distinct high wave group. The encounter with this wave group was guarantied by starting the tests with a certain time lag after starting of the wave maker for which always the same control signal was used.

The models were tested at 3 draughts with different heights of the centre of gravity to find a limit between safe and unsafe. During all tests the models had a static heeling of 2° to port which was the lee-side in the first leg. Beside some other quantities the rolling motion was measured. The maximum heeling angle of each single test run was used for the judgement of the safety. Despite of the quasi deterministic character of the seaway one has to take into account a larger scatter of the results due to unavoidable variations in speed and course influenced by the seaway. Therefore there is a need of a larger number of repeat runs at the same conditions to get reliable statistical values. Usually at least 10 runs were performed at the same conditions and near to the limit between safe and unsafe even more up to 22 runs.

In this test series two irregular seas named P 1.2 and J 2.2 were used. The first one has a Standard-ITTC-Spectrum (Pearson-Moskowitz) and the second one a JONSWAP-Spectrum with an enhancement factor of 5. The significant wave height and peak period of both seaways were nearly the same. These values and other quantities are compiled in the following table:

TABLE 2
Data of the test seaways

	$4\sqrt{m_O}$ [mm]	T_p [s]	$\zeta_{w\ sig}$ [mm]	$\zeta_{w\ max}$ [mm]	$\overline{H}$ [mm]	$\overline{\eta'}$ [mm]
P 1.2	374	2.58	375	728	667	483
J 2.2	376	2.49	370	631	850	507

The first columns are average results from measurements at two fixed locations in the tank. The last two columns contain the mean wave height $\overline{H}$ and the height $\overline{\eta'}$ of the crest above the calm water level averaged over the highest 3 waves of the actual wave group encountered on the first leg of each test.

RESULTS

The whole test series with the Models F and G consists of more than 600 test runs. For the judgement of the safety against capsizing during each run the remaining area E_R below the calm water righting lever curve between the maximum heeling Φ_{max} and Φ_0 of vanishing stability was used. From all runs at the same conditions the mean value $\overline{E}_R$ and the standard deviations s were calculated. Thereby the values of E_R were set to zero, if the model capsized or heeled more than Φ_0. For the determination of the limit between safe and unsafe the following condition was used:

$$\overline{E}_R - 3 \cdot s = 0$$

This procedure is the same as in the earlier investigations. Figure 5 shows an example. There are drawn $\overline{E}_R$ and $\overline{E}_R - 3\ s$ as function of the

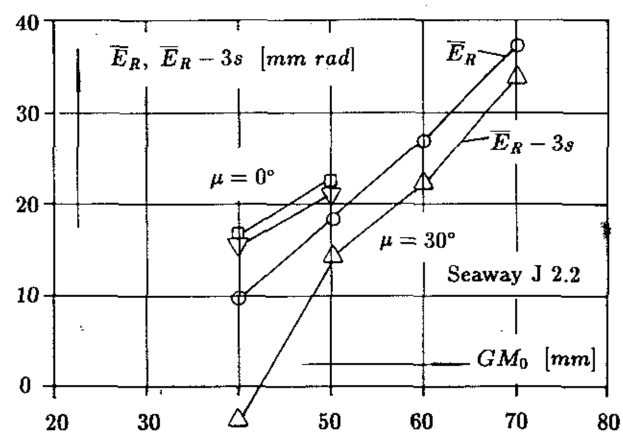


Figure 5 Mean value $\overline{E}_R$ and $\overline{E}_R - 3s$ as function of the metacentric height, Model G, draught 285 mm, Seaway J 2.2

TABLE 3
Stability at the limit between safe and unsafe for both models, Seaway P 1.2

Draught [mm]		285		222	
Model		F	G	F	G
GM_0	[mm]	98	37	94	29
KG	[mm]	329	294	345	299
Φ_m	[°]	31	35	32	40
Φ_0	[°]	63.3	62.8	65.3	67.8
GZ_{30}	[mm]	44.1	22.5	50.9	29.0
GZ_{40}	[mm]	40.0	22.9	47.5	33.6
GZ_m	[mm]	44.2	23.8	51.3	33.6
E_{30}	[mm rad]	13.40	6.14	13.42	5.92
E_{40}	[mm rad]	20.87	10.21	22.15	11.49
$E_{40} - E_{30}$	[mm rad]	7.47	4.07	8.73	5.57
E_0	[mm rad]	30.22	15.61	34.83	21.59

TABLE 4
Stability parameters at the limit between safe and unsafe for both models, Seaway J 2.2

Draught[mm]		285		222		185	
Model		F	G	F	G	F	G
GM_0	[mm]	108	42	112	33	136	29
KG	[mm]	319	289	328	295	327	310
Φ_m	[°]	32	37	33	41	38	37
Φ_0	[°]	67.4	65.7	71.5	69.6	74.3	66.7
GZ_{30}	[mm]	49.3	25.0	61.5	31.0	60.1	29.1
GZ_{40}	[mm]	46.3	26.2	60.5	36.1	65.0	34.0
GZ_m	[mm]	49.4	26.6	62.2	36.2	65.5	34.6
E_{30}	[mm rad]	14.84	6.80	16.39	6.53	17.39	6.02
E_{40}	[mm rad]	23.32	11.37	27.16	12.48	28.57	11.85
$E_{40} - E_{30}$	[mm rad]	8.48	4.57	10.77	5.95	11.98	5.83
E_0	[mm rad]	36.08	18.42	47.15	24.04	52.45	21.78

metacentric height GM_0. For the GM_0-value defined by the condition mentioned above righting lever curves valid for the limit between safe and unsafe can be calculated. Some usual stability parameters derived from the calm water righting lever curve are compiled in the Tables 3 and 4.

From the ealier investigations it is known that mainly the parameters GZ_m and E_0 are suitable to distinguish between safe and unsafe. The values of Model G are about 50 to 60 per cent of the values of Model F. The model with the large variation of righting levers in waves has to have much larger calm water righting levers.

COMPARISON OF STABILITY PARAMETERS FOR DIFFERENT HULL FORMS

The stability parameters at the limit between safe and unsafe determined for different hull forms cannot be compared directly. Different values depending on hull form properties have to be expected. In earlier investigations therefore a hull form factor C was introducted. The variance of the products of the weighting factor and the stability parameters is remarkable less than the variance of the parameters itself [1, 2]. The original factor is defined by

$$C = \frac{T \cdot D'}{B^2} \cdot \sqrt{\frac{T}{KG}} \cdot \frac{C_B}{C_W} \cdot \sqrt{\frac{L_0}{L}}$$

With the ratio C_B/C_W the factor contains already a part which should take care for the tendency to righting lever variations in waves. The linear dependency was taken because a better approximation could not be found because of the small variation of this ratio within the model series at that time. The idea was to use the ratio of the waterline coefficients of the mean waterline to the actual one as a measure for the tendency to righting lever variations in waves. For not rectangular main cross sections as at Model G therefore the factor has to be slightly modified:

$$\frac{\overline{C_W}}{C_W} = \frac{\frac{V}{T} \cdot \frac{1}{B_M \cdot L}}{C_W} = \frac{C_B \cdot \frac{B}{B_M}}{C_W}$$

Then for comparison of Models F and G a factor

$$C_1 = \frac{T \cdot D'}{B \cdot B_M} \cdot \sqrt{\frac{T}{KG}} \cdot \frac{C_B}{C_W} \cdot \sqrt{\frac{L_0}{L}}$$

can be used. Here the length L_0 is set instead to 100 m to a value $L_0 = 100/28 = 3.57\ m$ according to the scale ratio of the Model A, the first model we started with about 10 years ago.

The Table 5 contains the maximum righting levers GZ_m and the areas E_0 below the righting lever curve weighted with the hull form factor C_1 for all tested conditions.

TABLE 5
Stability parameters weighted with hull form factor C_1

Seaway	P 1.2		J 2.2		
Draught [mm]	285	222	285	222	185
$C_1 \cdot GZ_m$ [mm]					
Model F	4.31	3.48	4.90	4.32	3.50
Model G	3.20	3.34	3.61	3.62	2.65
"F"/"G"	1.35	1.04	1.36	1.19	1.32
$C_1 \cdot E_0$ [mm rad]					
Model F	2.95	2.36	3.57	3.28	2.80
Model G	2.10	2.15	2.50	2.41	1.67
"F"/"G"	1.40	1.10	1.43	1.36	1.68

The values for the Model F are in average 32 per cent higher than for Model G. That means that the hull form factor in the original form does not reflect the different behaviour of both models in a suffcient manner. Compared to other models the results for Model F are at the upper limit and for Model G at the lower limit of the spread of values as it can be seen in Figure 6. There are drawn the weighted maximum righting levers as function of a seaway parameter

$$P_{m0} = 4\sqrt{m_0}\,\Phi_{HH1}^{1/2}\,exp(-0.5(\overline{\lambda}/L - 1)^2)$$

introduced in [4]. The standard deviation around the regression line is 14 per cent of the mean value of all results compared to 33 per cent for the unweighted maximum levers.

By a better fit of the hull form factor the scatter can be reduced. For hull forms with larger variations of righting levers in waves the form factor must become relatively smaller. This can be done easily by a higher power of the ratio $\overline{C_W}/C_W$ because of simplicity preferably power 2. Tests with other values gave no better results. Then the adapted hull form factor C_N records as defined in Figure 7.

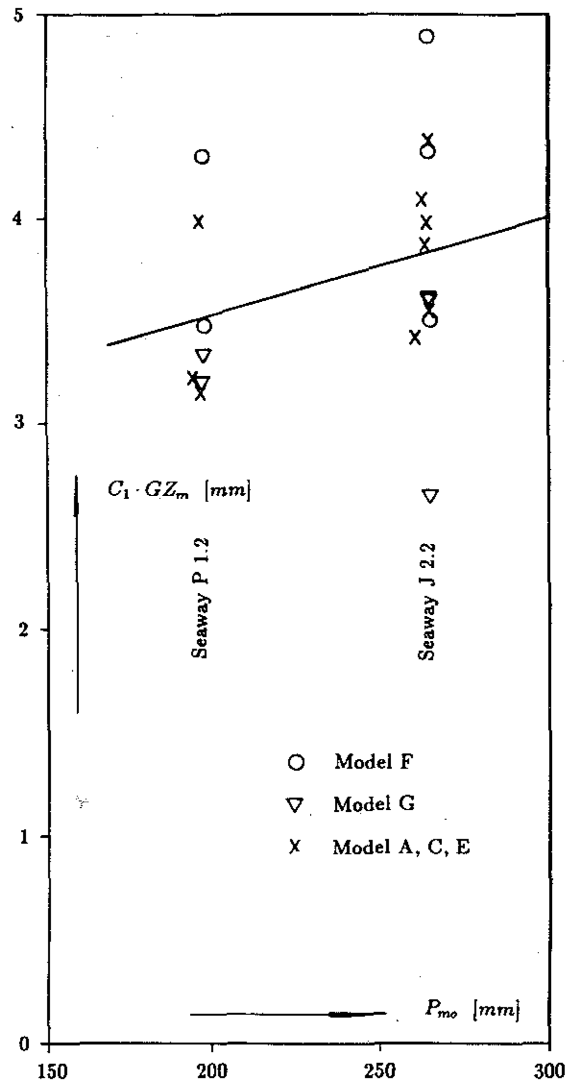


Figure 6 Maximum righting levers weighted with C_1 for different models and conditions

$$C_N = \frac{T\cdot D'}{B_M^2}\cdot\sqrt{\frac{T}{KG}}\cdot\left(\frac{C_B}{C_W}\right)^2\cdot\sqrt{\frac{L_0}{L}}$$

$$\begin{aligned} L_o &= 100\ m \\ L &\geq 100\ m \\ KG &\geq T \\ D' &= D + d\cdot\frac{2\,b - B_D}{B_D}\cdot\frac{2\cdot\sum l_H}{L}; \quad b \geq \frac{B_D}{2} \end{aligned}$$

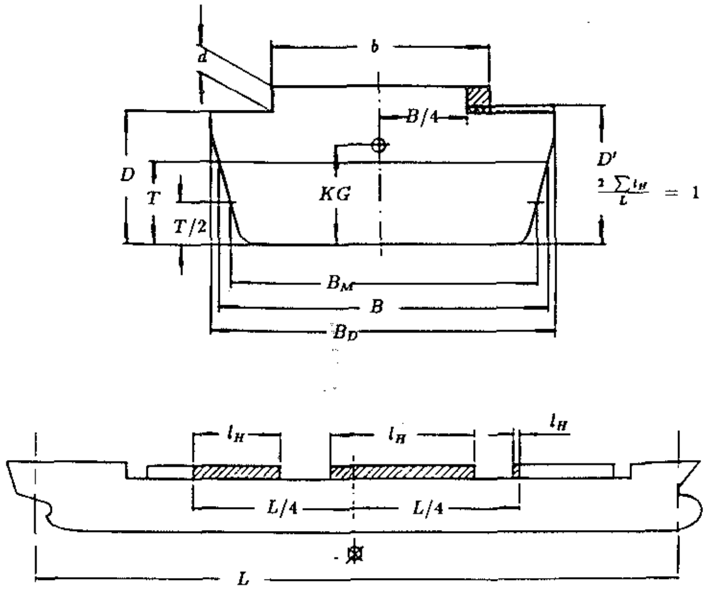


Figure 7 Definition of hull form factor C_N

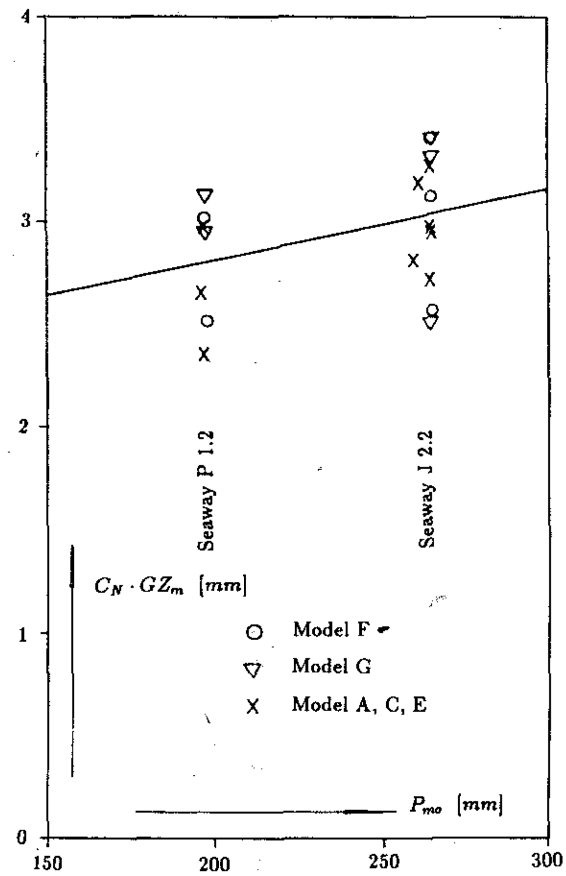


Figure 8 Maximum righting levers weighted with C_N for different models and conditions

The following Table 6 shows GZ_M and E_0 weighted with the modified form factor C_N.

Now the value for both models are in average the same. The standard deviation around the regression line as shown in Figure 8 is reduced to 10 per cent of the mean value. A further reduction can hardly be expected because of an unavoidable scatter of the test results in the same order.

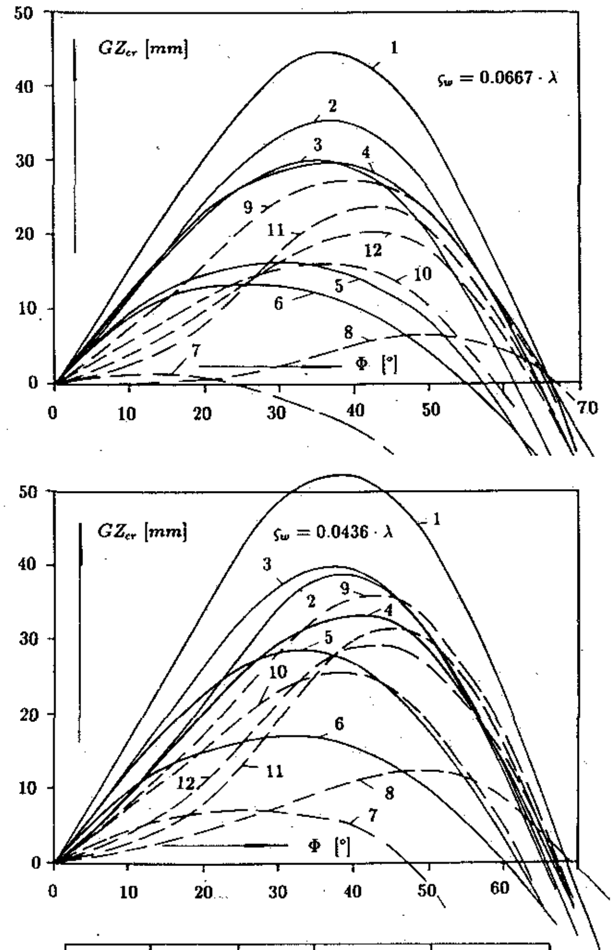
TABLE 6
Stability parameters weighted with hull form factor C_N

Seaway	P 1.2		J 2.2		
Draught [mm]	285	222	285	222	185
$C_N \cdot GZ_m$ [mm]					
Model F	3.01	2.51	3.42	3.12	2.55
Model G	2.94	3.14	3.31	3.41	2.50
"F"/"G"	1.02	0.80	1.03	0.91	1.02
$C_N \cdot E_0$ [mm rad]					
Model F	2.06	1.70	2.49	2.37	2.05
Model G	1.93	2.02	2.29	2.26	2.57
"F"/"G"	1.07	0.88	1.09	1.05	1.30

All considerations up to now are related to stability parameters derived from calm water righting lever curves. Because the danger of capsizing is mainly caused by the seaway it seems to be reasonable to look at the righting levers on a wave crest. A proposal given to IMO by Poland and the German Democratic Republic is based on such a hydrostatic righting lever on a wave crest [5].

The Figure 9 shows in the upper part righting levers on a wave crest calculated for a wave length λ equal to the length between perpendiculars and a height of $\lambda/15$. This steepness is often used by Germanischer Lloyd in such calculations. The curves for Model F and G at the same draughts are similar. But the large differences between different draughts indicate that hull form properties have to be considered, too. However it seems to be a hasty conclusion that these curves give a better basis for the judgement of the safety than the calm water righting lever curves. The results for the other models also drawn in Figure 9 show a wide varity and it is hardly imaginable to find common criteria from these curves.

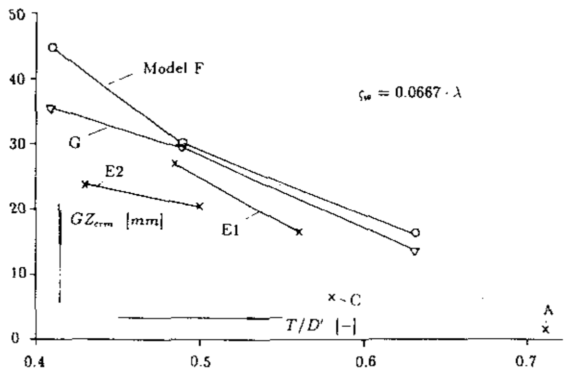
On the other hand the choice of the steepness of the wave is arbitrary. In the GDR-Proposal a wave height is used depending on the severity of the seaway which shall be determined following the concept of the effective wave by Grim [6]. For the Seaway J 2.2 the GDR-Proposal comes up with a wave height $\zeta_w = 0.044 \cdot \lambda$. For this



Curve	Model	T/D'	$GZ_{cr\ max}$ [mm] for ζ_w = $0.067 \cdot \lambda$	$GZ_{cr\ max}$ $0.044 \cdot \lambda$
1	F	0.41	44.5	52.4
2	G	0.41	35.4	39.0
3	F	0.49	30.0	39.9
4	G	0.49	29.7	33.4
5	F	0.63	16.3	28.8
6	G	0.63	13.5	17.3
7	A	0.71	1.5	7.3
8	C	0.58	6.5	12.5
9	E1	0.48	27.3	36.0
10	E1	0.56	16.3	25.6
11	E2	0.43	23.6	31.5
12	E2	0.50	20.3	29.3

Figure 9 Righting lever curves on a wave crest for the limiting KG in Seaway J 2.2

smaller value other curves for the righting lever on a crest are valid as shown in Figure 9 in the lower part. The curves for Model G don't change to much whereas the curves for Model F are quite different due to a strong dependency on wave height. The relative magnitude of the righting levers also compared to the other models changes with different steepness. For example in the steeper wave the smallest maximum lever on the crest is only 3.4 per cent of the largest one and in the smaller wave it is 14 per cent.



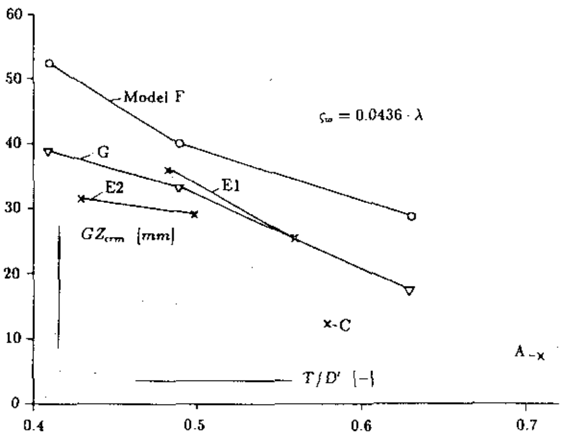


Figure 10 Maximum righting levers on a wave crest for the limiting KG in Seaway J 2.2 as function of the draught to depth ratio

The Figure 10 shows the maximum righting levers on a crest as a function of the draught to depth ratio T/D' for both wave heights. There is a significant trend to smaller values with increasing T/D'-ratio. But with this dependency alone the differences between the models cannot be explained. Also for criteria based on the righting levers on a crest the individual hull form characteristic has to be considered, but there is no idea how to do so at the moment. By the way in this figure also the influence of the wave height becomes evident. The values for Model F are in average 16 per cent higher than for Model G in the higher wave but 40 per cent higher in the smaller wave.

It can be concluded that stability parameters derived from righting levers on a crest itselves are not suitable for a fair judgement of the safety against capsizing. The hydrostatically calculated values do not reflect the physical phenomenon. Stability parameters derived from these curves therefore also can be seen only as comparative values which are not necessarily better than those derived from the calm water righting lever curves. They are rather not as good as the latter because of the dependency on the wave height which cannot be determined on a sound base. It seems to be more successful to take into account the tendency of a hull form to variation of righting levers in waves at the requirements for the calm water curve. Exactly this can be done using the modified hull form factor C_N.

CONCLUSIONS

During the last ten years eight models of seagoing merchant vessels were tested in the HSVA with variations of draught and sea state. In this way KG values at the limit between safe and unsafe could be determined for altogether 59 cases always with the same method. These data give a sound base for more general conclusions. This is unfortunately not true at the comparison of single results from different sources because there is a larger and in detail unknown influence of the conditions on the results. With these results the HSVA has an unique treasure of experience in the field of the safety against capsizing.

Already in an early stage the hull form factor C could be introduced evaluating the hull form properties regarding the safety against capsizing. In the basic elements this factor has been preserved up to today. The experience with more models led to further improvements which have been finished with the modification proposed in this paper.

In combination with usual stability parameters derived from the calm water righting lever curve the hull form factor C_N serves for a good equability of the weighted parameters. Thereby the parameters $C_N \cdot GZ_m$ and $C_N \cdot E_0$ are the best to distinguish between safe and unsafe. Further parameters were selected according to the IMO-Recommendation A 167. Herewith new stability criteria are at everybodies disposal which deal with the complex problem of stability assessment for a wide variety of hull forms much better than the criteria recommended in A 167.

Limiting values for the new criteria cannot be fixed directly from test results. Test conditions and the analysis has a influence on the absolute values. Therefore limiting values were derived from a statistical analysis of data from real ships in service [1, 2]. Now these limiting values have to be changed due to the modification of the hull form factor. The ratio C_B/C_W was set into power 2. Within the Models A to D at 3 draughts each this ratio showed a small variation with a mean value of 0.810 and a standard deviation of 0.038. Within the 143 vessels used in the statistical analysis the mean value was 0.848 with a standard deviation of 0.046. The new limiting values can be found by multiplication of the old ones with 0.80 to 0.85. Then the proposed criteria with the actual factor C_N are as follows:

$C_N \cdot GZ_{30}$	$\geq$	0.033	$[m]$
$C_N \cdot GZ_m$	$\geq$	0.042	$[m]$
$C_N \cdot E_{30}$	$\geq$	0.009	$[m\ rad]$
$C_N \cdot E_{40}$	$\geq$	0.016	$[m\ rad]$
$C_N \cdot E_0$	$\geq$	0.029	$[m\ rad]$
$C_N \cdot (E_{40} - E_{30})$	$\geq$	0.006	$[m\ rad]$

With the modification given in this paper the hull form factor concept can be seen as a fully ripened method. The new criteria are applicable for merchant vessels with a wide variety of hull forms. The proposed limiting values serve for a safety level which has been established with the IMO-Recommendation A 167 for traditional hull forms only. If the hull form properties are not taken into account at the assessment of stability there is the danger of a decreasing safety level with extreme modern hull forms.

LIST OF SYMBOLS

B, B_D, B_M	Breadth in WL, on deck and half draught
C, C_1, C_N	Hull form factors
C_B	Block coefficient
C_a	Waterplane coefficient
D, D'	Depth
E_0	Total area below the righting lever curve
E_R	Remaining area below the righting lever curve
GM_0	Metacentric height
GZ	Righting lever
KG	Height of centre of gravity above base
L_{oa}	Length over all
L_{pp}	Length between perpendiculars
m_0	Variance of the seaway
P_{mo}	Seaway parameter
s	Standard deviation
T	Draught
T_p	Peak period of the seaway
V	Volume displacement
ζ_w	Wave height
$\overline{\lambda}$	Mean wave height
Φ	Heeling angle

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DESIGN CONSEQUENCES OF PRACTICAL SHIP STABILITY CRITERIA

DR. İ. REŞAT ÖZKAN

In this paper, an account on the Concepts of "Rolling motion","Stability" and "Mechanism of Capsizing" is presented together with some design recommenda tions based upon the results of Practical Stability Analysis and its Applications. A discussion on the elements and formation of àn ad-hoc equation for a nonlinear forced rolling motion is given. Some considerations concerning the relevance and importance of roll damping moment is, also, discussed. Major finding of theoretical Practical Stability Analysis is outlined for the purpose of the completeness. A General discussion on the Applications of Practical Ship Stability Crieteria especially with a reference to the comparisons of the concepts of "Initial Stability" and "Stability in large Rolling angles" as well as a general discussion on design features of a ship for the stability and its efficiency points of view are included.

1) INTRODUCTION

Naval Architecture, like every other engineering branch, deals with the problem of why and, so how to determine the main dimensions of a system, the design of which are dictated, mainly, by

i- Economical targets
ii- Technical Requirements
iii- Safety Aspects.

The above listing does not imply an order of importance among themselves however, only apart from a few distinctions, reflects a common methology of the design problem of an engineering system.

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A ship, as an engineering system, is a very costly economical investment and its efficiency and profitability does, most certainly, depends upon its endurance, reliability or, more generally, its survival capacity. Unless the latters are satisfied, mentioning of the others would be of no significance as the existence of a ship would be always at risk.

At this stage it is worth mentioning that one may, for the purpose of clarity, find it useful to bring into the discussion of a conceptual misjudgement is making some conclusions i.e., "Resistance and Stability of a ship are of conflicting nature".This, to the present author, may not be true and is a consequence of our tendency of willing to pose some rules despite the limited information available with us in terms of the motion of a ship in a seaway and of the mechanism of capsizing, the instability. Indeed, the phenomena governed by the axioms of classical mechanics, that is to say the laws of motion, do not appear to become prosperous at the expense of the others however, the those stemmed from different viewpoints like share of cargo capacity of a ship in its total displacement and its strength are of conflicting nature.

In today's maritime world enormous efforts are being spent by a number of national and international advisory and regulatory bodies to achieve this end of safety standards and the same will, no doubt, be one of the major items of the international maritime agenda for many years to come.

2) HYDROMECHANICAL PROBLEM

In what follows we shall restrict ourselves to rolling motion only, however, the arguments may be considered to be covering the entire ship motions with some exceptions like assumptions concerning linearity of motion, kinematic body surface condition i.e., theoretical rigid body and similar. It, therefore, may be appropriate to examine the entire hydromechanical problem under some subtitles for the purpose of presenting a better definition.

2-a) Ship, a Dynamical System

In studying a forced rolling motion of a ship in waves for a stability analysis, first of all, two basic assumptions are made. They are, namely;

i- The frequency of external wave excitations is time-invariant i.e., constant.

ii- Ship's hull behaves as a theoretical rigid body.

The first assumption enables one to avoid the complexity of modelling the motion is a time-domain and hence to employ a frequency-domain analysis, instead.

The second one together with the above furnishes one with some means of confidently using an ordiniary differential equation and, furthermore, an important facility in the numerical evaluation of the solution of the hydrodynamic boundary value problem via the use of the kinematic rigid body surface condition by way of ignoring even every local infinitesimal deflections of the hull as well.

Under these general assumptions, the rolling motion of a ship may be easily derived from General law of motion, i.e.,

$$M\vec{\gamma} = \vec{F} \qquad (1)$$

Where;

$M\vec{\gamma}$ = Inertial forces

$\vec{F}$ = External forces

M = Generalised Mass moment of Inertia.

$\vec{\gamma}$ = Acceleration field.

$$A\,\frac{d^2q}{dt^2} + B.F\left(\frac{dq}{dt}\right) + C.G(q) = K(t) \qquad (2)$$

Where;

q : denotes, at present, a displacement

A,B,C : Some coefficients

K(t) : Time-dependent external excitations

t : time

2-b) Ship-Sea Interaction Phenomenon :

It is not the intention of this paper to present a detailed account on the hydrodynamical aspects of the interaction penomenon however, making an attempt to underline the nature of fluid damping (Roll damping) may be considered to be relevant for the explanation on the results of Practical ship Stability Analysis laterin the paper.

A ship, during being excited by an incident wave system, assumes some enegy onto itself from the environment and in consequence of which ship starts to perform a rolling motion.During this motion the surrounding fluid domain is disturbed and some of the kinetic energy of the ship is then returned back to the fluid domain. This, so-called radiated waves, are modelled at the infinity by a statical radiation condition which implies that the subsequent fluid reactive forces are conservative which can be derived from a potential field, i.e., the velocity potential. Based upon this assumption, outgoing radiated waves should do a work on the ship by means of the subject conservative force and that force act in the direction of limiting (damping) the rolling motion.Therefore it is referred to as the "Roll damping" and it is in phase with rolling velocity. As it is clearly seen, roll damping is a very important item in stability analysis as it, together with the potential energy gained during the rolling motion due to the anti-symmetric shape of the portion of the hull remained under the sea surface, tries to reduce the amplitudes of rolling angles. It is obvious that the potential energy is the integration of the righting moment which is in phase with rolling angle. After writing down the terms in the equation (2) by assuming that the body-related inertial force and abovementioned fluid reactive and active force are accurately calculated, it is seen that the eqution can not be satisfied, showing an important difference between the both sides. This difference is, apparently, in the dimession of force however, no further disturbance on the sea is present. Therefore it is in phase with the acceleration and referred to as the "Added mass moment of inertia"

In addition to what has been expained so far, viscosity effects tend to gain more importance in a large amplitude rolling motion of a ship and therefore a further, so-called, "Visous Damping" is taken into account in the equation.

Following the above argument one may choose the following ad-hoc equation for the nonlinear rolling motion of a ship.

$$(I+J)\ddot{x}+f_1(\dot{x})+g_1(x) = e_1(t) + \overline{WM} \quad (3)$$

Where;

I,J : Mass and added mass moments of inertia

$f_1(\dot{x})$ $= a_1\dot{x}+b_1\dot{x}|\dot{x}|$, Roll damping moment; $a_1, b_1 > 0$

$g_1(x)$ $= d_1x - e_1x^3$, Righting moment; $d_1,\ e_1 > 0$

$e_1(t)$ = Wave excitation

$\overline{WM}$ = Wind heeling moment

$x,\ \dot{x},\ \ddot{x}$ = Rolling angle, velocity and acceleration, respectively.

$(\dot{\ })$ = Derivation with respect to time.

Dividing each term by $(I+J)>0$, one obtains.

$$\ddot{x} + f(\dot{x}) + g(x) = e(t) + P \quad (4)$$

and by introducing a change of variable i.e., $\dot{x} = y$, the following phase (state) equations are obtained.

$$\dot{x} = y$$
$$\dot{y} = g(x) - f(y) + e(t) + P \quad (5)$$

3) CONCEPT OF STABILITY

Although the term "Stability" is commonly used in almost every branch of science there does not exist a universal definition covering all applications; it differs from one application to another according to the ultimate purpose for which the system is intended. As far as the engineering systems are concerned the term "Stability" means to confine the deviation or the errors from the original state of the system to a limited range so that the system can fulfil its task, satisfactorily, under the effects of initial disturbances, that is to say perturbations, which cause deviations from the original state. This definition of stability is virtually limited to free systems on which there exists no continuously acting excitations. In the absence of such continuous external excitations the system is required to return to its original state and that is referred to as "Asymptotic Stability".

In cases there exists continuously acting excitations upon the system which forms the subject of this paper i.e., there exists a wind heeling moment and time-dependent wave excitations causing non-linear rolling motion of a ship, the mechanism of the motion of system is entirely different so is the definition of stability. In these circumstances the ability of a system to survive despite the presence of continuous excitations can not be explained within the framework of the concept of Asymptotic Stability furthermore, Asymptotic Stability does not need to be assessed as this would not be required in studying this problem of stability. This can be best illustrated by saying that the returning to the original undisturbed state of the system is not possible as long as the excitations continue to exist. In other words, studying stability of a forced system i.e., forced rolling motion of a ship as it happens to be in our case, by asymptotical means is of no use and totally misleading.

Marine casualties caused by the capsizing of a ship are, unfortunately, frequently encountered and these capsizing incidents may be grouped in various types:

i- Capsizing of vessels during loading and unloading operations (mainly Ro-ro vessels and similar)

ii- Capsizing due to shifting of cargo during loading and/or unloading of cargo and during a voyage in a calm sea.

iii- Capsizing in waves.

iv- Capsizing due to a structual failure, in consequence of a collision case and due to water entry whilst the vessel is intact (Damage Stability-Water Entry)

The first two capsizing cases start and develop in consequence of the loss of initial stability characteristics and terminate as the ship can not assume a new equlibrium position during the increase of heel angles.

The third item is caused by the effects of external environmental excitations like waves and winds and occur during a rolling motion, in due course. In this case assuming a new equilibrium position after leaving the initial equilibrium state is not of any concern as this does not, practically, take place.

As it is clearly seen from above there happens to be different mechanism of roll, heel and capsizing and therefore it should not be expected that one set of criteria may apply to all types of abovementioned capsizing cases.

Damage stability may take place in both calm and wavy sea conditions and it should, therefore, by treated as per the above definitions with contributions coming from statical and dynamical effects of water entry into the hull.

Mathematical modelling of the ship's behaviour and its subsequent stability analysis must be so carefuly made that the model should reflect the physical aspects of the phenomenon at the most possible level and the criteria to be derived from the stability analysis should have a character permitting generalisations. Such a generatisation is inevitable if one requires to set up a criteria and this can only be achieved by using rather qualitative approach instead of a quantitative one which would only provide information for a specific input data.

4) FORCED ROLLING MOTION

Nonlineer forced pure rolling motion of a ship is, generally, given by the equation (3). The rolling motion does not necessary need to be a periodic system neither of a symmetric character in general and therefore, can be considered to be a non-autonomous system which is represented by the following vector equation.

$$\dot{\vec{x}} = \vec{f}(\vec{x},t) \qquad (6)$$

Where;

$\vec{x} = \left[x_1, x_2, \ldots\ldots, x_n\right]^T$, state vector

t : Time, s.t. $I = \left\{t \mid t \in \left[t_o, +\infty\right)\right\}$

$\langle\vec{x},t\rangle \in E^{n+1}_{\vec{x},t} = E^n_{\vec{x}} \times I$ (An Inner Product space)

$E^n_{\vec{x},t}$: n dimensional Euclidean space.

It is a well-known fact that, in most cases, rolling motion occurs in coupling with other modes of motion especially like "Heaving", "Pitching", "Yawing" particularly in the following and quartering seas. With this character, representing the rolling motion and the evaluation of hydrodynamic coupling terms make the problem even more difficult. In the present analysis only pure rolling motion is considered and all coupling effects are ignored.

5) PRACTICAL STABILITY

In view of various character of capsizing mechanism it is extremely important to choose or to pose the most appropriate definition of stability for the stability definitions available. Prior to mentioning about the definition of Practical Stability let us give the basic definition of stability which forms the starting point of all others.

Definition (Stability) : Let the system be defined by the vector equation i.e., $\dot{\vec{x}} = \vec{f}(\vec{x};t)$; $\vec{x} = (x, \dot{x}, \ddot{x}, \ldots\ldots)$. For a given arbitrarily small number $\varepsilon > 0$, i.e., $\|\vec{x}(t)\| < \varepsilon$, there exist a positive number $\eta > 0$ such that $\|\vec{x}(t_o)\| < \eta$ holds.

Above definition simply indicates that the problem of stabilty is, in fact, concerved to the problem of existence of a region of initial states (η) in view of the fact that ultimate values of displacement, velocity and acceleration should be less than required value, (ε).

As mentioned previously, among the various available definitions of stability, "Practical Stability" is, probably, the most appropriate one and defined as follows. Ref. (1)

Definition: (Practical Stability)

"Practical Stability is the Uniform Boundedness of the solutions with respect to the set of initial states and the family of excitations".

The interpretation of this definition in naval architectural terms states that the maximum prossible values of displacement, velocity and acceleration of the entire cycles of roll should be smaller than a set of required limits and the values of these limits should be independent of the initial time or the initial state (initial values of displacement etc.) from and with which the motion initiates and developes. It is not the intention of this paper to go into the details of the theory and applications of "Practical Ship Stability" as the same have been already publisheed elsewhere Refs.(2-3). However, for the completeness of this present account, the following summary might be appropriately accomodated within this paper.

The uniform boundedness of solutions of the equation (3) was first examined by using a theorem given by Yoshizava, Ref, (4) via the use of a time-dependent Lyapunov function. The analysis carried out in a sufficiently large was definition interval of the rolling angels and it was shown that the solutions would be uniformly bounded in range which is a subset of the definition interval provided that

$$\int q(x)dx > P.x+\int_0^t |e(t)|dt \qquad (7)$$

The interpretation of this inequality in Naval Architectural terms resulted in the evaluation of the open form of above which is nothing else in the sense what present IMO criteria states for the minimum angle of roll where $\overline{GZ}$ curve should reach to its maximum value, Figure 1.

$$x_m(=\emptyset_m) > \frac{\overline{WM}+\sqrt{\overline{WM}^2+10\Delta.\overline{GM}(E_1T'+5\Delta.S)}}{5.\Delta.\overline{GM}}$$

for positive rolling angles $x_m(=\emptyset_m)>0$

$$x_m(=\emptyset_m) < \frac{\sqrt{\overline{WM}+\ \overline{WM}^2+10\Delta.\overline{GM}(E_1T'+5.\Delta S)}}{5.\Delta.\overline{GM}}$$

for negative rolling angles $x_m(=\emptyset_m)<0$

Where;

$x_m=\emptyset_m$: The angle of roll on or after which $\overline{GZ}$ curve should assume its maximum value

Δ : Ship's displacement (Tonne)

$\overline{WM}$: Wind Gust moment (Tonne-Metre), Constant.

E_1 : $|\max(e_1(t))|$; (Tonne-Metre)

T' : Quarter roll period... (Seconds)

$\overline{GM}$: Initial Metecentric height (Metre)

Note : A new $\overline{GM}$ criterion will be proposed later in this paper.

S : Area under $\overline{GZ}$-$\emptyset$ curve. (radians-metre)

As it happens to be for every nonlinear system, stability is a local concept for rolling motion and the conditions of stability would apply only

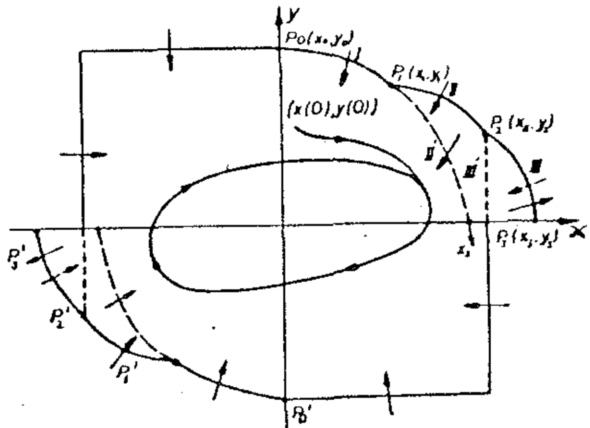

Figure 1- Domain of Practical Stability

in a region to be determined mainly, by the nonlineer features of the motion. Such a region of Stability was constructed in the phase-plane by using piece-wise continuous curves which are independent of initial time and that region is the domain of Uniform Boundedness, a Jordan domain. Whilst constructing the subject domain further conditions in terms of the comparisons of external excilations with damping and restoring characteristics were obtained.

6) A $\overline{GM}$ CRITERION :

Apart from the stability analysis hitherto summarised, a new $\overline{GM}$ criterion was devised by using the phase equations (5) and a requirement imposed on the analysis that rolling towards both sides should terminate on or before $\overline{GZ}$ curve assumes its maximum value. It was found that, Ref. (5)

$$\overline{GM} > \frac{1.89}{\Delta}\cdot\sqrt[3]{(E+\overline{WM})^2.e_1} \qquad (9)$$

Metacentric height, $\overline{GM}$, is in principle, a measure for the initial stability of rolling motion. However the present one deals with the stability at large rolling angles.

7) SAMPLE APPLICATION

Practical Stability has been applied to a number of already capsized and operational ships i.e., Fishing vessels, LPG tankers, Cargo ships and similar and the some results of application have been already published elsewhere. Refs. (3,6,7)

In this paper we shall content with presenting the results of an applicantion to an LPG tanker which already capsized.

Main Particulars :

$L = 55.5$ metres
$B = 9.0$ metres
$T = 4.25$ metres
$\Delta = 1122$ m.tons
$x_{max} = 0,567$ radians (32.48^{o})
$\overline{GZ}_{max} = 0,500$ metres
$\overline{GM} = 1.13$ metres
$I = 488.190$ ton.m/sec^2
$J = 217.917$ ton.m/sec^2
$\overline{GZ}(x) = l_o x + l_1 x^3 + l_2 x^5 + l_3 x^7 + l_4 x^9$

with;

$l_o = \overline{GM}$, $l_1 = 0,334911$
$l_2 = -5,280952$ $l_3 = 6,526840$
$l_4 = -2,517549$

Let us, now, apply the above results for various loading conditions :

Case-I

Wave Excitation : $E_1 = 33.842$ tonne-metres
Wind Excitation : $\overline{WM} = 23.858$ tonne-metres
Maximum
Righthing Moment: $\Delta.\overline{GZ}_{max} = 561$ tonne-metres

Intersection points of
Restoring and Excitations: $a_1' = 0,084$ radians
$a_2' = 1,057$ radians

Intersection point of
Damping moment and
Excitations: $b_1 = 0,272$ radians/sec.
$f_1(b_1) = 86.678$ tonne metre

The scalar values of the curves are:

$C_1 = 0,047$ $C_2 = 0,034$

Comparision of the values of metacentric height and the angles of maximum $\overline{GZ}$ values are as follows:

Present: $x_m = 0,56$ Radians
$\overline{GM} = 0,868$ metres (for initial stability)

Practical Stability:

$x_m = 0,9877$ radians
$\overline{GM} = 0,330$ metres (Stability at Large angles)

Case-II

$E_1 = 135.701$ tonne-metre
$\overline{WM} = 23.858$ tonne-metre
$\Delta.\overline{GZ} = 561$ tonne-metre

$a_1' = 0,233$ radians
$a_2' = 0,929$ radians
$b_1 = 0,458$ radians/sec.
$f_1(b_1) = 239.338$ tonne-metre

The Domains of Practical Stability are shown in Figures 2 and 3.

$C_1 = 0,282$ $C_2 = 0,190$
$C_3 = 0,377$

Comparison of x_m and $\overline{GM}$ values are as follows:

Present:

$x_m = 0,567$ radians
$\overline{GM} = 1,13$ metres
(For Initial Stability)

Practical Stability:

$x_m = 0,790$ radians
$\overline{GM} > 0,765$ metres
(Stability at large angles)

8) DISCUSSION OF APPLICATIONS :

It is seen from the above applications that the size and the type of domain of Practical Stability are decided by the magnitude of the excitations. In the first case the domain of practical stability is surrounded by the first two curves. This apparently shows that there exist no risk of capsizing whilst being subject to unimportant excitations (Figure 2).

This, also, indicates that maximum angle of roll can not exceed 0,383 radians (21.9 degrees). This result does not imply that roll angle should assume this value but only shows that even this maximum angle is attained the motion will, still, be reversed.

Depending upon the infinite number of configurations of ship and excitations, the motion path in the domain may reach to some roll angles and this angles may, still, be comparatively smaller than the abovementioned maximum angle of roll. After all, this is a qualitative approach, therefore, one should not expect a quantitave evaluation of motion's path.

In the second case Curves I and II do not cover the entire domain in the first quadtant and curve III emerges. It means that if an angle of roll shown in figure 3 is attained, the vessel will continue to have increasing rolling diplacement and will eventually capsize. Therefore one may call this angle as the "Angle of Capsizing". It is quite important to observe the fact that, even in this case, the restoring moment is substantially higher than the values of excitations however, the danger of reaching the capsizing angle is, still, present. This clearly shows that righting moment is not, by on its own, enough to reciprocate the rolling motion. Refering to the equation (3) one can easily see that acceleration

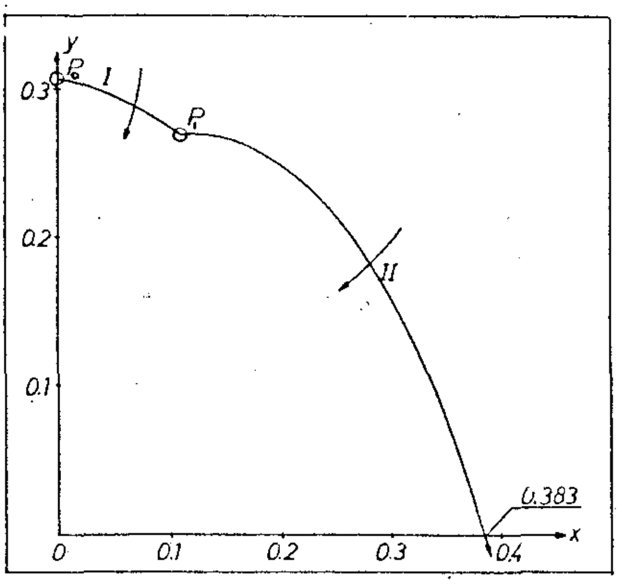

Figure 2- Domain of Practical Stability

values are of negative sign. The motion towards one side is being tried to slow down however, the magnitude of the negative acceleration is not sufficient to terminate the rolling motion i.e., to make the rolling velocity zero so that it starts to gain values of opposite sign, afterwards. It shows that rolling acceleration should start, much earlier, to assume values of opposite sign to that of rolling velocity during the rolling motion continues to progres in one direction and, magnitude of opposite signed acceleration should be large enough so that rolling velocity assumes opposite signs as well whilst the restoring moment is still higher that the excitations.

Following such a lenlgthy argument one may, now, say that values of roll damping is so important that, only by its support, the required opposite signed accelerations can be attined.

In fact, in some other applications of Practical Ship Stability it was demonstrated that the domain of stability could be constructed after by increasing the value of roll damping in the amount required by the theory whereas, originally, the domain did not exist beforehand with the actual values of ship.

The angle in the instances like in case-I can be named as "Maximum Attainable Angle of Roll" whereas for cases of the nature similar to the case-II should be regarded as "The angle of Capsizing". Not let us try to present discussion on the implication of the $\overline{GM}$ value i.e.,

$$\overline{GM} > \frac{1.89}{\triangle} \sqrt[3]{(E+\overline{WM})^2 e_1} \qquad (9)$$

As it has been already explained, this expression of initial metacentric height has been derived under the consideration of a forced rolling motion. Therefore it differs from the present one which is, generally, treated to be a measure of initial stability. The evaluation of present $\overline{GM}$ value is based upon the discussion that rolling motion should terminate at an angle (i.e.

Rolling velocity should become zero) where the maximum acceleration of the opposite sign should have been assumed.

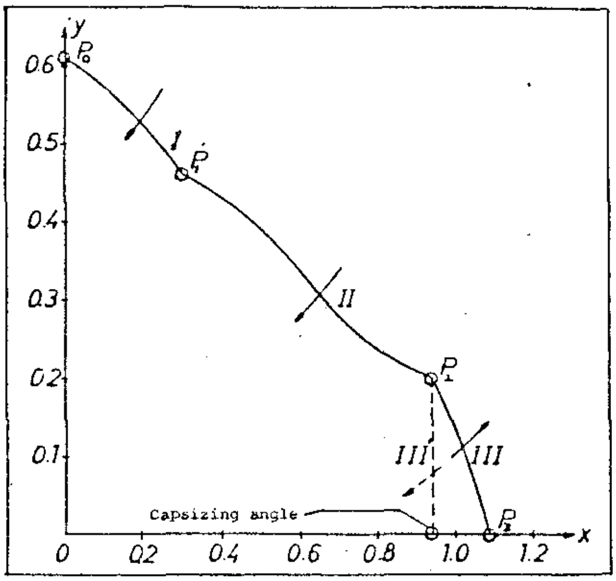

Figure 3- Domain of Practical Stability

Apparently,

this present $\overline{GM}$ value tries to limit the maximum angle of roll from above. In fact, maximum rolling angle during the motion may become any value within the range of $a_1' \times a_2'$. Therefore it might be the best solution to have this happens where $\overline{GZ}$ curve assumes its maximum value so that the maximum magnitude of acceleration of the opposite sign can be obtained. Even in cases where there is a risk of emergence of the third part of the curves i.e., $\widehat{P_2P_3}$ (Curve III) this present $\overline{GM}$ value can have a control over the rolling motion.

9) GENERAL DISCUSSION :

1- The uniform boundedness, Domain of Practical Stability and Lower limit for a $\overline{GM}$ value have been considered together, in assesing the Stability of a Forced Rolling Motion of a ship.

2- Analysis on Uniform Boundedness of the solutions yielded to a lower limit for the rolling angle where $\overline{GZ}$ curve assumes its maximum value. This requirement is entirely in agreement with Present IMO regulations i.e., 30 degrees, however, appears to be a more realistic one as it takes the main characteristics of a ship as well as environmental conditions.

3- Formation of a domain of practical stability resulted in a significent account on the comparisons of enviromental excitations with restoring and damping moments.

4- The above, also, yielded to a possibility of making a qualitative evaluation of rolling motion within the same domain.

5- It has been demostrated that ships become quite vulnerable against capsizing after the rolling angle of maximum $\overline{GZ}$ value i.e., restoring moment again starts to become smaller than the excitation terms.

6- Emergence of the Curve III indicates a divergent character of rolling motion if reaches to that particular angle which was described to be capsizing angle. In such cases an additional damping moment in the amount shown in figure 1 is necessary, perhaps, by providing a roll damping mechanism to be fed by the values of angle and velocity of rolling motion.

7- The results of the theoretical and numerical work (2), (3) have, various time, demonstrated that the $\overline{GM}$ value should be also bounded from above. In other words $\overline{GM}$ value should be taken in both lower and upper limits. This important result have not been numerically applied here however, other finding like $\overline{GM}$ and x_m also imply this situation. It was interesting to observe that all vessels appear to have excessive $\overline{GM}$ and insufficient x_m values. It is, also, interesting to note here once more that this feature of

Practical Stability completely agrees in principal, today's IMO criteria. It is also worth of withdrawing the attention to the fact that ships are, generally, designed with excessive $\overline{GM}$ values despite IMO's recommendation of 20 cm. for a minimum $\overline{GM}$.

8- To achieve large $\overline{GM}$ values by way of increasing beam/draft ratio does not appear to be reliable approach. For a ship of specified tonnage, the potential energy, i.e., the area under restoring moment ($\Delta.\overline{GZ}$) curve does not vary in great amount by changing the dimesions of a ship. Hence obtaining a required sufficiently large $\overline{GM}$ value by increasing the beam/draft ratio will confine the $\overline{GZ}$ value to a narrower band of rolling angle and cause to end up with higher $\overline{GZ}$ curve in that range. However it should be bear in mind that capsizing is an incident starts and develops at large angles roll and therefore, this approach of designing vessels with large beam values would yield to weak stability characteristics. It can, therefore, be said that initial stability and stability at large angles are of conflicting nature and designing vessels with higher initial stability standards worsen the stability characteristics at large angles.

At this stage one, may, further recommend the followings for the design considerations.

1- Beam/Draft ratio should be reduced by means of increasing Length/Beam ratio rather than reducing Length/Draft ratio.

2- Required metacentric height $\overline{GM}$ should be assessed by decreasing the vertical distance of centre of gravity i.e., $\overline{KG}/D$, rather than increasing metacentric radius, $\overline{BM}$.

3- In hull form design circular-shaped cross sections and hulls with large parallel bodies (ratio of length of parallel body to ship's length) must be avoided as possible as one can. This can be assessed by introducing longer fore and aft sections and increasing Length/displacement i.e., $L/\Delta^{1/3}$ ratios especially for small size vessels, i.e. fishing boats. In other words prismatic coefficient should be increased by having rather fine forms and more lengthy vessels.

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HYDRODYNAMIC PHENOMENON GENERATED BY BULWARK SUBMERGENCE AND ITS INFLUENCE ON SHIP SUSCEPTIBILITY TO CAPSIZING

Stefan Grochowalski [1]

A comprehensive study of the mechanism of ship capsizing in heavy seas is being conducted at the National Research Council Canada. The focus is put on the behaviour of small fishing vessels in quartering extreme waves. The paper presents some detailed analyses of model experiments. The composition of ship motions during the whole cycle of a quartering wave passing the ship length, and the hydrodynamic forces, exerted on the hull, are identified. The majority of the paper is dedicated to analysis of the influence of bulwark and deck edge submergence on a ship propensity to capsizing. The hydrodynamic phenomenon and the subsequent couplings and heeling moments created in such a situation are discussed.

INTRODUCTION

The solution of the problem of ship safety against capsizing is not achievable without full knowledge of the physics involved in the capsizing process. The mathematical model, which has to be developed, must represent adequately a general case of ship behaviour in extreme waves and has to be validated against reliable data. Therefore a fundamental study of the physics of the capsizing process should be undertaken first, in order to lay out a sound scientific base for any future stability regulations. With this in mind, the research program which was commenced several years ago at the Institute for Marine Dynamics of the National Research Council of Canada and aimed at formulating stability criteria for fishing vessels, was focused in its first phase on the study of the mechanism which brings a ship to capsizing in quartering, extremely steep and breaking waves. The program contains theoretical studies, model testing and development of numerical simulation codes. [1]

As the experiment data are considered to be the main source of information on the complex physics of the capsize phenomenon and at the same time the best base for the validation of the theoretical formulations, the model testing was specially designed in order to provide this necessary data.

The philosophy of the experimental approach, the test technique developed, and the program of experiments performed have been presented in [4]and [5]. The work is not completed yet. The detailed study of the test results is still ongoing. Part of the analysis of the kinematics and dynamics of ship motions in quartering breaking waves, as well as analysis of various types of capsizing have been reported in [5], while the first version of the computer code was presented in [6].

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Although the experiments were performed with the model of a small fishing vessel, the adopted methodology and the results provide insight into the mechanism of capsizing process and the findings have a more general sense. They can be applied to other classes of ships.

This paper presents a hydrodynamic phenomenon which has been revealed during the experiments and confirmed by the analysis of the test results. The phenomenon can be created when the bulwark and part of the deck become submerged during dynamic motions of a ship in quartering breaking waves, and it can bring to capsizing a vessel which, according to the existing criteria, may be considered safe. It is believed that because of the dramatic influence on ship susceptibility to capsizing, these hydrodynamic effects should be brought to the scientists' attention at first.

MODEL TESTS

A typical small Canadian hard-chine stern trawler of 18.6 m length of waterline was selected as a subject of the experimental studies. The body lines are presented in Fig. 1. The tests were carried out at the SSPA maritime Consulting AB (Sweden).

The model in the scale 1:14 was tested for two loading conditions: port departure (condition I) and full load (condition II), with two different stability characteristics for each. The righting lever curves for each of these conditions are given in Fig.s 2 and 3. Conditions I/A and II/B satisfy the IMO stability requirements while I/B and II/A do not.

According to the philosophy developed, the model test program consisted of:

Free-running tests - with the objective of investigating the dynamics of motions and capsizing at various loading conditions, various breaking waves and forward speeds;

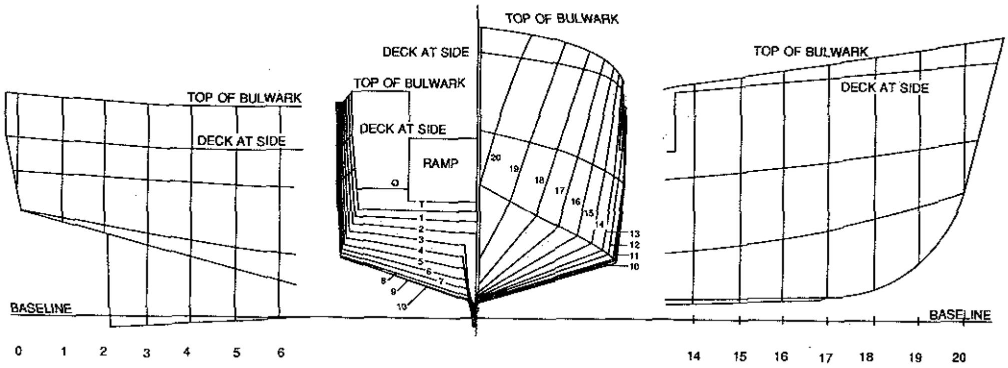

Fig. 1 Body lines of tested model

Captive tests (fully and partly captive) - with the objective of identifying the hydrodynamic forces exerted on a ship by extremely steep and breaking waves for various wave directions, forward speeds, heel angles and drift velocities.

In order to provide the possibility of reconstructing the capsizing mechanism, the free-running and captive model tests were correlated so that for any instantaneous position of the model with respect to the wave profile in the free-running situation, the appropriate situation in the captive tests could be found and as a result the composition of the hydrodynamic forces can be interpolated. This was made possible by use of video-recording in which the time counter was synchronized with the time base of the main data acquisition system.

As many as 440 runs all together were performed in the experimental program. The results were recorded in the form of time histories of motion components or hydrodynamic forces, and in the form of video-records.

SHIP BEHAVIOUR IN QUARTERING WAVES

When a ship is moving in large quartering waves, it performs a very characteristic composition of motions. A typical example of such a composition is given in Fig. 4 which presents a fragment of time record of motions during one wave cycle in the free-running model test. The vertical lines indicate the time points at which the model was in a wave trough (T) and when the wave crest reached the after perpendicular (AP), a quarter of the model length ($^1/_4$ L), the midships (O), three quarters of the length ($^3/_4$ L) and the forward perpendicular (FP). In order to facilitate the analysis, the time history of Fig. 4 has been transformed into a sequence of situations occurring for these time points, and is presented in Fig.5. The numerical values indicate the instantaneous position of the model in the adopted reference system (roll Ø, pitch θ, 2yaw ψ, and heave Z), while the vectors represent the instantaneous velocity in each mode of motion.

The motions in the wave trough are a result of the action of the back slope of the previous wave. Surge and sway velocities are directed toward oncoming wave, yaw turns the bow closer to a position perpendicular to the wave crests. The ship is essentially without a trim but in a large weatherward heel angle, and just starts recovering from this heel.

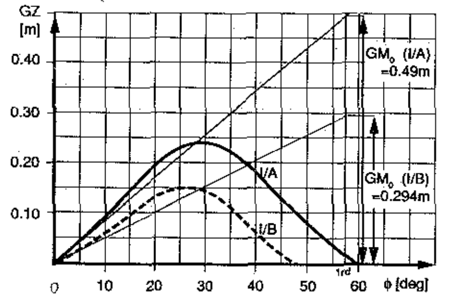

Fig. 2 Righting arm curves for tested port departure conditions

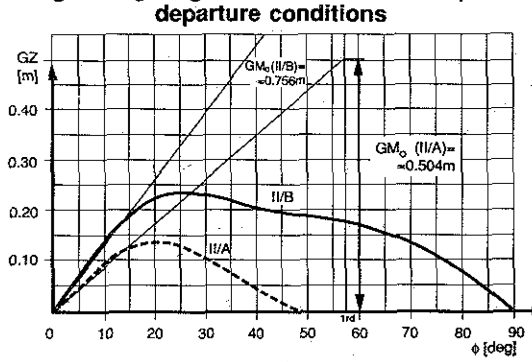

Fig. 3 Righting arm curves for full load conditions

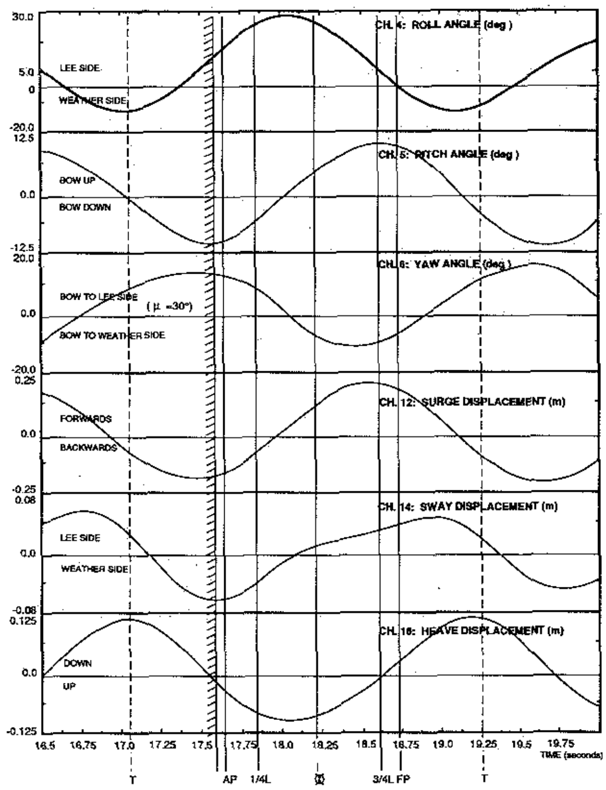

Fig. 4 Time record of motion components during one cycle in quartering waves. Free-running model test No. 13

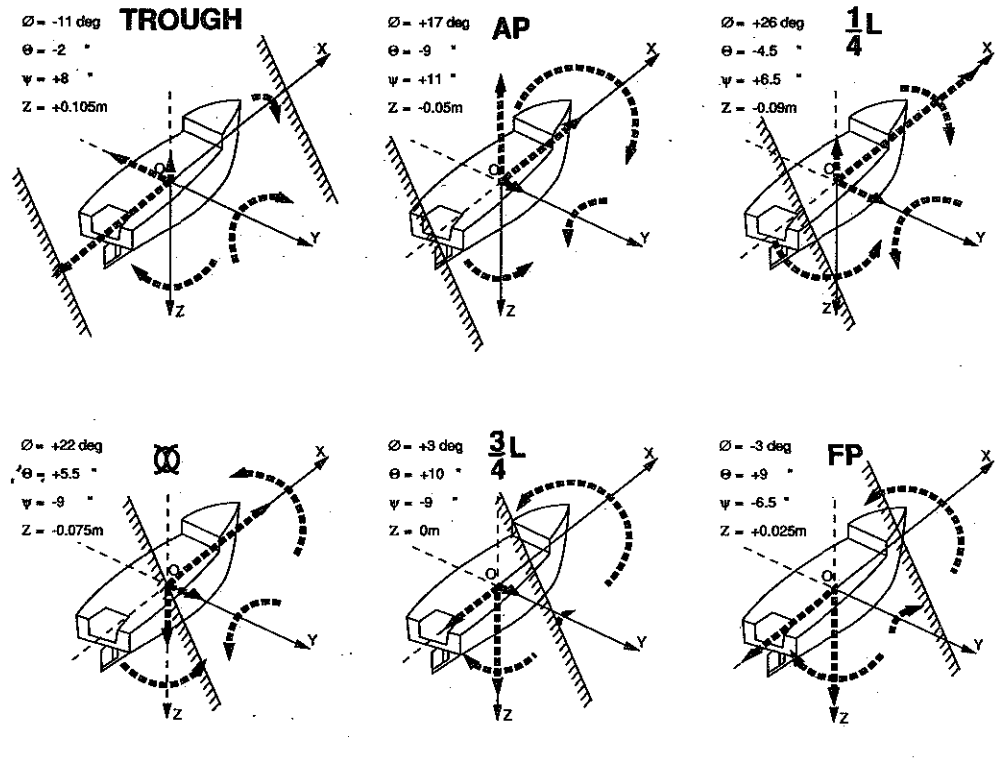

FIG. 5. Motion components in quartering waves (Free Run No. 13)

On the front slope of the oncoming wave the ship moves upward and gets increasing trim by the head. The roll motion becomes very dynamic and the ship changes its heel position from the weather to lee side.

As a result of a wave impact on the stern, the ship is dynamically pushed forward and aside (leewards) and acquires a large leeward heel. The stern, which is being pushed violently toward the lee side, causes turning of the hull towards beam position (see the motions for AP and $^1/_4$ L). Advancing of the wave crest toward the midships increases further the forward speed and leeward sway, while the roll continues to increase the heeling to the lee side. The combination of sway and yaw causes a large lateral motion of the after body toward the lee side while the ship is in large leeward heel angle and the wave crest is at this part of the hull.

When the wave crest approaches the midships the ship starts to recover from the maximum leeward heel if the bulwark was not submerged. Large yaw, leeward sway and surge forwards are still maintained.

After the wave crest passed the midships (about $^3/_4$ L) the directions of yaw and surge are reversing and now the bow is being pushed towards the lee side and upwards. Dynamic roll toward weather side develops.

On the back slope of the wave (FP) the hull moves down, turning toward "following waves" position, and acquiring significant surge backwards. The roll continues to heel the ship towards the weather side. Shortly, the ship gets to a wave trough and a new cycle of motion begins.

The presented composition of motions is consistent with the hydrodynamic forces generated on the hull in quartering waves. An example of a composition of forces and moments in a corresponding partly captive test is given in Fig. 6 (forces = solid line vectors). It is worth noting that the roll moment M_x, which on the front slope of the oncoming wave acts as a heeling moment and tends to heel the ship to the lee side, changes its direction and becomes a restoring moment after the wave crest passed approximately a quarter of the hull length. In the case of the tested ship form, this moment reaches its maximum value when the wave crest is at the midships.

THE HYDRODYNAMIC PHENOMENON GENERATED BY BULWARK AND DECK EDGE SUBMERGENCE

The characteristic composition of the motions in quartering waves when the wave crest is travelling between the stern and about three quarters of the hull length is very unfavourable from the capsizing point of view. As a result of fast yaw and sway motions to lee side with a simultaneous dynamic leeward roll after a wave impact, the after part of the deck edge at the leeside is moving down and sidewards, attaining a large lateral velocity. At the same time, the wave crest is moving forward increasing the dynamics of the lateral motion of the after part of the hull, reducing the restoring moment, and increasing a chance of the bulwark and deck edge immersion at the lee side. The increase in forward speed, caused by the wave impact, makes the duration of this dangerous situation relatively long.

In the previously analyzed case, the bulwark remained unsubmerged during the

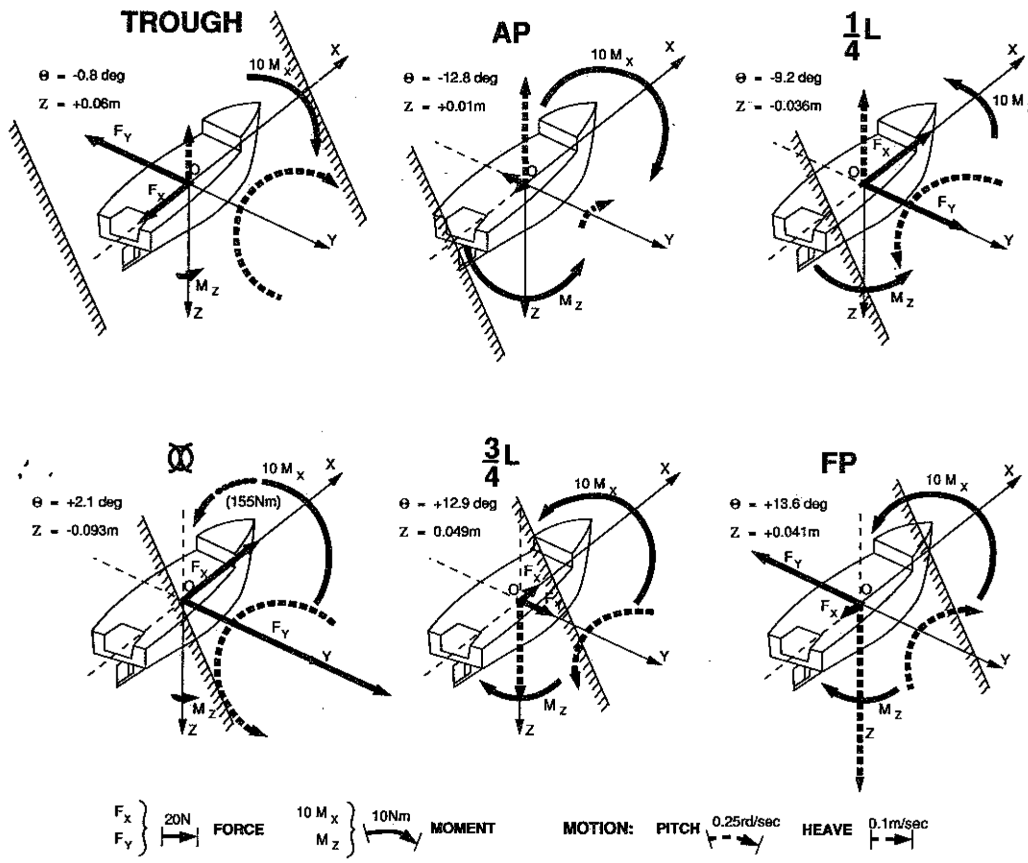

Fig. 6. Forces and motions in a semi-captive model test (Run No. 225)

whole cycle of wave action. If the bulwark and, as a result, part of the deck became submerged, the ship is in danger of capsizing.

The bulwark and deck edge submergence causes radical alterations in the roll motion. During the rolling a hydrodynamic reaction R is generated (Fig. 7A) on the immersed part of the deck and bulwark which constitutes a resistance of the surrounding water to the hull motion. This reaction creates a moment δMx which counteracts the restoring moment. The resistance to motion of the immersed part of the deck may be so large that it prevents the ship from the usual rolling back to the vertical position. Such effects were observed during decay tests with forward speed if, initially, the bulwark was submerged (fig. 7B).

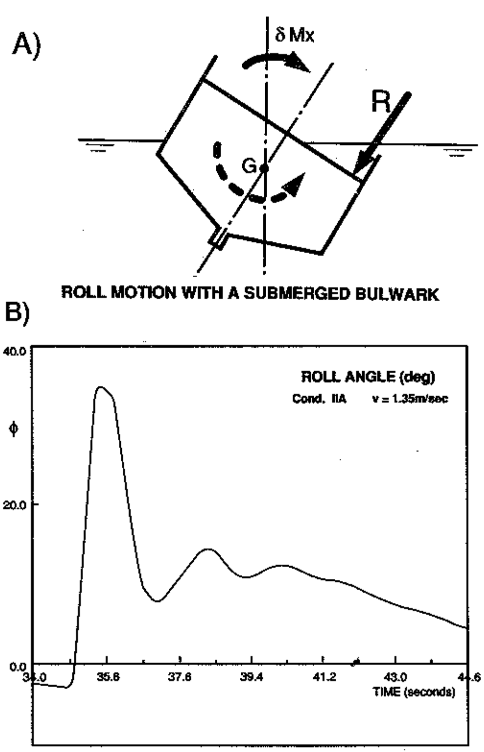

Fig. 7 Decay test with the bulwark submerged

The influence of bulwark submergence becomes much more emphatic if the ship executes lateral motions towards lee side. As a result of a lateral movement of the hull the submerged part of the deck is being forced to plough under the water. The resulting pressure on the submerged part of the deck generates a hydrodynamic resistance to the motion and the resultant reaction R (Fig. 8A) creates an additional moment δMxs which tends to increase the heel angle. If, on the other hand, the weather side bulwark becomes submerged and the lateral motion is leewards, the hydrodynamic reaction on the immersed part of the deck is much smaller (Fig. 8B) and does not create any significant danger. It may prevent, however, the ship from returning to the up-right position and, as a result, the ship may roll about the heeled position (so-called pseudo-static angle of heel). The difference between Fig. 8A and 8B cases provides some explanation of why, during some reported model tests in beam waves, [2,3] the low - freeboard model which was heeled to the weather side a8nd systematically subjected to water shipping on deck, never fully capsized to the weather side during the tests in beam waves, but did capsize on a next wave if, accidentally, a large wave impact heeled the model with the water on deck to the lee side.

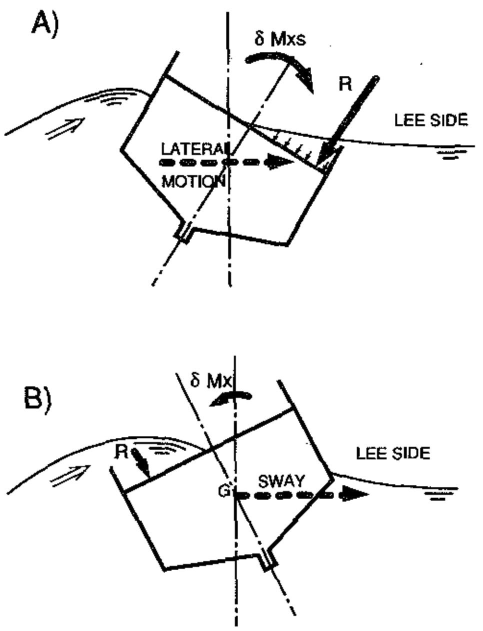

Fig. 8 Hydrodynamic effects generated by lateral motion during bulwark submergence

The hydrodynamic effects created by the underwater ploughing movement prevents the bulwark and deck edge from coming out of the water. This causes local restraints to the hull motion. The stiffness of this restraint depends, first of all, on the lateral relative velocities of the surrounding water and on the size of the immersed part of the deck. If, simultaneously with a fast lateral motion and bulwark submergence, the ship is forced by the wave to heave up, the restrained deck edge causes the hull to turn about a longitudinal axis located close to the bulwark and a pivot-like effect occurs. This creates a strong coupling between roll, lateral motion, and heave, increasing strongly the heeling moment (Fig.9).

The restraint of the motion of the submerged bulwark and deck edge has also another negative influence on stability safety. If this restraint lasts long enough and causes the ship to remain in a heeled leeward position at an angle ϕ^* (Fig.10) until the next wave crest reaches the hull, then the potential restoring energy of the ship is significantly reduced. Assuming that the GZ curve reflects, to some extent, the restoring potential energy of the ship, the new zero level (O') of this energy is established due to the heel angle ϕ^*. It can be seen that only a significantly smaller wave can be counter balanced by the ship in this configuration.

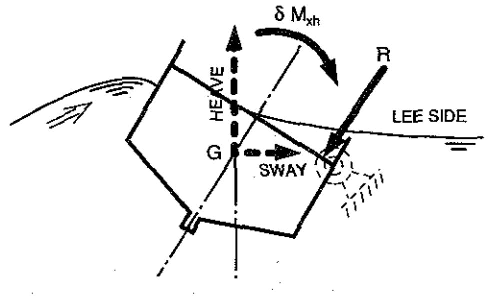

Fig. 9 Pivot-like effect caused by bulwark submergence, lateral motion and heave

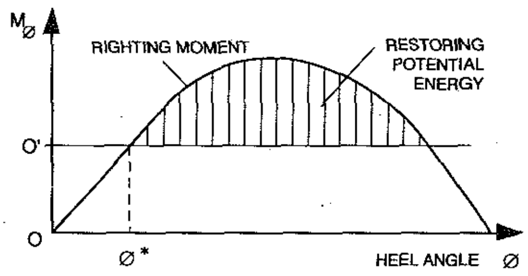

Fig. 10 Reducing effect of bulward submergence on ship potential restoring energy

Furthermore, if the ship remains in the restrained heeled position, the initial conditions of the next wave action are altered. The whole energy of wave impact is applied to the ship with the bulwark already submerged. As a result, all the negative effects generated on the submerged part of the deck are significantly enhanced, and the phenomenon lasts much longer than during the first wave action. As a result, the leeward heel angle increases further, threatening the ship with capsize.

In order to investigate the hydrodynamic phenomenon created by the bulwark/deck edge submergence, some of the captive tests were carried out with a lateral leeward drift and the hydrodynamic forces were measured for various combinations of drift velocities and leeward heel angles. The tested situations correspond to that presented in Fig. 8A. An example of the results of the partly captive tests for two different fixed heel angles: 20 and 45 deg. to lee side, are presented in Fig. 11. At the heel of 20 degrees the bulwark and deck edge were submerged when the wave crest was between the AP and $^3/_4$ L. At the heel of 45 deg., the leeward bulwark and deck edge were deeply submerged throughout the whole action of the quartering wave. The graphs present the time histories of the moment Mx during one cycle of a wave action. The position of the wave crest with respect to the model (Trough, AP, L/4, O, 3/4L, FP, T) is marked by vertical lines.

It can be seen that for 20 deg. heel angle the roll moment is all the time positive which, according to the adopted reference system, means that it is a restoring moment. When leeward drift occurs, the moment is decreasing. The larger the drift velocity, the larger the reduction of the restoring moment. This confirms the generation of an additional heeling moment which counteracts the restoring action of the Mx moment . At a certain velocity of the lateral motion, the roll moment becomes a heeling moment throughout the whole cycle of the wave action.

An even more dramatic change occurs for the larger heel angle (45 deg) as the leeward bulwark remains submerged all the time.

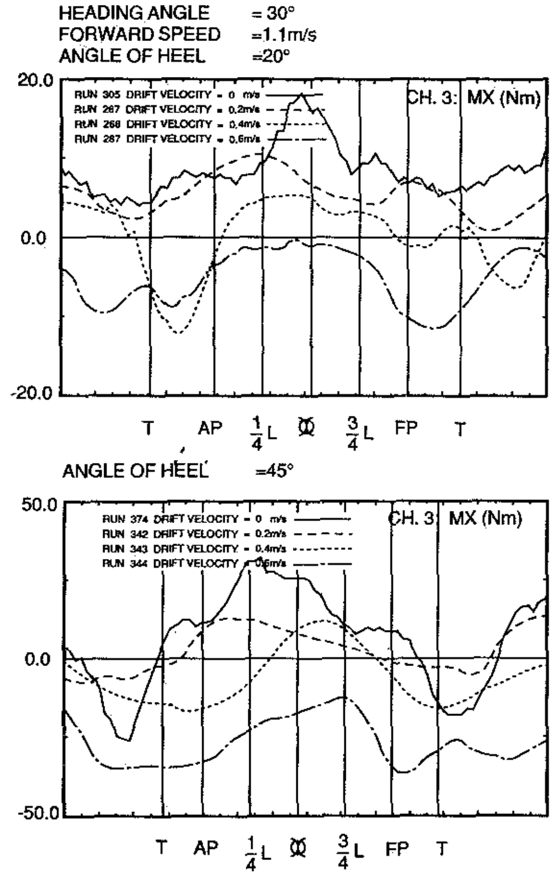

Fig. 11 Influence of bulwark submergence and lateral motion on roll moment in quartering waves (semi-captive model)

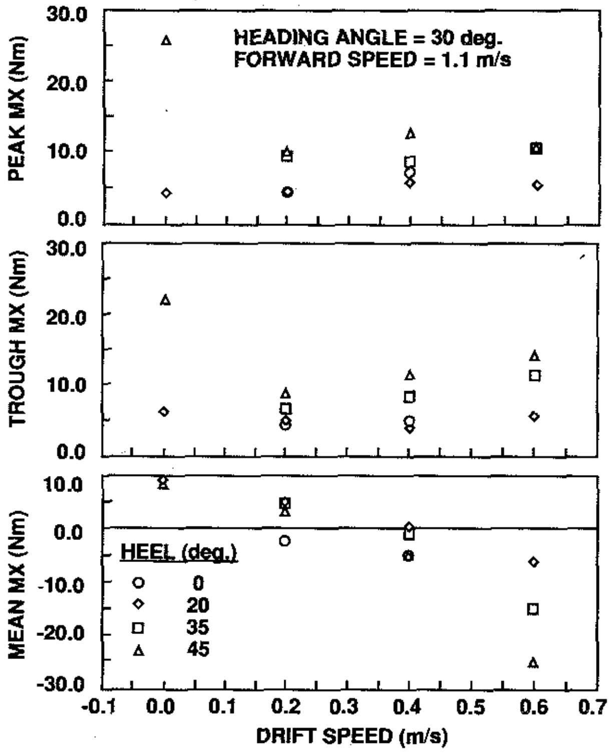

Fig. 12 Influence of drift speed (semi-captive tests, regular waves)

The influence of drift velocity on the total roll moment is also shown in Fig. 12, where the moment Mx is presented as a mean value and the responses to the action of wave crests and wave troughs (measured with respect to the mean Mx). If there is no drift, the mean moment at the tested heel angles is positive. When the leeward drift occurs the mean value of Mx is shifted toward negative values. The larger drift and heel angle, the larger the shift of mean Mx. The change of Mx is very dramatic particularly for larger heel angles, as larger part of the deck remains deeply submerged through a longer time period. For instance, at the heel angle 45 deg. the mean Mx changed from about +8Nm without drift (restoring moment) to -26Nm (heeling moment) at the drift velocity 0.6 m/sec! The difference between these two values (i.e. 34 Nm) is a measure of the heeling moment generated by the lateral motion of the hull when the bulwark and part of the deck are submerged.

Clear evidence of the discussed phenomenon can also be found in the time histories of the free model runs. Fig. 13 presents a fragment of a run at the light load condition (I/A) with the formally sufficient GZ curve. The full scale nominal parameters of this test were: wave height = 8.0m, wave period = 6.4 sec and the vessel's speed v= 8 kn. The time when the bulwark on the lee side is submerged is marked as a horizontal thick line at the bottom of the time histories of yaw, sway and heave. The horizontal lines above these graphs mark the time intervals when sway, yaw and heave are conducive to the generation of the additional heeling moment, that is: for sway and yaw - the direction from the weather to lee side, for heave - upwards. Then all these lines are collated on the record of roll motion. The dashed line represents the anticipated roll motion if the bulwark does not become submerged. Similarly to Fig. 4 the position of the wave crest with respect to the model are marked by vertical lines (T, AP, $^1/_4$L ...)

After the impact of the first wave, part of the bulwark became submerged and the characteristic composition of sway and yaw started to create the additional heeling moment due to bulwark and deck edge submergence. As a result, the model was heeling further instead of coming back to the up-right position. (point A). Although the moment due to the wave is a restoring moment after the crest passes the midships, the model remains at a large heel angle as long as this hydrodynamic phenomenon generated by sway and yaw on the submerged part of deck, exists (point B). When these two motions change their directions the model starts to recover and gains large angular velocity. Even though at a certain time point heave motion became conducive, the lack of lateral motion meant that the discussed mechanism did not appear. The energy of the returning roll was balanced by the heeling energy of the next wave and the model started rolling again to the lee side. But this time the bulwark was essentially above water and, although there was some conducive sway motion, the additional hydrodynamic coupling was not created. The model rolled back to the opposite side.

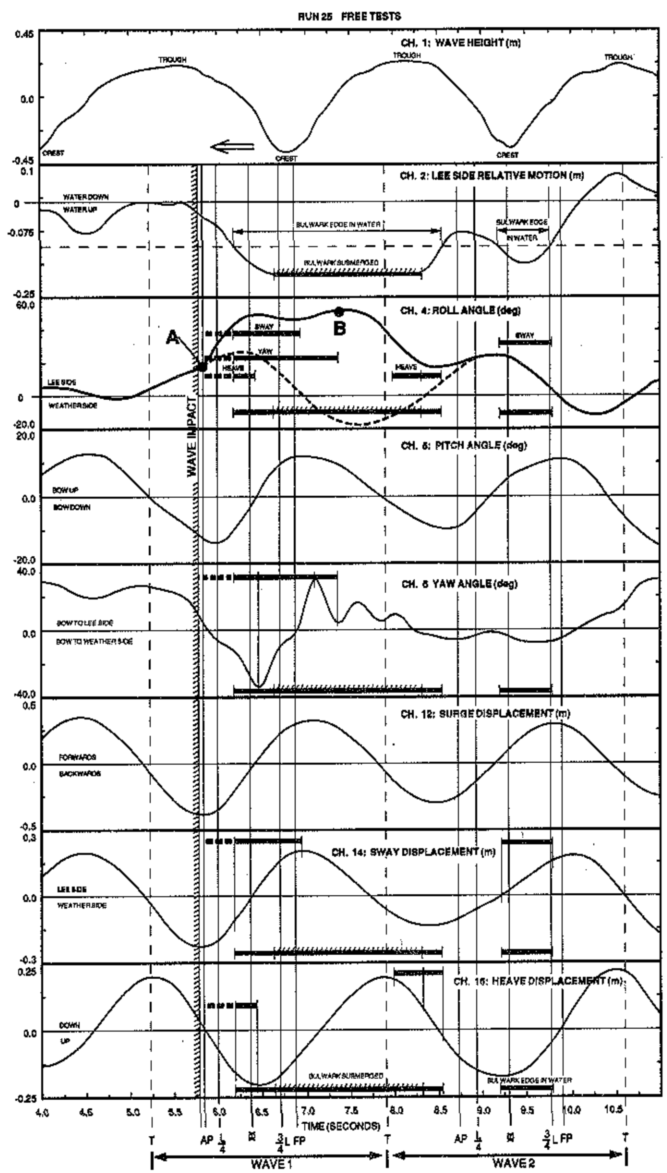

Fig. 13 Influence of hydrodynamic phenomenon created by bulwark submergence

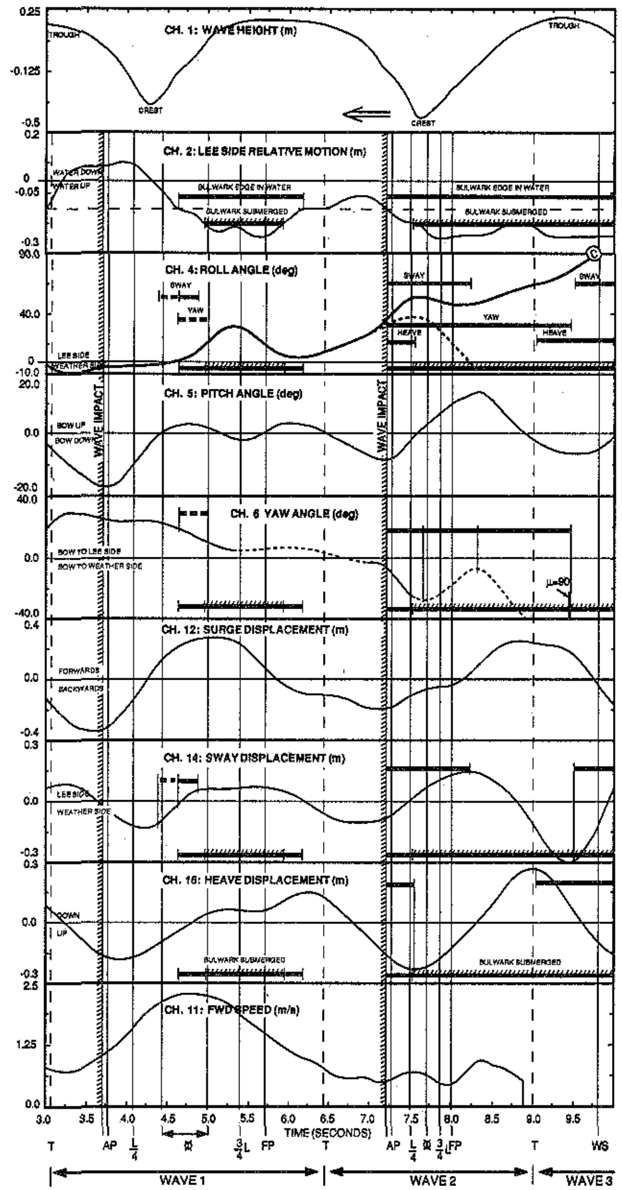

Fig. 14 Time history of model capsize in breaking quartering waves

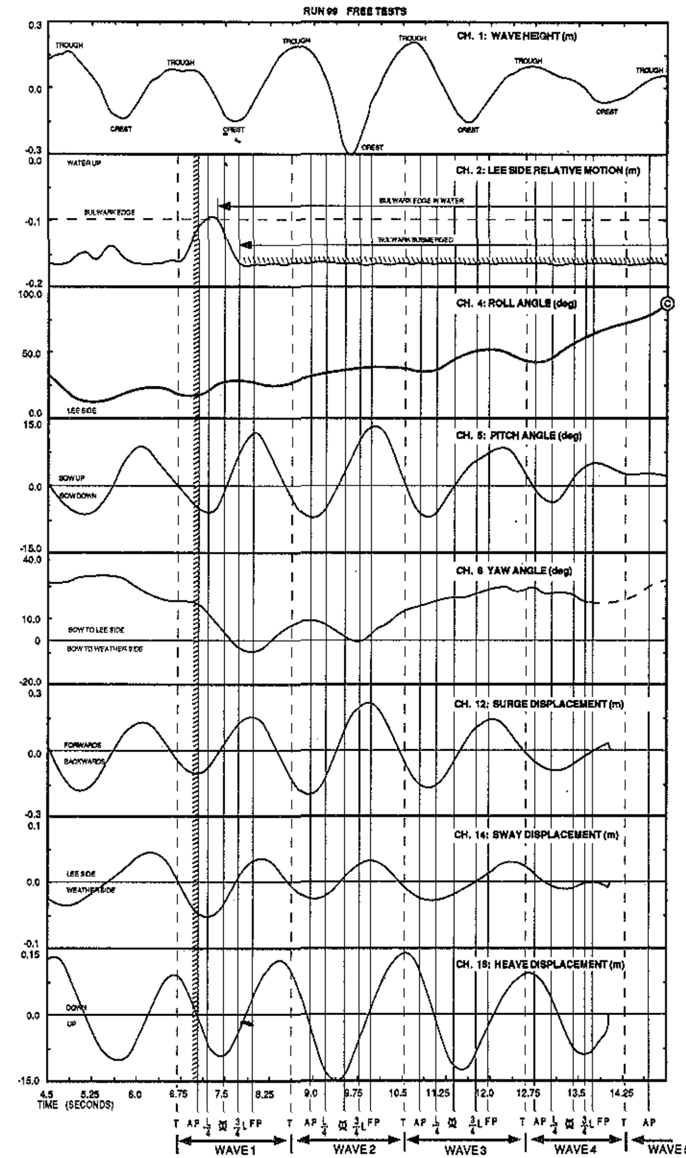

Fig. 15 Model capsize due to water on deck in irregular quartering waves

In the next run (Fig. 14) the loading and the wave conditions were the same. The only difference was the larger forward speed in this run. The first wave met the model in the position perpendicular to the crest (see: yaw graph). The wave impact pushed the model forwards without any heeling effects. The model was riding on a wave crest and started to broach a little bit. Although it was heeling to the lee side and the bulwark became submerged. there was no conducive lateral motion and therefore the additional heeling moment was not generated. The model came to the upright position.

During the action of wave 2 the situation is different and due to sway, yaw and heave the heeling mechanism is created during bulwark submergence and the model remains in deep heel after the first wave passed. Because of a permanent action of the discussed phenomenon, the third wave met the model in a large heel. The generation of the additional heeling moment by the submerged part of the deck during the third wave action was enhanced and heeled the model further. As a result, the model capsized up-side-down.

It must be emphasized that the phenomenon discussed here has a hydrodynamic nature and is generated by the reaction of the surrounding water to the movement of the submerged part of the deck. It should not be confused with the effects of water trapped on deck. In the latter situation, the water moving in the space of the deck well does not constitute a continuous extension of the surrounding water and the dynamic effects are of different nature. In order to present the difference between the discussed hydrodynamic phenomenon and the influence of water accumulating on deck a history of capsize due to water on deck is presented in Fig. 15. A systematic increase of heel angle due to accumulation of water during the action of several waves can be observed. It can be seen that the whole capsizing process is different from the previous one.

OTHER FACTORS INFLUENCING SHIP BEHAVIOUR IN EXTREME WAVES

The model tests analysis which has been carried out so far, confirmed the importance of some traditionally recognized elements and put some other factors in a new perspective. The findings have been reported in [5]. They can be outlined as follows:

1. The tests confirmed that, to some extent, the behaviour of a ship in extreme waves is correlated with the shape and values of the GZ curve. However, capsizes were recorded at the light loading condition I/A with the GZ curve satisfying the IMO criteria, when the model was running in quartering breaking waves. The described phenomenon created by bulwark/deck edge submergence is deemed to be the reason for this situation.

2. The experiments in beam breaking waves have indicated that this course does not constitute the most dangerous situation. The model operated safely when running beam to the waves in which, in the same load conditions, it always capsized when moving at quartering courses. Running in following waves may cause the vessel to ride on a wave crest, which facilitates the reduction of the restoring capability. With sufficient GZ curves, running in the following waves does not induce dangerous roll unless some additional heeling moment occurs, or the ship loses course and starts to broach. Then the typical situation of motions in quartering waves is created. The tests proved that the most dangerous situations arise when the ship is moving in quartering seas. It seems that the most hazardous course lies in the range of heading angles of 15 to 45 deg. Some phenomena which are characteristic for operation in quartering seas, do not occur in beam or following waves. They also cannot be obtained by a superposition of the hydrodynamic effects which appear in these two separate cases. This means that the stability level which could be achieved by the separate studies of ship behaviour in beam and in following waves may not provide sufficient safety for a ship operating in quartering seas.

3. Significant differences in the model's behaviour in waves were observed at the two load conditions tested. At the light condition, the model was more responsive to wave actions. The motions were very dynamic and with large amplitudes. A greater tendency to riding on wave crests and to broaching was noticed. The heeling mechanism created by the bulwark submergence was mostly involved in model capsizing. At the full load, the motions were smaller and less dynamic. Less tendency to broaching and crest riding occurred. Water on deck was the dominating factor in the majority of the capsizes.

4. The degree of danger which impends over a ship exposed to wave action depends on two wave characteristics: wave steepness and wave height relative to ship size. Extremely steep and breaking waves generate the most dynamic course of the induced phenomena, while wave height stimulates the magnitude of energy applied on a hull. Obviously the magnitude of energy necessary to capsize a ship depends on the size of that ship. Thus, the wave height to a ship depth ratio may be one of the indicators of the potential threat created by the waves.

5. The influence of forward speed on ship propensity to capsize is different for different load conditions. In light load conditions, the greater speed enhances large ship motions, facilitates the occurrence and enlarges the negative effects of broaching and bulwark submergence and, thus, significantly increases the probability of capsizing. At full load, when the hull motions are smaller and water on deck constitutes the main cause of capsizing, the higher speed

prevents a ship from intense water shipping on deck (in particular, through the stern) and increases a chance for ship survival. At the full load condition II/A, the model capsized many times when running with a low or moderate speed, but survived despite large motions and difficulties with course-keeping when running in the same waves but with a high forward speed.

6. The model tests have demonstrated an essential influence of the initial conditions at the moment of wave impact on the subsequent history of ship motion and, thus, on a probability of ship capsizing in waves. The position of a hull in space, the direction and the velocity of the motions, as well as their composition at the moment of wave impact, decide the ship's response to a wave action.

CONCLUSIONS

The submergence of bulwark and part of the deck at the lee side during dynamic motions of a ship in steep quartering seas, creates a dangerous hydrodynamic phenomenon.

After wave impact on the stern the ship is pushed violently forward and aside, acquiring a large heel angle to the lee side and dynamically yawing toward beam position. These cause a large lateral motion of the after body toward the lee side. If as a result of large heel angle and advancement of the wave crest, the bulwark becomes submerged, the lateral motion of the submerged part of the deck generates a reaction of the surrounding water. A hydrodynamic heeling moment is generated, which either increases the heel or restrains the immersed bulwark and deck edge preventing them from coming out of the water. The stiffness of the local restraint and the magnitude of the additional heeling moment generated depend, first of all, on the lateral relative velocities and on the size of the immersed part of the deck. This restraint activates new hydrodynamic couplings between roll-sway-yaw-heave increasing further the heeling moment. In addition, the whole process is happening when the ship is on the wave crest.

This phenomenon requires further study, in particular with respect to the quantitative influence on the total roll moment and to its mathematical representation. However, the experiments demonstrate that the effects created by bulwark submergence may dramatically increase a ship's susceptibility to capsizing, and may cause a capsizing of a vessel which, according to the existing stability criteria may be considered safe. Furthermore, it is not just the roll but its composition with other modes of motions which determine the probability of ship capsizing.

The tests also confirmed that the most dangerous situations are created when the ship moves in quartering waves. Some phenomena which are characteristic for operation in quartering waves never occur in beam or following waves. They also cannot be obtained by a superposition of the hydrodynamic effects which appear in these two separate cases. This means that the stability level which could be achieved by the separate studies of ship behaviour in beam and in following waves would not provide sufficient safety for a ship operating in quartering seas. Therefore, ship movement in extreme steep/breaking quartering waves, and the hydrodynamic phenomena generated at this course, should be considered as a basis for the establishment of future stability safety requirements.

ACKNOWLEDGEMENTS

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The Prediction of Deck Wetness in Oblique Waves and Effects of Shipping Water on Stability of Ships

Chanik Shin*

ABSTRACT

A theoretical method of predicting the critical wave height for deck wetness of ships in oblique waves and effects of shipping water on the stability of ships is presented. The dynamic swell-up due to the disturbance of orbital motion of water particles in the oblique waves, named dynamic swell-up due to the oblique waves, on the ship's sides in the oblique waves was investigated theoretically. The critical wave height for deck wetness of ships considering that effect in the oblique waves was calculated by using the strip method.

The model experiments on the dynamic swell-up due to the oblique waves were carried out for a model ship with the Lewis form cross sections fixed in the oblique waves. Another model experimets on the safety of ships in the waves were carried out for that model ship and a model ship of fishing vessel.

Numerical results and experimental observation teach us that deck wetting occurs on a ship when the magnitude of incident waves exceeds the critical wave height for deck wetness. In such case if deck wetting occurs repeatedly, some amount of shipping water is accumulated on deck. In this situation, stationary oscillation around heeling condition and/or capsizing of the ship occur in waves.

1. INTRODUCTION

Shipped water caused by waves exceeding the critical wave height for deck wetness produces stationary oscillation around an equilibrium heeling angle, and increases the likelihood of capsizing.

Considerable attention has been given to the problem of the critical wave height for deck wetness.

Ganno [1] took the effects of dynamic swell-up into the computation of the critical wave height for deck wetness. However, the diffraction waves he included in his computation were approximate ones in which the diffraction waves are expressed in the same form as radiation waves by Tasai [2], in Ganno's case, the ship section's velocity and acceleration are replaced by the relative motions of the incident waves.

Shin [3,4] concluded that unless the diffraction waves are computed correctly, prediction of the critical wave height for deck wetness in beam seas is not accurate even when the wave frequency is close to the heave resonant frequency or higher. Grochowalski [5] pointed out that "the deformed wave profile on the ship's side has to be determined by means of the radiation and the diffraction potential".

However, the dynamic swell-up due to the oblique waves on the ship's side is not obtained, and the critical wave height for deck wetness in the oblique waves can not be predicted correctly.

To know really the critical wave height for deck wetness in the oblique waves, it is neccessary to predict the dynamic swell-up due to the oblique waves.

The purpose of the present study is to develop a prediction method of the critical wave height for deck wetness considering the dynamic swell-up due to the oblique waves and to indicate the stability of ships when the deck wetting occurs.

2. COMPUTATION OF DYNAMIC SWELL-UP DUE TO THE OBLIQUE WAVES

We assumed that hydrodynamic flow field around a hull is two dimensional ignoring three dimensional effects for using the strip method. In other words, the orbital motion of water particle around the hull in oblique waves consists of transverse component only along cross section of the ship.

The 2-D flow field around a section of the ship fixed in the incident waves

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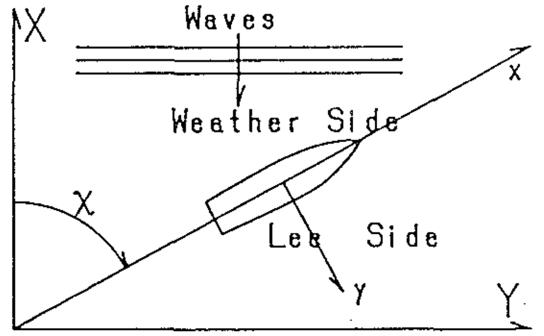

Fig.1 Co-ordinate systems

$\zeta = \zeta_a e^{-Kz} e^{i(Kx\cos\chi - Ky\sin\chi + \omega t)}$ coming at angle of encounter χ is expressed by the velocity potential (see Fig.1):

$$\phi_D(y,z,t) = \mathrm{Re}\{\ i\frac{g}{\omega}\zeta_a\int_C \sigma_D(Q) \cdot G(P,Q)ds\ e^{i\omega t}\ \} \quad (1)$$

where ζ_a is the amplitude of the incident waves, K the wave number, ω the incident wave frequency (in the case of V≠0, the encounter frequency) ; g acceleration of gravity and $G(P,Q)e^{i\omega t}$ is the velocity potential at P(y,z) of a pulsating source located at Q(y',z'), which satisfies the linear free surface condition:

$$[\ K\phi_D + \frac{\partial\phi_D}{\partial z}\ ]_{z=0} = 0$$

G(P,Q) is given by,

$$G(P,Q)e^{i\omega t} = -2\lim_{\mu\to 0}\int_0^\infty \frac{e^{-kz'}\cos k(y-y')}{k-K+i\mu}dk\ e^{i\omega t}$$

$\sigma_D(Q)$ is the density of source distribution along the section's contour C. $\sigma_D(Q)$ in equation(1) has to be determined so that the velocity potential satisfies the boundary conditions on the diffraction problem.

Velocity potential of the apparent incident waves along cross section of the ship is expressed by

$$\phi_w = i\frac{g}{\omega}\zeta_a e^{-Kz} \cdot e^{i(Kx\cos\chi - Ky\sin\chi + \omega t)}$$

For the diffraction problem, the boundary condition on the hull is expressed by

$$\frac{\partial}{\partial n}\phi_w = -i\frac{g}{\omega}\zeta_a \cdot \frac{\partial}{\partial n}(e^{-Kz + i(Kx\cos\chi - Ky\sin\chi)}) \cdot e^{i\omega t}$$

where n is the outward normal to the contour.

Actually equation (1) is rewritten in the form of the stream function instead of the velocity potential for computational *convenience; the section contour is* approximated by a polygon with 30 sides on each of which $\sigma_D(Q)$ is assumed to be constant.

With $\sigma_D(Q)$ thus determined, the dynamic swell-up, that is, the wave elevation due to the diffraction on the ship's side, is expressed by

$$\zeta_D = \mathrm{Re}\{\ 2\int_C \zeta_a\sigma_D(Q)[\pi i e^{-Kz'-iK(y-y')} - \int_0^\infty \frac{k\cos kz' - K\sin kz'}{k+K} \cdot e^{-K(y-y')}dk]ds\ e^{i\omega t}\ \} \quad (2)$$

where σ_D is the density of the source distribution giving the diffraction potential.

We have the dynamic swell-up at the weather or the lee side on the ship's side with substituting y=-B/2 or y=+B/2 into the equation (2).

3. THEORETICAL PREDICTION OF CRITICAL WAVE HEIGHT FOR DECK WETNESS

The deck wetting occurs on the ship when the amplitude of the relative wave motion between the hull and wave surface on the ship's sides exceeds the effective freeboard height whose magnitude is defined by the distance between the bulwark-top and the water surface on the ship's sides[1,4,5]. Therefore this is the condition of the deck wetting.

This condition also defines the critical wave height for deck wetness in the waves as follows:

$$\frac{H}{\lambda} = \frac{1}{\overline{\eta}_{RW,RL}}\frac{f}{\lambda}$$

where H denotes height of the regular incident waves; λ the wave length; f the effective freeboard height; $\overline{\eta}_{RW,RL}$ the nondimensional amplitude of the relative motion, $\eta_{RW,RL}/H$; $\eta_{RW,RL}$ the amplitude of the relative motion on the ship's side.

The relative motion, assuming the linear superposition principle, is a sum of all the effects - vertical displacement of the incident waves on the ship's sides, vertical displacement of the ship's sides due to ship motions, and vertical displacement of the dynamic swell-up due to ship motions and due to the oblique waves.

Theoretical prediction of the latter two effects, which are called dynamic swell-up, needs a rather lengthy computation.

Ganno [1] took approximately the effects of dynamic swell-up due to the oblique waves into the computation of the relative motion.

Shin [3,4] showed that the prediction of the relative motion in beam seas becomes accurate enough to be available for the prediction of deck wetness with inclusion of the diffraction waves computed exactly as well as the radiation waves. Grochowalski [5] pointed out the same idea, too.

The relative motion including effects of the dynamic swell-up in the oblique waves, at the weather and at the lee side, then, are

$$\left.\begin{aligned} \eta_{RW} &= \xi + \frac{B}{2}\varphi - (x - x_G)\theta - \zeta_W \\ &\quad - \zeta_H - \zeta_P - \zeta_{RW} - \zeta_{SW} - \zeta_{YW} \\ &\quad - \zeta_{DW} \\ \eta_{RL} &= \xi - \frac{B}{2}\phi' - (x - x_G)\theta - \zeta_L \\ &\quad - \zeta_H - \zeta_P - \zeta_{RL} - \zeta_{SL} - \zeta_{YL} \\ &\quad - \zeta_{DL} \end{aligned}\right\} \quad (3)$$

where B is breadth of ships; ξ displacement of heaving motion; θ displacement of pitching motion; φ displacement of rolling motion; $(x - x_G)$ the distance from the center of gravity of the ship to the square station where the relative motion is calculated; $\zeta_{H,P,R,S,Y}$ the dynamic swell-up due to heaving, pitching, rolling, swaying and yawing motions [1,2,4,6]; ζ_D the dynamic swell-up due to the oblique waves; subscript $_W$ denotes weather side on the ship's side, that of $_L$ denotes lee side.

4. RESULTS OF CALCULATION AND EXPERIMENT

The numerical calculations and experiments on the dynamic swell-up due to the oblique waves were carried out for a model ship with the Lewis form cross section, named model ship A. Another calculations on the critical wave height for deck wetness and experiments on capsizing of model ships in the oblique waves whose height is higher than the critical wave height were carried out for the model ship A and the other model ship of fishing vessel, named model ship B.

4.1 Models

Body plans of the model ships used in this experiments are shown in Figures 2, 3 and the principal particulars of those model ships are given in table 1.

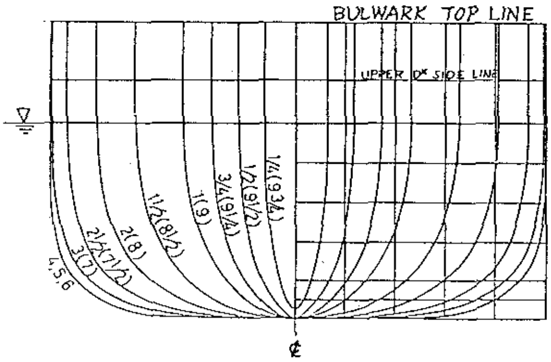

Fig.2 Body Plan of Model Ship A

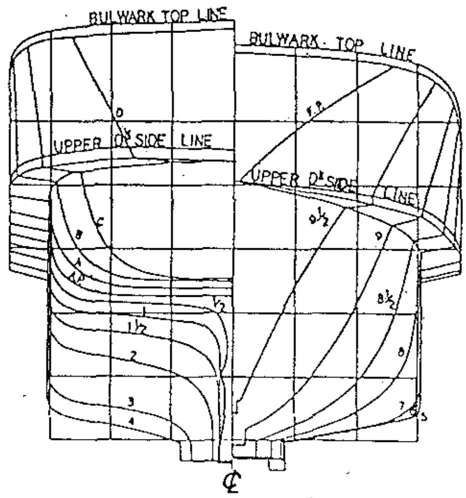

Fig.3 Body Plan of Model Ship B

The model ship A has wall-sided at all square station with Lewis form cross section and the model ship B has the wide over-hanged deck above water line.

Figures 4, 5 show the righting arm curves, GZ-curves, of each of the models. In Figures 4, solid line shows GZ-curve on bulwark height of 2 cm (the effective freeboard height f=5 cm) and dotted line shows it on 3 cm (f=6 cm).

TABLE 1
Principal particulars of The Model Ships

	Model Ship A		Model Ship B	
Length, Lpp	150.0	cm	144.0	cm
Breadth	30.0	cm	30.5	cm
Depth	15.0	cm	13.8	cm
Draft at midship	12.0	cm	10.6	cm
Displacement	36.9	Kg	33.61	Kg
GM	2.4	cm	1.4	cm
KG	9.9	cm	15.29	cm
$\otimes$G	0.0	cm	-16.0	cm
T_R	1.21	sec	1.95	sec

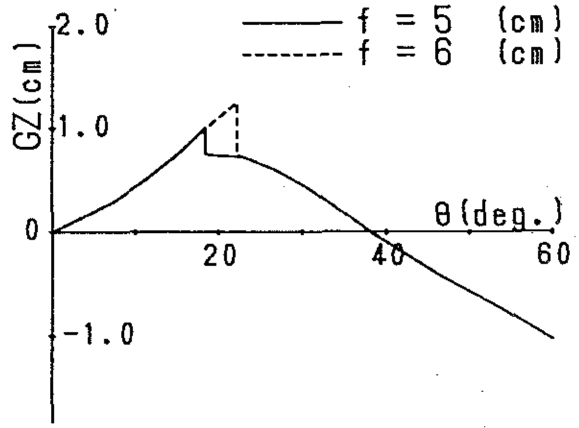

Fig.4 GZ-Curve of Model Ship A

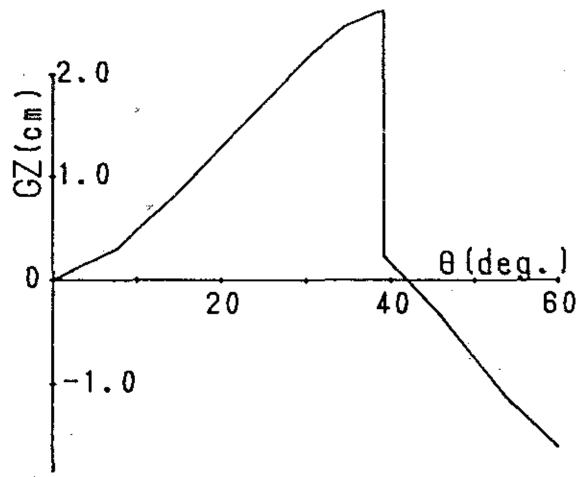

Fig.5 GZ-Curve of Model Ship B

4.2 Dynamic Swell-Up due to The Oblique Waves

The model ship A was used in this experiments. The model ship was fixed by cramp iron in the oblique waves and it had no freedom of motions.

Measurements were made on the dynamic swell-up due to the oblique waves, the angles of encounter χ=30°, 60°, 90°, 120° and 150°, at various stations (at square stations 1/2, 5 and 8) of the fixed model ship with wave height meter by using electric resistance.

The results of the calculations and the experiments are shown in Figures 6, 7 and 8. Solid lines in Figures show the theoretically predicted dynamic swell-up with the incident waves at the weather side of the model ship B fixed in the oblique waves and dotted lines show it at the lee side. $\omega\sqrt{L/g}$ wave frequency non-dimensionarized with the model's length.

Figure 6 shows the results on square station 1/2. The calculated values at the weather side is larger than the experimental values for each of the angles of encounter χ=30°,60°, 90° and 120° and good agree for χ=150°. At lee side, the calculated values for χ=120°, 150° is much smaller than the

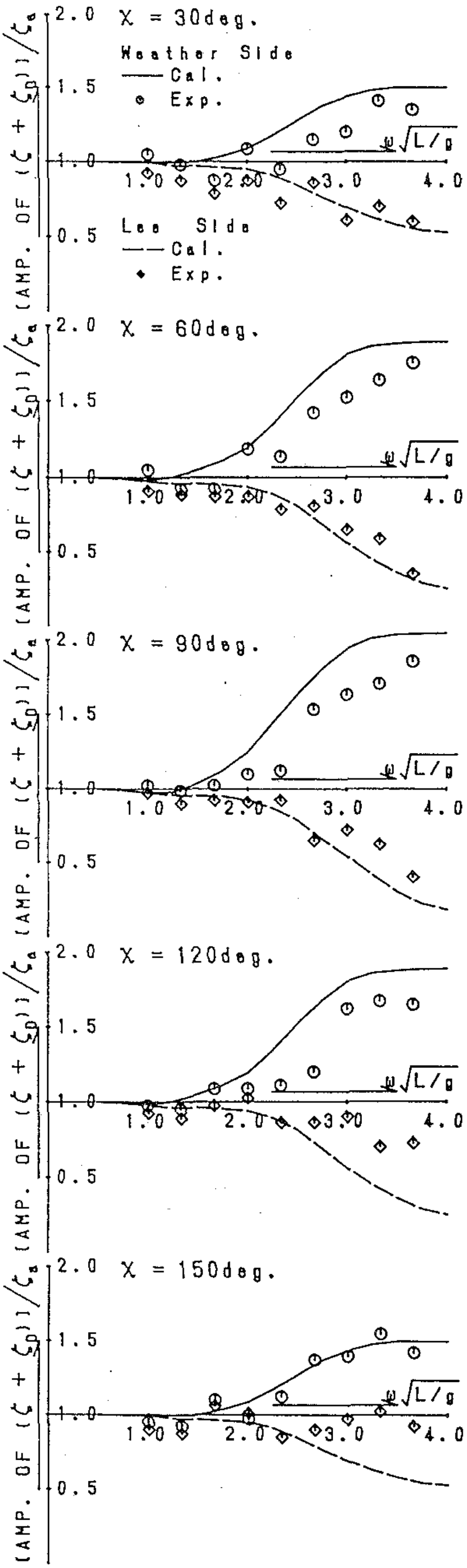

Fig.6 Dynamic Swell-up at Sq.St.1/2

experimental values and others almost agree.

As show in Figure 7, the calculated values on the square station 8 agree approximately with the experimental values at the weather and the lee side for each of the angles of encounter χ, exclusive at the weather side for χ=120°, 150°.

Figure 8 shows that close agreement between the calculated values and the experimental values was obtained at the weather and the lee side on the square station 5 for each of the angles of encounter χ.

The experimental values which were measured on the side of the fixed model ship in the oblique waves include three dimentional effects, however, these results can be explained approximately by the two dimentional theory, equation (2), exclusive some conditions for near ends of the model ship.

The results of the calculations and tne experiments teach us that the effect of the dynamic swell-up due to the oblique waves at the ship's side cannot be disregarded.

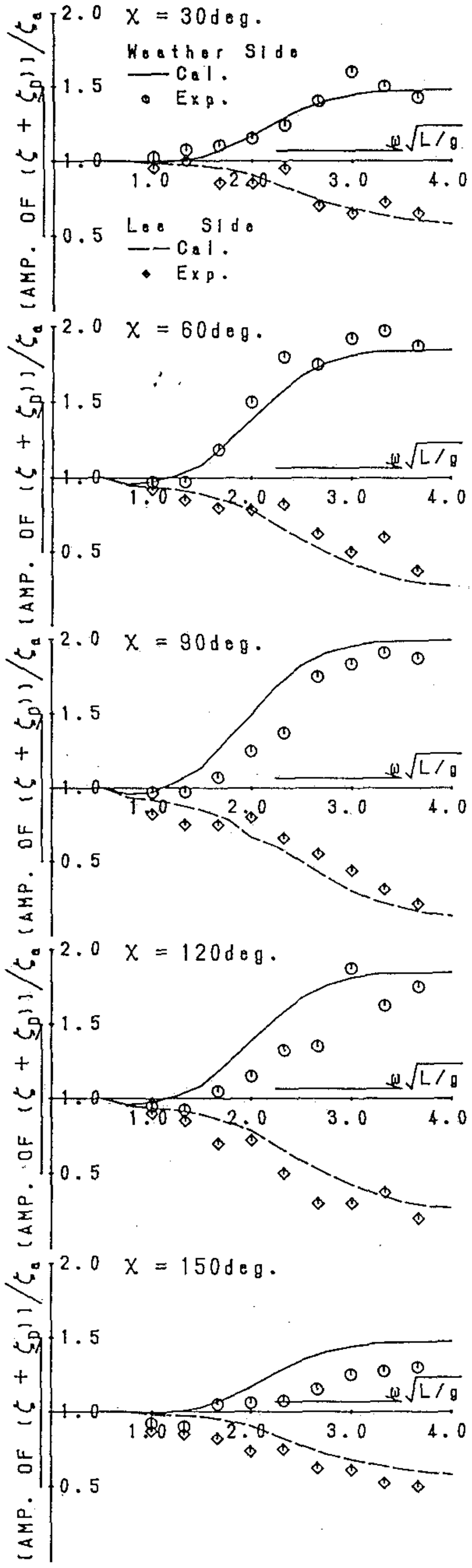

Fig.7 Dynamic Swell-up at Sq.St.8

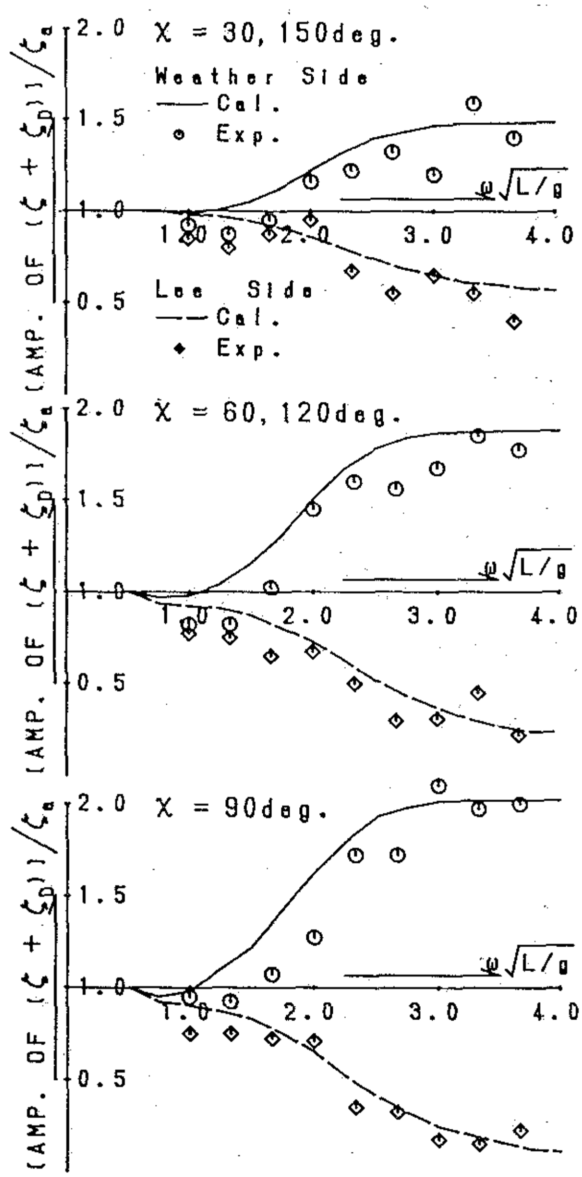

Fig.8 Dynamic Swell-up at Sq.St.5

However, it can be considered that there is less effect on that at the weather side of the square station 1/2 within $\chi<90^{o}$ and at the lee side $\chi>90^{o}$, and at the weather side of the square station 8 $\chi>90^{o}$.

4.3 Critical Wave Height for Deck Wetness and Capsizing

The critical wave height for deck wetness on the model ships is calculated by equation (3) considering the dynamic swell-up due to the oblique waves. The calculations for the model ship B with over-hanged deck were performed assuming that its side above water line is wall-sided at all square stations.

Experiments are made to find out the relation between the critical wave height for deck wetness and capsizing of ships for various frequencys of the incident waves.

The experiments were carried out for two types of ships, model ship A and model ship B, in the condition of the incident wave height considering the critical value for deck wetness. The wave frequency varied from lower than the roll resonant frequency to the heave resonant frequency or higher.

The model ship was stopped and floated in the incoming regular waves and had six freedom of oscillations.

The direction of the floating model ship relative to the incoming waves was chosen bow seas.

In the experiments, it was observed that from the bow seas condition, free behavior of the model ship in waves is gradually transferred to stationary oscillation around an equilibrium heeling angle because of shipping water on deck and/or to capsizing in beam seas.

The results of the calculations and the experiments are shown in Figures 9, 10 and 11. Figures 9 and 10 show the results on the model ship A with the bulwark height 2 cm and 3 cm respectively. Figure 11 shows it on the model ship B. Solid line in the figures shows the theoretically predicted critical wave height for deck wetness for each of the models tested. In the Figures, less wave height does not ship water on the deck as marked by the open circles; the semi-black circles indicate that shipping water gets on the deck but the model does not capsize; the black circles show the model ship capsizes after the shipping water gets on deck.

It is to be noticed that the shipping water gets on deck in the condition of the wave height exceeding the critical wave height for deck wetness for several frequency, as is two-dimentional model ship [7].

At the roll resonant frequency, the critical wave height is very low, because the roll motion is very large, the shipping water, however, does not occur in succesion. Therefore the model ship does not capsize.

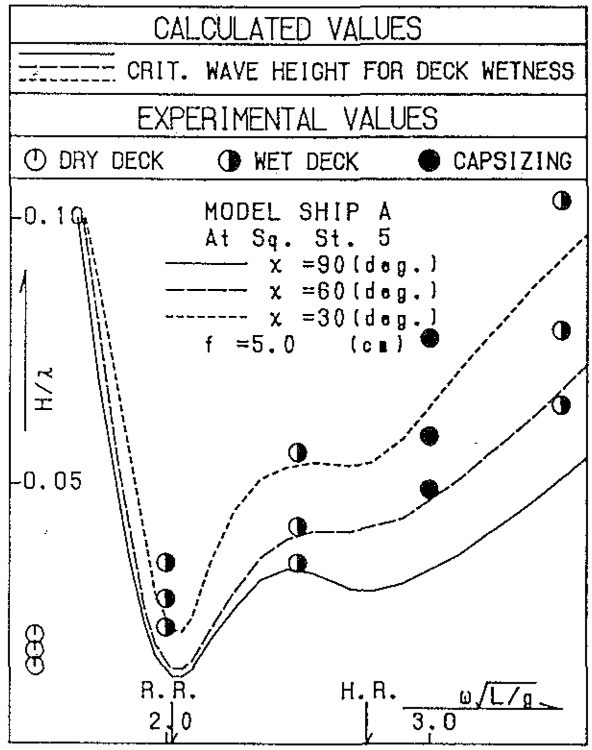

Fig.9 Relation between Crit. Wave Height and Capsizing Model Ship A (f=5 cm)

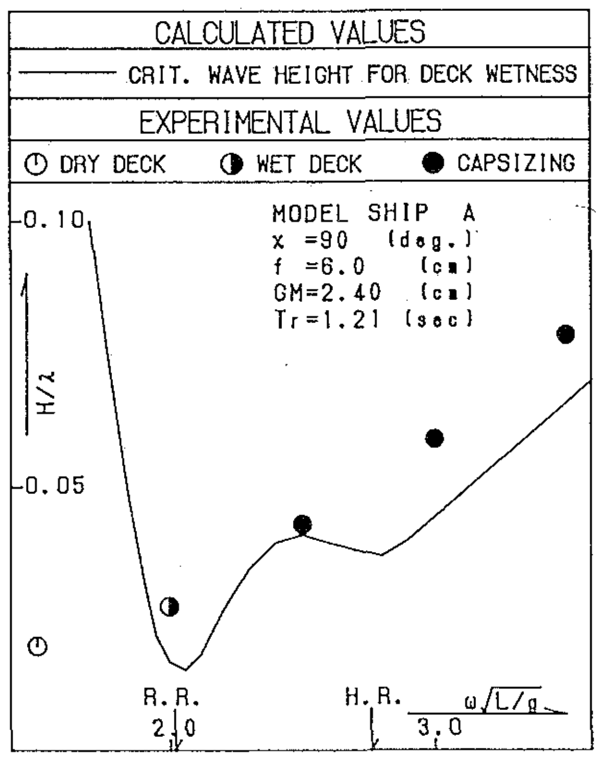

Fig.10 Relation between Crit. Wave Height and Capsizing Model Ship A (f=6 cm)

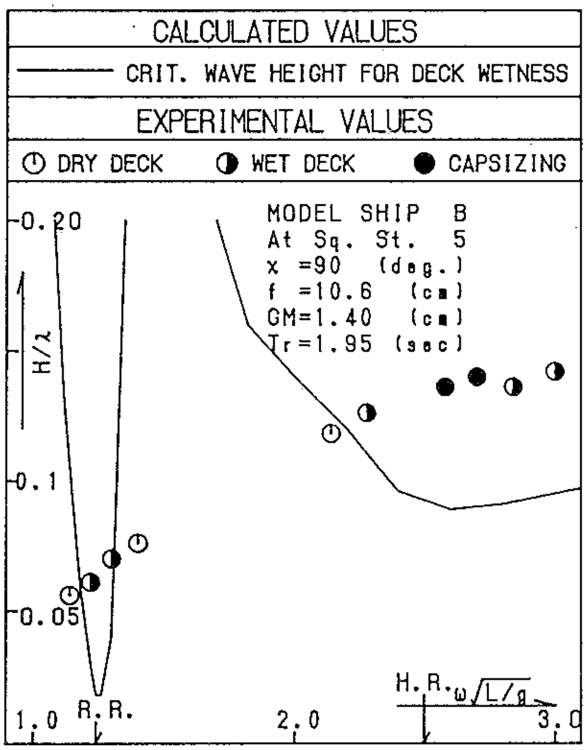

Fig.11 Relation between Crit. Wave Height and Capsizing Model Ship B (f= 10.6 cm)

The black circles, indicated in the Figures as capsizing of the model ships, show that the deck wetting occurs even for small rolling when the the frequency of the incident waves is the heave resonant frequency and higher, then the model ship capsizes under the effect of the shipping water accumulated on deck. However, the semi-black circles at the high frequency show that the model ship does not capsize even if the frequency of the incident waves is higher in the condition of its height exceeding the critical value for deck wetness.

5.CONCLUDING REMARKS

The present theoretical and experimental results made it possible to suggest the relation between the critical wave height for deck wetness in the oblique waves and capsizing of ships.

From the present results, the following conclusions are reached:

1) The dynamic swell-up due to the oblique waves is required to predict the critical wave height for deck wetness and can be approximately calculated by using the two-dimensional linear theory.

2) The method of calculation for the critical wave height for deck wetness considering the dynamic swell-up enables us to predict the occurrence of the deck wetting of ships in the oblique waves.

3) Capsizing does not occur for the incident wave height less and slightly higher than the critical wave height for deck wetness.

4) Capsizing occurs easily for the incident wave height higher than the critical value when wave frequency is nearly the heave resonant frequency. In future, the method of the prediction of the critical wave height for capsizing is necessary.

5) In case the shipping water occurs on deck, the bulwark height has direct effect upon capsizing of ships, however, the relation between them is not clear.

ACKNOWLEDGEMENT

The author wishes to thank Mr. T. Kanto who is a graduate student of Nagasaki Institute of Applied Science for his assistance in these experiments.

NOMENCLATURE

B	:ship breadth
C	:section's contour of ship
f	:effective freeboard height (the distance between the bulwark-top and the water surface)
G(P,Q)	:velocity potential at P(y,z) of a pulsating source located at Q(y',z')
g	:acceleration of gravity
H	:incident wave height
K	:wave number
ζ_a	:incident wave amplitude
ζ_D	:dynamic swell-up due to the oblique waves
ζ_H	:dynamic swell-up due to heaving motion
ζ_L	:vertical displacement of incident waves at the lee side on the hull surface
ζ_P	:dynamic swell-up due to pitching motion
ζ_R	:dynamic swell-up due to rolling motion
ζ_S	:dynamic swell-up due to swaying motion
ζ_W	:vertical displacement of incident waves at the weather side on the hull surface
ζ_Y	:dynamic swell-up due to yawing motion
η_{RW}	:amplitude of relative motion at the weather side
η_{RL}	:amplitude of relative motion at the lee side
θ	:displacement of pitching motion
λ	:incident wave length
ξ	:displacement of heaving motion
σ_D	:density of source distribution along the section's contour
ϕ_D	:diffraction potential

ϕ_w :velocity potential of the incident waves

χ :angle of encounter

ω :frequency of the incident waves

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COMPUTERS ON BOARD A GENERAL CHALLENGE

Bruno Arndt, Cert. Eng. 1)

A B S T R A C T

Digital computers on board now offer the possibility of performing comprehensive calculations, but the accuracy - or better confidence - of the results depends on that of the input data, and there are great differences in quality. But even when attempts, to reduce the tolerances of some essential parameters are successful, it may be a problem to adjust calculated results with measured ones. Other items suffer from the fact, that they are based on methods or regulations which have their roots in times prior to the computer era, and do not match the conditions or necessities of E.D.P., or are not standardized on an international base.

So, the presence of a digital computers asks for precision in definition and theory, where guesswork might have been sufficient in the past. The supplier of onboard computers has to find solutions momentarily, but final and comprehensive ones could be constributed by research institutes at Technical Universities, Classification Societies etc.

THE ELECTRONIC CREW MEMBER - RELIABLE AND ACCURATE

Since some ten years a new crew member is boarding our merchant vessels, supported by safety regulations and classification societies' requirements: The freely programmable digital computer, mostly in form of a personal computer. This new crew member is ready for 24 hours round-the-clock service without overtime payment, with-out demand for vacation or even recreation time, more resistant against seasickness than his human collegues, and fit with a memory, which stores only what is told to him to be important. His outstanding feature is the capability to perform calculations extremely quickly and always with the same and accurate result, if input data are accurate and the calculation method unchanged. And whereas the computer basically is compulsory for longitudinal strength control calculations, it is also used for stability calculations, thanks to its versatility.

Sometimes the opinion is met, that normal micro computers, especially so-called personal computers, are not fit for usage on board of ships. But based upon the installation of some 130 board computers of PC-type with altogether approx. 400 operational years. among them some 40 systems with harddisk (Winchester type) and 100 operating years, with respect to failures it can be stated, that

- the frequency of breakdowns is of the same order of magnitude than that of the same systems ashore,
- the systems are working exactly - or not at all. Intermediate stages are in general caused by the software,
- the weak point may be the monitor - if ever -, but not the harddisk,
- with respect to vibrations and climatic stresses, the operators are less resistant than the hardware,
- the type-approval testing conditions have the character of quick-motion (or statistical) procedures, and therefore e.g. local vibrations of inward boards should not be overestimated (this is an appeal to the classification societies!): If the vibrations imposed upon the specimen during type-approval tests would occur on board during normal service, the vessel would not get the sailing permit from the authorities.

The reliability of the software is somewhat different, because even simple loading control programs for general cargo vessels consist of several modules with cross references, so that odd combinations of

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figures may result in breakdowns, but in course of time the probability of such failures should be reduced to nearly nil.

Apart from the strange behaviour that sometimes 2+4+3 are summed up to 8 or 10, due to the fact, that less decimal places are printed than were used during the calculations, the same hardware is producing always identical results, if the input data remain unchanged, i.e. they can be reproduced as often as wanted.

It may also happen, that results of calculations, performed with software of different origin, differ from each other by one unit, i.e. 0,1 t or 1 cm. That are not the tolerances which are causing headache, because they have no essential effect upon the safety against capsizing . If necessary, certain calculations may be performed with double word length, and generally speaking, the accuracy even of personal computers is sufficient for the basic computations of stability, longitudinal strength and many others.

And even if there should be a definite demand for more powerful computers, it would be a matter of financial effort only, to install larger and quicker systems on board with 12 MHz clock frequency, 4 MByte RAM, 80 MByte harddisk capacity, or even higher.

So it can be stated, that repeated calculations result in identical and accurate output, if the used input data are corect and not modified.

This trivial statement is not very exciting, but the inverse, not less trivial form may broach serious considerations:

- If the input data are wrong, the results cannot be correct.

THE CONFIDENCE OF DATA

The data, used for calculation of static and "dynamic" stability, consist of two groups:

- Constant or fixed data, most of them depending upon the geometry and construction of the vessel in question,
- Variable data, mainly all deadweight items and their positions.

For calculation of GM, data of both types are used. The distance of the metacentre M above the reference point K is available from the hydrostatic curves (or tables), and the accuracy can be assumed to be sufficient. The other end of the initial stability, the centre of mass G, is normally the result of a momentum calculation, where the light-ship weight and its centre of mass can be looked upon as considerably accurate figures, whereas the items of the deadweight may have more than only small tolerances. Thus it may happen that the calculated draughts are differing remarkably from the read-out ones. That means that also the calculated displacement is faulty, and if this value is used for further calculations, not only the righting levers GZ are inaccurate, but even the position of M, in spite of the fact, that KM has been derived from the lines plan.

In literature about ships stability no advice could be found how to solve this problem. So the "field engineer" had to find a solution by himself. The first idea, to put in only the read-out draughts, and let the program then take the correct hydrostatic data from the stored tables, is no good one, because KG is not the result of a hydrostatic calculation and therefore the second point of the distance GM is yet missing. So the so-called "adjusting bale" method was introduced: After input of the read-out draughts the program calculates the weight of an adjusting bale which would bring the computed draughts into coincidence with the actual ones. The centre of gravity of this adjusting bale can be derived in longitudinal direction from the trim situation of the vessel, whereas in vertical direction it is assumed to be in the centre of gravity of all dry cargo, as long as no better information is available. This assumption would result in a completely correct position of G, if the deviation of weight would be in the same percentage for all items of the loading, e.g. for all containers.

If then the adjusting bale is added to the deadweight, which has been introduced already by "dead reckoning", the basic situation for stability calculations is sufficient or possibly even correct (Fig. 1)

The position of the centre of gravity of parts or all of the deadweight is the most serious source of considerations when discussing the accuracy of stability calculations. For tanks, if their data have been calcu-lated properly, the CoG values have a good degree of trustworthiness, and that of bulk cargo - especially when loaded with

Designation	Weight	CGL	CGV	CGT	Ib*gamma
Crew	6.00	125.00	27.50	0.00	0.0
Provision	20.00	119.50	23.50	0.00	0.0
Stores	150.00	60.45	11.50	0.00	0.0
Fuel oil	535.00	71.60	1.31	1.19	720.0
Diesel oil	64.70	30.75	2.03	-2.83	67.3
Lubric. oil	53.70	23.61	1.57	-2.32	64.0
Fresh water	217.40	127.83	4.94	-0.44	97.0
Ballast-water	85.30	23.08	5.92	-2.85	227.6
Misc. tanks	82.30	52.06	1.90	2.43	126.0
Empty ship	5834.00	68.63	9.92	0.04	0.0
Adjust.Bale	233.87	57.03	10.71	0.00	0.0
Cargo	6167.40	74.24	10.71	-0.06	0.0
Displacement	13449.67	71.34	9.77	-0.00	1301.9

Fig.1: Loading Condition Sheet with Adjusting Bale

levelled surface - are of similar degree of accuracy. For containers, a standard height of CoG as fraction of the height of containers is used, namely 40, 45, or 50 %. A value of 45 % of container height may give good approximation to reality, because most containers will be filled to an extent less than 100 % and due to physics the free space will be below the top of the container.

With rolling cargo like cars, trucks, railway carriages, caterpillars, graders, scrapers etc., it is somewhat more difficult to get reliable CoG values, but as far as ferry boats are concerned, the payload is merely a fraction of approximately 15 - 30 % of the displacement, and therefore a failure in payload CoG is affecting the GM in a reciprocal ratio only. With RoRo-ships the situation is similar, because in most relations a mixed cargo of containers and rolling cargo is carried. And moreover, both types of vessels have in common, that at least the vertical position of the different cargo decks is very well known.

The worst situation is met with general cargo vessels, where the vertical position of the numerous bales, boxes, palettes etc. cannot be indicated with sufficient accuracy. Experiments, performed some decades ago by members of the chair of Prof. Wendel, Hanover, resulted in a failure of 0.3 m and more, compared with the results of in-service inclining tests.

MEASUREMENTS AS ALTERNATIVE

The key-word "inclining test" may give reason to hope that the problem of determinating KG with sufficient accuracy could be solved by this well-known indirect measurement method. From the past it is known that sufficient accuracy could be reached only, if the test was performed carefully by experienced people. Nowadays when more precise gauges for inclination, draughts and tank contents are available, the presuppositions for successful inclining tests are much better as was found by an investigation (1), the results of which were presented by S. Kastner.

A program for the evaluation of an inclining test itself is simple, and existing here and there already since years. But after having determined an accurate GM', i.e. GM including the free surface effect of liquids in tanks, the consequences must be drawn with respect to the loading calculator, because

- additional stability values must be determined,
- the actual loading condition must be recorded ,
- the facility to derive future loading conditions must be established,

i.e. KG of the system lightship plus deadweight must be adjusted according to the result of the inclining test. For that purpose it must be decided, whether KM, lightship weight and CoG of lightship can be assumed as to be faultless, how the reduction of GM through the free surface effect can be reconstructed and whether CoG of only the cargo shall be adjusted. To meet these decisions, the user must at least be supported by an official guide line, derived from further investigations or at least considerations and issued by national or international authorities.

THE OTHER SIDE OF THE BALANCE

GM, righting levers, and the areas below the righting lever curve as equivalent of dynamic stability are only one side of the balance. On the other side there are heeling levers due to moving liquids, beam wind,asym-metrical loading, shifting grain, and others, which may occur with special ships. Naval architects with special interest in ship's stability know very well, that e.g. the free surface effect, represented by the transverse moment of inertia of the cross-section of a tank, gives accurate values only as long as this cross-section does not change its shape by touching bottom or top of the tank. The calculation of heeling levers due to beam wind suffers even from several doubtful parameters: The wind resistance coefficient lies in a range from 0.9 to 1.5, depending of the shape of the element of the lateral area, and - especially with container vessels - this area is sometimes a collection of separated towers, with canyons in between. Further, a heeling can only be generated by beam wind if below the waterline a beam resistance occurs, acting upon the underwater lateral area. The centre of this area is assumed to lie at half draught, but this too is only an approximate value with the character of a standard, as most of the coefficients used in calculation of heeling moments.

With asymmetric loading and shifting cargo the situation may be somewhat better, but there are at least two effects, which are not at all taken into account by definite calculation: The variations of righting levers in waves, and rolling amplitudes.

STRATEGY OF STABILITY CONTROL

With these latter remarks it is by no means intended to pull down the whole building complex of stability control. The author, when acting decades ago on the opposite side, namely participating in the elaboration of stability regulations for the surface vessels of the West German navy (2) (3), came to the conclusion, that as many elements as possible of the stability balance should be determined accurately, so that the margin for uncertain interior or environmental effects could be reduced to practical limits. For the other ones, algorithms and coefficients should be standardised internationally. This method has several advantages:

- Error areas will be restricted
- The order of magnitude of heeling effects can be made visible
- Characteristics of different types of ships will become obvious
- Discussions with "field surveyors" of classification societies, which not always are specialists in ship's stability, can be reduced, when performing test calculations with loading computers on board

Special emphasize should be laid upon international standardization. Nowadays, every classification society has regulations for stability control of its own, which are identical to a certain extent, but also differing in some essential details so as if the danger of capsizing would be graduated, depending upon the flag the respective vessel flies, or the society, under which surveillance it was built. In some cases even the regulations of two societies must be satisfied, e.g. of the Polish and the Chinese register, or of the Indian and Lloyd's register, etc., but obviously without the effect of doubling the safety margin. Something similar happens to the hardware, which has to undergo type-approval procedures of several classification societies, when supplied to vessels of respective surveillance, whereas in the area of ship construction mutual acknowledgement of the rules is common use already. (But at least one of the important classification societies does without hardware type approval, obviously trusting, that the instruments will not be installed on the flying bridge or mounted on the casing of the main diesel engine respectively).

A further approach to uniformity of calculations and reliability of software could be achieved, if the basic algorithms would be programmed and distributed to all makers of loading calculators by the classification societies or even by IMO. The author makes this proposal in full awareness of the fact, that the realisation of it would mean a restriction of his creativity as well as a loss of corporate identity of his company, but around the basic procedures of stability, and longitudinal strength calculation there is yet a lot of input, output, and statistical tasks to be elaborated, to say nothing of other problems which could be assigned to the electronic crew member, like optimizing of trim with respect to fuel saving, lashing of containers, control of separating rules of containers with dangerous goods etc. And above all, one should not forget that the introduction and spreading of board computers has the purpose of increasing the safety against capsizing (and to avoid over-stressing of the ship's structure) and not, to give the makers the opportunity to earn money.

If the idea mentioned above would be realized, they would have to expect a loss of profit, but this could be compensated for to some extent, if the installation of two identical and self-sufficient systems would become compulsory at least for ships in worldwide traffic. Then the same degree of redundancy would be achieved as with two radar systems, but much cheaper.

PROGRESS, ADVANTAGES, IMPROVEMENTS

Basically, a computer on board offers the opportunity for steady control of stability by momentum calculation, whereas in the past stability calculations on board were executed only - if ever - if a critical loading condition seemed to be expected. At the same time the probability of mistakes has been reduced, because the calculation method is stored errorfree, and the input data are controlled to some extent by means of plausibility checks. The trustworthiness of results can be improved by combination of momentum calculation with adjusting bale module and in-service inclining tests. Telecommunication with stowing centres ashore shall be mentioned only as a means of preplanning of stowage free from the hustle and bustle on board. Computers with mass storage device offer the opportunity to save standard loading conditions for later modification and reuse.

Accuracy and trustworthiness can be improved further by on-line input of measured data like tank contents, draughts, temperatures of fluid cargo, and others, provided, the respective measuring gauges and instruments are installed. Even simple PCs are prepared for such tasks, and several installations have proved already the feasibility.

The introduction of the third (transverse) axis in momentum calculation opened not only the way for calculation of list, but also for control of torsional or even combined longitudinal and torsional stress calculation.

The longitudinal strength calculation can be carried on to calculation of bending of the hull, and then the question arises, whether it would be meaningful, to adjust the hydrostatic data. The answer should not be left to the field engineers, because it seems to be somewhat difficult to define KM for a vessel in bent condition, not to speak from GZ values.

A LOOK BEYOND THE EDGE OF THE PLATE

A freely programmable computer on board is by far not occupied fully by standard loading calculations, but when preparing programs for other tasks, the analyst very often comes to the point, that investigations and algorithms have not yet been developed to a state, which allows the transformation into a program for practical use on board. E. g. a program for prediction of rolling amplitudes due to ship's speed and direction of encounter of waves has been developed already (PCs are indeed suited to solve Mathieu differential equations in due time), but the results are more or less of qualitative character, because the roll damping coefficients are not very precise. A weather routeing program also could be improved, if the findings of theoretical investigations could be transferred reliably into practice.

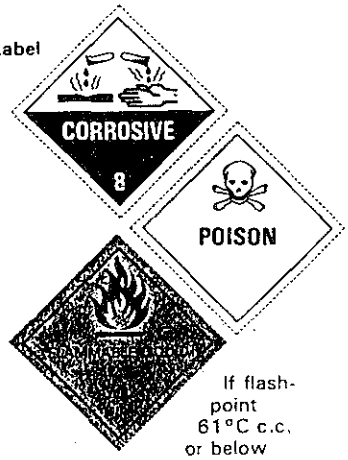
Quite other problems arise, when the stowage and separation of dangerous goods according to the requirements of the International Maritime Dangerous Goods (IMDG) Code shall be controlled. The goods defined to be dangerous are marked with a class and a UN-Number, but already here begins the trouble, because for several goods there are different separation requirements, depending upon the content of water, or the state of matter. So, in these cases different goods have the same UN-Number, and the problem is, how to teach the computer the method of distinction. (Fig. 2)

If a dangerous good is to be stowed "away of" or "apart from" foodstuffs, or strong acids, or dusts of heavy metals, the analyst has to find out these groups of goods from among over 2000 UN-Numbers. The worst case are foodstuffs, or goods which emit strong smells, because they have no UN-Number at all, i.e. they have no identification mark, which fits to electronic data processing.

Nowadays in most cases computers on board of ships are used only for control of stability and longitudinal strength, but they are prepared to take over additional and more difficult tasks, provided, that they can be translated into the language the computer is able to understand. In order to overcome this problem, the cooperation between scientists, institutions and users should be intensified, with the connecting link of the systems

Packaging group: II

Label



Observations
Toxic if swallowed, by inhalation or by skin contact.
Corrosive to skin eyes and mucous membranes.

Packing
See table 1 in the introduction to this class.
For tanks see section 13 of the General Introduction.

Stowage
Category B.
For tanks see also section 13 of the General Introduction.
Keep as cool as reasonably practicable.
"Away from" acids
If flashpoint 61°C c.c. or below, segregation as for class 3, *but "away from" class 4.1*

Packing, Stowage & Segregation
See also General Introduction and introduction to this class

Fig.2: Stowage Advice for UN-Number 2683 with Condition "if"

analyst in between. The advantages with respect of safety and economics then even may justify the incorporation of a special EDP-operator into the ship's staff.

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AN EXPERT LOADING SYSTEM FOR CHEMICAL AND PRODUCT CARRIERS

L. Bardis[1] T. A. Loukakis[2] G. A. Vouros[3]

ABSTRACT

Chemical and product carriers are highly specialized ships, carrying a multitude of cargo types, sometimes very hazardous to the sea environment. The operation of this ship type is a complex task, requiring sometimes the cooperation of experts from several engineering disciplines. A great amount of regulations exists, pertaining to cargo/tank compatibility, safety procedures during loading/unloading and tank cleaning. For the above reasons the preparation of a loading plan for a chemical carrier is a complex procedure requiring high skills and a lot of experience.

The primary objective of this work is to investigate methods to make expert knowledge available to ship masters. This is achieved by applying artificial intelligence techniques to deal with the problem of loading of chemical carriers. A scheme for encoding existing rules and regulations in a fact base is provided and a preliminary version of a rule-based system has been developed to assist the expert during the preparation of a loading plan. A prototype able to communicate with algorithmic routines performing standard hydrostatic and strength calculations has been created and its operation is explained.

INTRODUCTION

The Loadmaster is a very common tool used on board ships to perform standard hydrostatic and strength calculations. Advanced equipment of this type is interfaced to measuring instruments installed in tanks of ships carrying liquid cargo, such as liquid level and temperature sensors, thus enabling real-time monitoring of cargo loading/unloading processes. Container ships are often equipped with Loadmasters providing extensive bookeeping facilities for the containers.

Ordinary Loadmasters require data pertaining to cargo distribution from the user. They may be classified as "dummy" instruments, in the sense that the user takes the decisions on the basis of the results provided by the instrument. On the other side, modern software technology has made available the tools, which help to simulate human actions in complex situations such as loading/unloading of special ships. These tools are known as expert shells and the resulting computer programs are called expert systems. In an expert system, domain knowledge is encoded in a suitable form and is applied to solve the problem following logical steps emulated by some reasoning mechanism. Ref. [1] gives an overview of expert systems in the marine industry.

Loading of specialized ships such as chemical and product carriers is a complex problem requiring skilled operators, [3], [4]. A great number of rules and regulations related to stability and strength on one side and handling of dangerous chemicals on the other side have to be taken into account. Compatibility between cargo and ship equipment (tank coating, cargo piping), chemical compatibility between cargo grades stored in adjacent tanks, disposal of cargo residues are examples of critical issues, which have to be taken into account when cargo allocation and handling is considered. Economy in ship operations affects also the load plan. It is generally recognized, that loading and unloading of the ship in a minimum time as well as trim optimization resulting in reduced fuel consumption contribute to the overall economy goal.

In the present work, we propose an Expert System for loading of chemical and product carriers. This system, the Expert Loading System (ELS) is a component of a larger, integrated system, named KBSSHIP, which is developed in the ESPRIT P

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2163 project. This project is funded in part by the EEC within the framework of the ESPRIT II programme. Private firms and research organizations from four european countries are participating in the above project, which started in January 1989 and is scheduled to finish in June 1992. Feasibility studies were carried out in a predecessor project, funded under ESPRIT I.

The main objective of KBSSHIP is to integrate a number of expert systems related to operation of ships. KBSSHIP is concepted as a decision support tool with coordination and reasoning capabilities. The individual subsystems are :
The System Manager Expert System (**SMES**) providing communication and high level control over the other subsystems.
The Expert Voyage Pilot (**EVP**), whose task is to set up a voyage plan taking into account charter requirements and the propulsion and seakeeping characteristics of the particular ship.
The Expert Loading System (**ELS**), which is used for the preparation of an optimum load plan and the specification of a sequence of operations required to realize this plan.
The Expert Maintenance System (**EMS**), which produces maintenance plans for the ship machinery and equipment by distributing available resources.
The Expert Diagnostic System (**EDS**) for the identification of faulty components and issuing of warnings about imminent ship equipment malfunction by analysing sensory information.
The Statutory requirements Classification Expert System (**SCES**), which provides advice about rules and regulations applicable for the particular ship using an expert query and information retrieval scheme.
Further, KBSSHIP includes an Information Storage and Retrieval System (**ISRS**) maintaining a common data base for data shared by all subsystems.
Each subsystem performs reasoning by using its own knowledge base and data coming either from other subsystems or the ISRS. Handling of requests for supply of data issued by individual subsystems is performed by SMES.

The Naval Architecture Laboratory of the NTUA is the task leader in the work package dealing with the development of the ELS and is involved in the development of the EVP. In this paper, the design of the ELS and the environment used to model domain knowledge are discussed. Section 2 contains the description of the ELS in terms of its components. Section 3 gives an overview of knowledge modeling techniques, in particular the PHOS conceptual formalism. Finally, in Section 4 some conclusions are drawn.

THE EXPERT LOADING SYSTEM

Fig. 1 gives a overview of the ELS, [2]. Knowledge or data bases are represented by ellipses. Rectangles are used to represent modules effecting transformations of the above data. The ELS is designed to function either as a stand-alone system or as a subsystem integrated within the KBSSHIP environment as explained in the introduction. It consists of two layers (Fig. 1), an upper or expert layer and a lower or algorithmic layer.

The lower layer contains two modules, the Algorithmic Module (AM) and the Visualization Unit (VU). All routines performing low level tasks, such as trim, stability and strength calculations, interaction with the operating system of the host computer and the user, and plotting routines are located in this layer. The upper layer contains the Loading Planner (LP) and the Cargo Handling Unit (CHU). This is the expert layer of the ELS, whose main tasks are the optimization of the distribution of the cargo among the cargo tanks while observing all constraints imposed either by international and national authorities or by operational restrictions of the ship equipment in connection with cargo to be transported. The KBSSHIP & User Interface (KUI) takes over all communication between the ELS and other KBSSHIP subsystems through SMES.

The upper and lower layer modules and the KUI communicate with several data and knowledge Bases. These are the Rules and regulations Knowledge Base (RKB), the ship and tank Geometry Data Base (GDB), the Charter requirements Knowledge Base (CKB), the cargo handling Equipment Knowledge Base (EKB), the Products Knowledge Base (PKB), the Stability and longitudinal strength Data Base (SDB), the cargo Distribution Data Base (DDB), the Loading/unloading operations Knowledge Base (LKB) and the INstrument Interface (INI). The components of the ELS are described below in detail.

<u>The Loading Planner (LP)</u>
The Loading Planner plays a central role in the ELS. The main task of the LP is to produce a load plan, i.e. a table containing the quantity of liquid cargo to be placed in each tank. In order to prepare the load plan the LP takes into account the

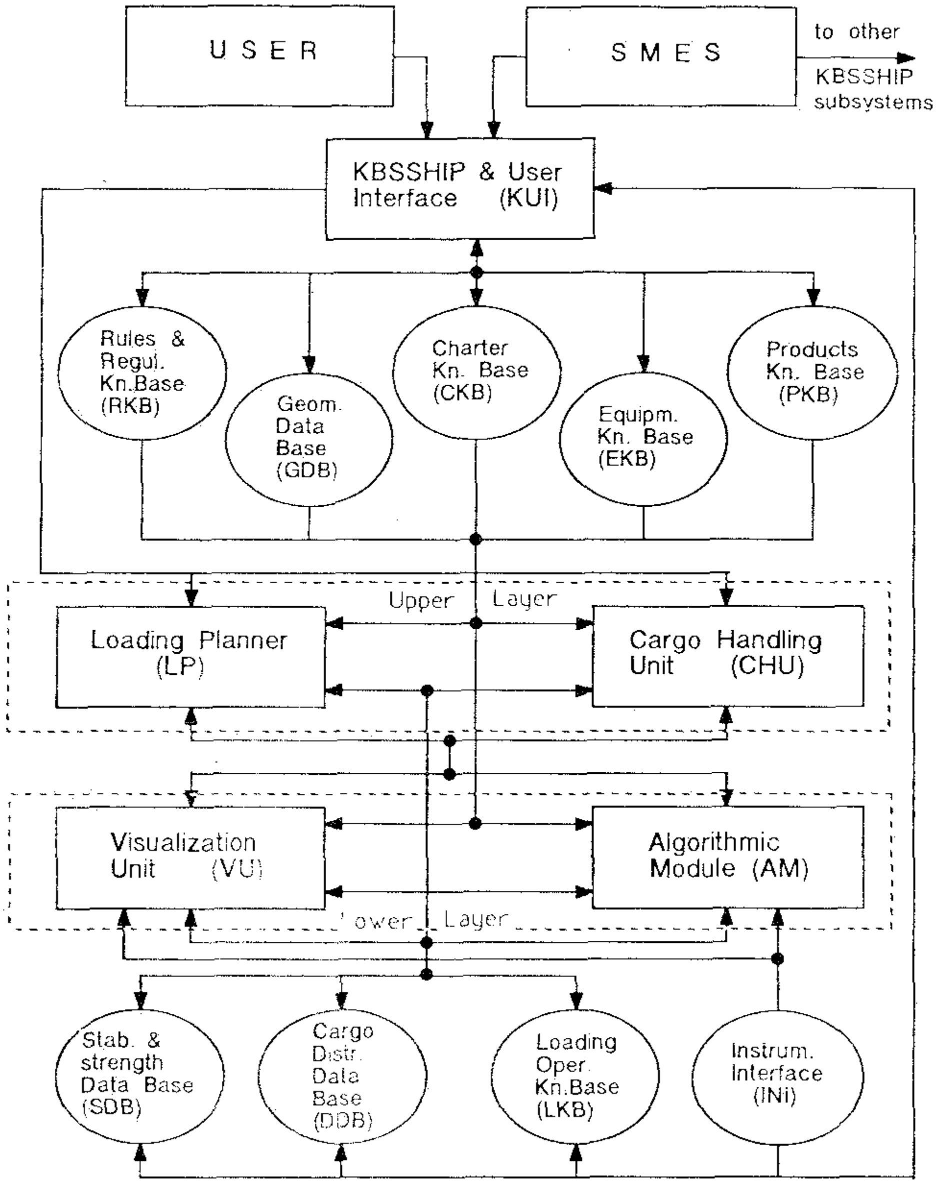

Figure 1. The Expert Loading System

following restrictions :

- Rules and regulations by classification societies, national and international organizations and international conventions. The pertinent data are drawn from the RKB. Stability and strength limits are included in this category ;
- Charter requirements contained in the CKB. The description and quantities of cargo items is the most important information drawn by the LP from the CKB ;
- Properties of the cargo to be transported, which prohibit containment of special liquid grades in some ship tanks, for example due to chemical interaction between the cargo and the tank coating ;
- Compatibility between cargo and cargo handling equipment such as cargo pipes, pumps, valves, packings, etc.

Further, the LP :

- places requests to the AM to perform hydrostatic and strength calculations ;
- draws data from the SDB to test compliance with ship operating limits ;
- updates the DDB after producing an acceptable load plan ;
- performs optimization of the cargo and water ballast distribution. The objective function is usually the trim, which is either prescribed by the user or is communicated to the LP from other KBSSHIP subsystems, such as the EVP ;
- includes a rule base able to handle data and coded knowledge contained in the other components of the ELS. The rule base is able to handle restrictions and encodes empirical knowledge acquired from experienced ship operators. The strategy used by experts to load chemical and product carriers is implemented in terms of Prolog rules in the prototype. The provi-

sion of an explanation facility showing in a succinct and concise way the reasoning steps to reach the final load plan will be implemented ;
- accepts user requests (via the KUI) overriding the load plan produced by his rule base. This is a necessary feature, since ship operators are often reluctant to let a computer program perform their job.

The Cargo Handling Unit (CHU)
The main task of the CHU is to prepare a plan showing the sequence of loading/unloading operations and all actions and precautions to be taken by ship or shore personnel during these operations. It is generally known, that for most load conditions stability and strength limits are satisfied, since they have been already investigated in the Trim and Stability Manual. However, intermediate loading stages may become dangerous as large bending stresses may arise due to a sequence of filled and empty tanks. Further, trim has to be continuously monitored to facilitate discharging. Therefore, detailed instructions for the loading of the ship including intermediate steps are necessary. To accomplish this task the CHU :

- uses information contained in the CKB related to the voyage schedule with respect to loading/unloading at intermediate ports. Further, it draws information from the RKB, the EKB, the PKB and the GDB ;
- places requests to the AM to carry out additional stability and strength calculations to test critical values during intermediate stages. The CHU detects critical phenomena (e.g. high hull stresses, loss of stability) during loading/unloading and prescribes progressive ballasting/deballasting to alleviate them ;
- updates the LKB with the operation sequence plan ;
- includes a rule base capable to handle suitable encoded empirical knowledge. The rule base of the CHU and the associated inference mechanism exhibit similar characteristics to the rule base of the LP ;
- accepts user requests prescribing operations departing from the sequence produced by the CHU.

The Algorithmic Module (AM)
The Algorithmic Module contains all routines implementing typical tasks of a conventional Loadmaster such as trim, stability and strength calculations. The AM reads data from the GDB and the DDB and updates the SDB. The main output is the final floating condition expressed in terms of hydrostatic parameters such as drafts at several locations along the ship, the metacentric height and the bending moment and shear force distribution. The AM contains also routines to provide trim, stability and strength data in cases where the watertight intergrity of the ship hull is not preserved due to damage.

All AM routines are written in the C programming language to allow portability in different computer environments using the Unix operating system. Interaction with the databases is performed through appropriate input/output routines, which are separated from the routines implementing the algorithms. Thus, portability between different Data Base Management Systems is ensured.

The Visualization Unit (VU)
The VU is the module of the ELS, which implements the Man Machine Interface (MMI). The VU contains all presentation routines and makes maximum use of the hardware and software environment offered by modern Unix workstations. Plots and graphical representations are made with the PHIGS graphical standard.

Maximum use is made of the windowing technique available in modern workstations. Fig. 2 shows a typical output from the VU of the prototype ELS. Unnecessary information disappears from the screen if it is no more needed. Mimic diagrams of ship piping will be implemented for the visualization of loading/unloading operations. Maximum use of a pointing device (mouse) for input of data is made.

The KBSSHIP and User Interface (KUI)
The KBSSHIP and User Interface is responsible for the communication of the ELS with other KBSSHIP subsystems and the user. The KUI handles all requests by SMES to make available local data to other subsystems and all ELS requests to other subsystems. Further, KUI handles any user supplied data, which affect the ELS data and knowledge bases. All routines translating data from the common to the local representation reside in the KUI.

The Rules and Regulations Knowledge Base (RKB)
This knowledge base contains all rules and regulations by national or international authorities pertaining to the loading, unloading and distribution of liquid chemicals among the cargo tanks of the particular ship. Sources for the above are :

- SOLAS (Safety of Life at Sea).
- International Load Line Convention
- MARPOL (Marine Pollution), [5].
- IMO International Code for the

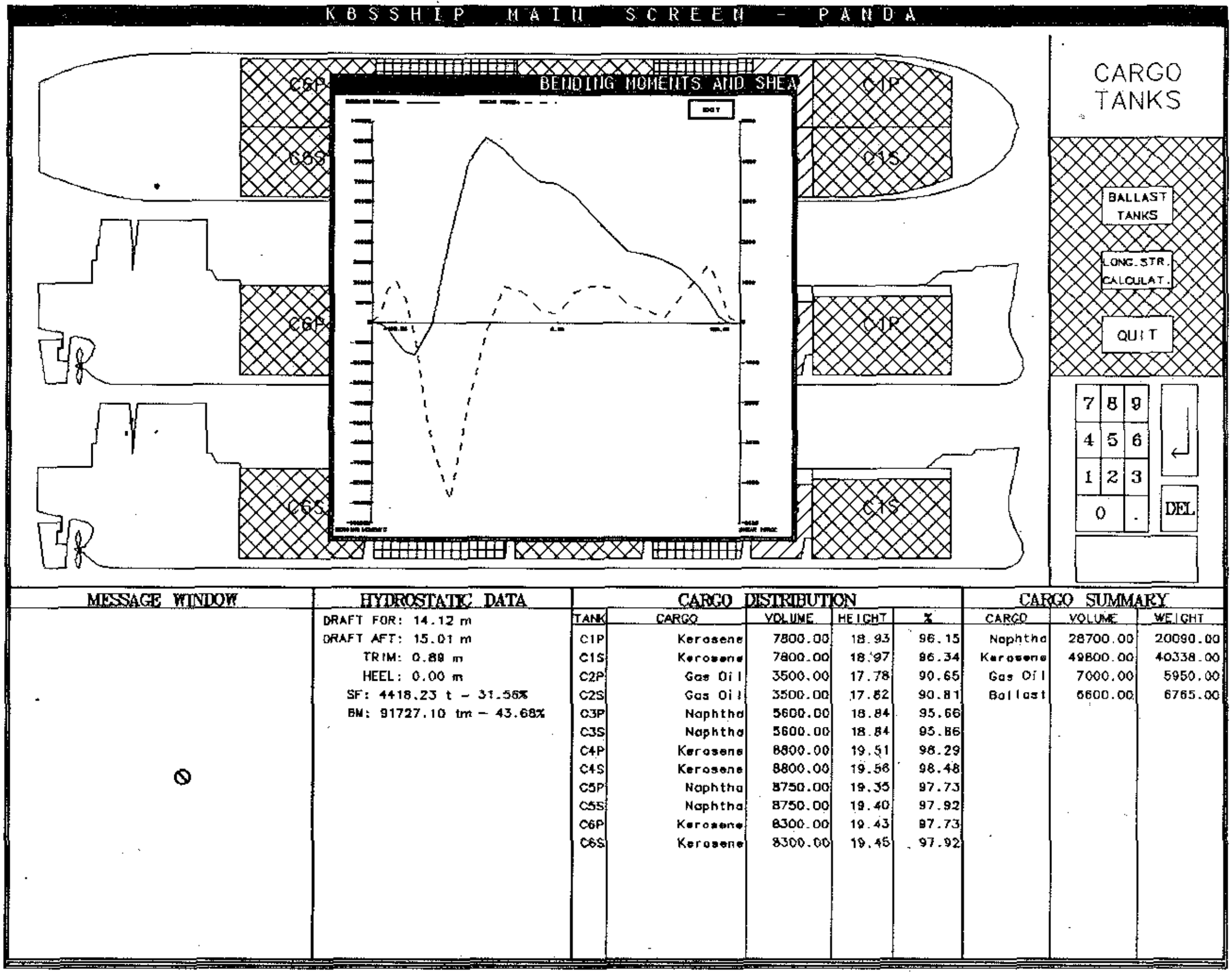

Figure 2. The Man-Machine Interface of the ELS

Construction and Equipment of Ships Carrying Dangerous Chemicals in Bulk, [6].
- International Maritime Dangerous Goods Code, [7].
- specific national or port regulations (e.g. U.S. Coast Guard regulations)

In particular, the RKB contains the description of precautions and special arrangements for handling of liquid cargo grades allowed to be carried by the specific ship, usually incorporated as notes in the Procedures and Arrangements Manual.

The Ship and Tank Geometry Data Base (GDB)

The GDB contains all geometrical data pertaining to the ship hull and the liquid cargo tanks, which are necessary for the calculation of trim, stability and strength data by the AM. The data are organized in files of the indexed sequential type. The structure adopted in the implementation enables the easy and quick access to the data. Such data are, for example, tabulated Bonjean curves for a number of ship stations and a number of drafts, tabulated capacity plans for all liquid tanks, etc.

The Charter Knowledge Base (CKB)

In the CKB all data related to charter requirements which affect either the load plan or the sequence of operations during loading/unloading are stored. Such data include, for example, types and amounts of chemical products, ports of call and amount of cargo to be discharged or loaded at each port, special cargo treatment required (heating, mixing with grades of similar type), etc. The CKB may be either filled by the user or draw information from the the KBSSHIP environment.

The Products Knowledge Base (PKB)

This knowledge base contains physical and chemical properties of chemical products allowed to be carried by the ship. More specifically, information about :

- physical and physiological properties such as colour, specific gravity, viscosity, boiling point, flash point, upper and lower explosive limits, MARPOL category, effects on the human organism upon skin contact or vapour inhalation, etc ;
- chemical formula, pH value, water solubility ;
- chemical reactivity with other

products and sea or fresh water
- compatibility with tank coatings and ship piping ;
- precautions to be taken during transportation such as heating, blanketing, vapour release ;

is included. It should be emphasized that the above list is by no means exhaustive. The structure of PKB enables incorporation of new type of information, which will be eventually needed during the course of implementation.

The Cargo Handling Equipment Knowledge Base (EKB)
The EKB contains all information pertinent to the cargo handling equipment, especially those characteristics related to interaction between the equipment and the liquid cargo. Such information is, for example :

- type, material and operating curves of cargo pumps ;
- object oriented description of piping arrangements, cargo pipe material ;
- location, type, material of pipe fittings such as valves, manifolds, gaskets, stuffing boxes.

The Stability and Strength Data Base (SDB)
This data base accepts the results of the calculations from the Algorithmic Module. These data include, among others :

- the draft at a number of ship stations in the final floating condition ;
- stability data such as the longitudinal and vertical position of the center of gravity, the metacentric height and the GZ vs heel curve ;
- shear force and bending moment distribution along the ship.

The SDB is fed exclusively by the AM in response to a request placed by the LP to perform standard stability and strength calculations. On the other side, the information contained in the SDB is available to any ELS module. Further, a data link to other KBSSHIP subsystems is established via the KUI.

The Cargo Distribution Data Base (DDB)
The DDB is essentially a table containing the quantity of liquid cargo stored in each ship tank for a particular voyage. Further, the DDB contains some statical data such as lightship, machinery and equipment weight distribution. Other weight categories, such as consumables and crew provisions are also contained.

The DDB is updated by the Loading Planner or by user requests through KUI. The information contained in the DDB is made available to the ELS modules and other KBSSHIP subsystems.

The Loading/Unloading Operations Knowledge Base (LKB)
This knowledge base accepts a description of the sequence of operations during loading/unloading of the ship as produced by the CHU. In particular, the LKB contains the loading sequence for each individual tank.

The Instrument Interface (INI)
The Instrument Interface handles all incoming data from instruments installed in ship tanks, like level gauges, cargo temperature and pressure in the tank.

KNOWLEDGE REPRESENTATION IN ELS

Domain knowledge of the ELS is modeled using the PHOS (Procedural and Heuristic knowledge Organization using Structural primitives) knowledge representation formalism, [8], developed in NRCS "DEMOCRITOS". PHOS is a conceptual formalism, which provides a uniform framework for dealing both with structural and procedural knowledge. The way knowledge is structured as well as the reasoning mechanism in PHOS are akin to those of semantic nets, [9].

Knowledge Representation
Objects and procedures are represented in a uniform way as concepts. A concept C is described in terms of its attributes, c_i. Attributes are attached to concepts with the primitive link "attr". An attribute may be associated with a pair (A,a) consisting of another concept, A, and an attribute a of A. This pair specifies the set of values an attribute c_i may take and is connected to c_i via the "vr" (value restriction) link. In some cases, the attribute a of the pair (A,a) may not be specified, thus associating the entire concept A to the attribute c_i. For example, the concept TANK (Fig. 3) has a 'code', a 'coating_material', a 'content', the 'volume_occupied' by the tank content and the tank 'capacity' as attributes. The attribute 'code' of the concept TANK is associated with the pair (STRING,'char*'), the attribute 'coating_material' with the pair (MATERIAL,'code') and the attributes 'volume_occupied' and 'capacity' with (REAL,'value'). The attribute 'content' is connected to the entire concept PRODUCT. Moreover, specific values may be assigned to attributes of a concept, for example in order to describe in-

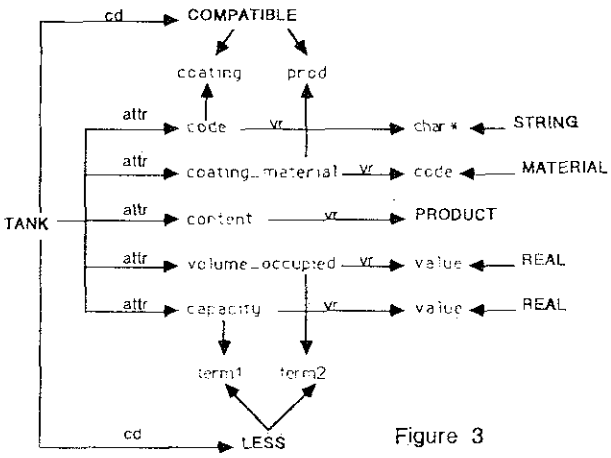

Figure 3

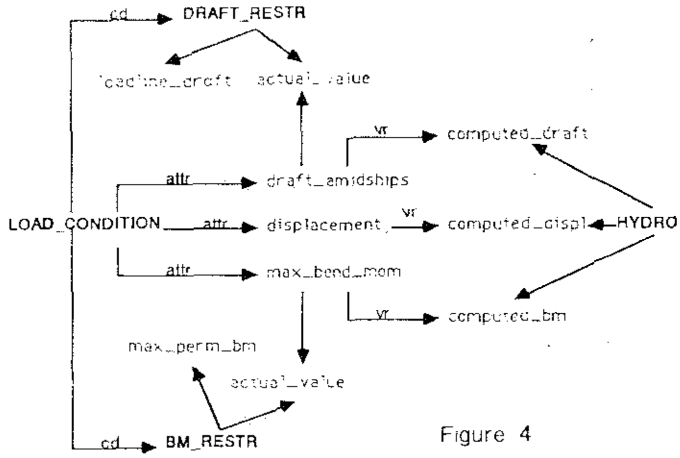

Figure 4

dividual tanks.

Another type of link used for defining concepts in PHOS is the "cd" (conditional dependency) link. The "cd" link specifies conditions for the attributes of a concept in order for that concept to be valid. For example, the requirement that the 'volume_occupied' should be always less than the 'capacity' of the tank is expressed by a "cd" link connecting the concept TANK to the LESS concept. The definition of a "cd" link requires further the specification of those attributes of the concept TANK, which are related to the attributes of the concept LESS. Compatibility between tank 'coating_material' and tank 'content' is enforced by a "cd" link between the concepts TANK and COMPATIBLE. Fig. 4 shows another concept, LOAD_CONDITION. For this concept to be valid, the draft amidships should be less than a maximum value, as implied by the Loadline Convention. This restriction is expressed by the "cd" link between the concepts LOAD_CONDITION and DRAFT_RESTR. The concept HYDRO is a primitive concept, which cannot be further detailed with PHOS. This concept is attached to a C routine delivering the computed draft, displacement, bending moment, etc.

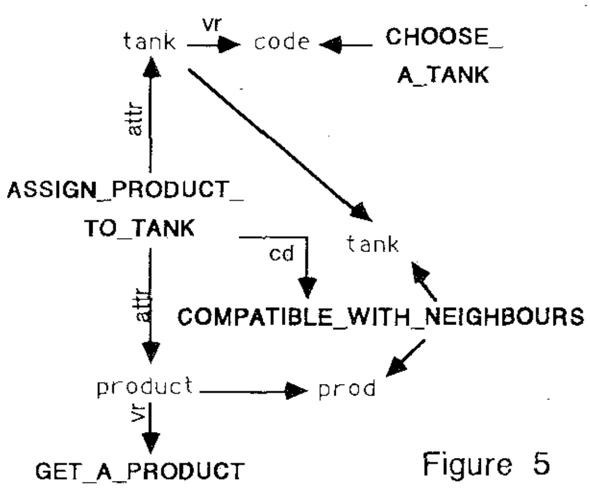

Figure 5

Fig. 5 shows an example of a procedure described by the concept ASSIGN_PRODUCT_TO_TANK. The concept CHOOSE_A_TANK is also a procedure. The concept COMPATIBLE_WITH_NEIGHBOURS associates a tank with a product and is valid if this product does not react chemically with the contents of all neighbour tanks. This concept is attached to a Prolog rule.

Concepts are organized in a hierarchical structure. The hierarchical relation among concepts is realized by "is_a" links. The child concept inherits the definition of the parent concept and refines this definition by eventually adding attributes, new "vr" or "cd" links or specific values to attributes. Fig. 6 shows such a hierarchy with TANK as the parent concept.

<u>Reasoning in PHOS</u>

In PHOS, concepts are templates instantiated during execution, i.e. during evaluation of a concept, the inference engine assigns particular values to its attributes. These values must be in accordance with restrictions imposed by "vr" and "cd" links. In order to evaluate the attributes of a concept, a depth-first search is performed to traverse the hierarchical network. When the engine reaches a terminal concept P, then the attributes of P

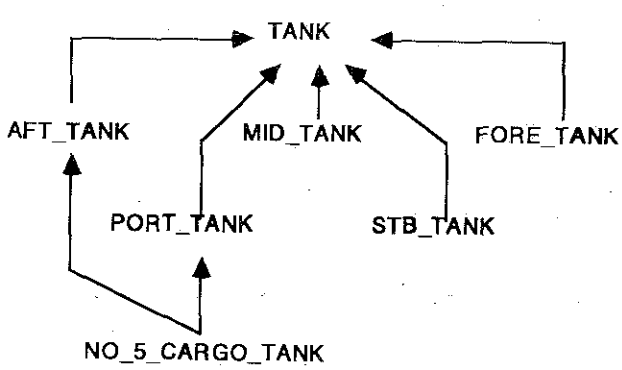

Figure 6

are either assigned those values that have been specified as particular values or they are evaluated through their "vr" links. Then, the conditions attached to the concept P through "cd" links are checked. Backtracking delivers alternative solutions.

CONCLUSIONS

In this work, we have investigated the possibility of applying artificial intelligence techniques to solve a complex problem, that of loading chemical and product carriers. An expert system for the solution of the above problem, the Expert Loading System, has been designed. The system differs from commercially available Loadmasters in the sense that a solution to the cargo allocation and the planning of sequence of operations during loading/unloading is provided. This solution is found using encoded knowledge used by experts in the field. Knowledge elicitation sessions have clarified specific problems encountered in loading of the above ship types.

The structure of the proposed system is modular, enabling easy customization to a specific ship. Algorithms implementing common tasks such as calculation of stability and longitudinal strength data and presentation of results are grouped in modules separated from and subordinated to expert modules performing reasoning. Domain knowledge is modeled with a powerful tool enabling uniform representation of objects and actions or procedures. The prototype developed has shown the viability of the concepts. It is believed, that the problem of loading of any ship type, in particular loading of a container ship, may be handled in the same manner, i.e. using the ELS skeleton structure of Fig. 1, provided that domain knowledge is acquired and coded appropriately.

Finally, the structure of the ELS enables integration within the KBSSHIP environment, which is an open system capable to accomodate several systems onboard ships.

LIST OF ABBREVIATIONS

AM = Algorithmic Module
CHU = Cargo Handling Unit
CKB = Charter Knowledge Base
DDB = cargo Distribution Data Base
EDS = Expert Diagnostic System
EKB = Equipment Knowledge Base
ELS = Expert Loading System
EMS = Expert Maintenance System
EVP = Expert Voyage Pilot
GDB = ship and tank Geometry Data Base
INI = Instrument Interface
ISRS = Information Storage and Retrieval System
KBSSHIP = Knowledge-Based Systems onboard SHIPs
KUI = KBSSHIP and User Interface
LKB = Loading/unloading operations Knowledge Base
LP = Loading Planner
PHOS = Procedural and Heuristic knowledge Organization using Structuring Primitives
PKB = Product Knowledge Base
RKB = Rules and regulations Knowledge Base
SCES = Statutory requirements Classification Expert System
SDB = Stability and strength Data Base
SMES = System Manager Expert System
VU = Visualization Unit

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ON THE POSSIBILITY OF ESTABLISHING RATIONAL STABILITY CRITERIA

Lech Kobylinski

The paper deals with problems connected with the possibility of establishing rational stability criteria. Present stability criteria, IMO criteria, national criteria as well as philosophy behind those criteria are criticized. Proposals to establish stability criteria on the basis of probability of capsizing and difficulties involved with practical application of this method are discussed and various pragmatic approaches to solve this problem are pointed out. Discussions within the working group on stability as well as discussions at STAB Conferences related to stability are summarized and various proposals and philosophies arising from those discussions are mentioned, Proposals of adopting system approach to safety of ships against capsizing is finally advanced and author elaborates on this point.

INTRODUCTION

The purpose of developing of stability criteria is to achieve safety of ships against capsizing and loss. Criteria may consist of a set of minimal values of stability parameters for use by designers and operators the satisfying of which should ensure safety against capsizing. Those minimal values may also appear as a result of certain calculating procedures or as a result model test performed according to certain specified procedures. During last three decades tremendous effort has been exercised aimed at developing stability criteria and as a result of this effort some international and national requirements and recommendations containing such criteria were developed. However, there is common agreement, that although those criteria increased in general level of safety against capsizing, they are not fully satisfactory and there still exists considerable risk of capsizing even if those criteria are satisfied. This is the reason that the Working group on within the IMO Sub-Committee on Stability, Load Lines and on Fishing Vessels Safety is working on "improved" or "rational" stability criteria.

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The meaning of "rational" criteria is not fully understood. In the meaning which was adopted by the IMO working group, rational criteria mean criteria taking into account all external forces acting on ship in a seaway and physics of capsizing. However, broader understanding, as according to Oxford Dictionary where "rational" means "sensible", that can be tested by reasoning" does not exclude e.g. criteria obtained by means of statistic or on the basis of model tests, or by any other sensible method. Therefore, considering possibilities of developing rational criteria we should not exclude all possible methods.

Critical remarks with respect to the present stability criteria.

Stability criteria are at present included in the international and national requirements and recommendations. International criteria developed by IMO were thoroughly discussed in the papers by Jens and Kobylinski [1] and by Plaza and Petrov [2] as well as in several other papers. National criteria exist-

-ing in several countries were described and compared by Jens [3] and by Lugovsky[4], Specific requirements of some countries developed later were discussed in other papers (e.g. [5 , 6]).

All existing criteria hardly could be defined as being "rational". IMO requirements consist of two basic sets. Criteria developed in the period 1964-1968 and included in the resolutions A. 167 and A. 168 were based on a semi-statistical method where stability parameters of two groups of ships - - those which were capsized and those which were considered to be operated safely - were compared and conclusions were drawn from this comparision. Criterion included in the resolution A. 562 and in some other resolutions applicable to certain ship types is based on the calculation of the wind heeling moment and rolling angle.

Regarding national criteria of countries, where such criteria do exist, they consist of a set of critical values of stability parameters as in IMO resolutions A. 167 and A. 168 and/or of some kind of weather criterion in the form basically similar to IMO weather criterion.

The main drawback of the semi-statistical method used in the developing of IMO criteria in the form of a set of critical values of stability parameters consist of the fact that the population of ships investigated, and particularly ships which were capsized was rather small. In addition those ships were ~~of~~ different types and sizes, of different age, the circumstances of casualty were widely different and in many cases additional effects, such as shifting of cargo, icing etc. were of importance. The stability characteristics in many cases were also uncertain. Moreover, when establishing critical values, it appeared that quite large proportion of ships having stability parameters in excess of critical values capsized, whereas many ships with stability parameters below critical were considered safe. This critical remarks are well known and were pointed out at many occasions. Similar critics is applicable to any existing criteria consisting of a set of critical values of stability parameters.

On the other hand weather criterion in the form included in IMO resolution A. 562 and also in some national requirements is based on the calculation of the wind heeling moment acting on the ship taking account of the rolling angle due to waves, assuming the ship is in beam seas.

Generally the physical model in this approach is highly simplified, moreover, wind pressure and angle of rolling were choosen so that the criterion is satisfied for ships considered safe.

In IMO resolutions and in some national requirements there are also included some other criteria based on calculation of heeling moments due to external forces, however all of them include arbitrary assumptions and a reaching simplifications.

In this context it is worth while to quote the words of Professor Bishop at RINA Ginger Group in 1986: [7].

"Design rules for stability are still based on forlorn attempts to describe the actual process of capsizing without understanding the physics, treat hudrostatic and directional stability as quite distinct phenomena and ignore the possible coincidence of resonance and dynamic instability. Constantly trying to polish these rules up is really rather waste of time and money".

Probability of capsizing during ship's lifetime.

All phenomena occuring in reality are basically of a random character and data on accidents enable a posteriori risk assesment therefore it seems logical to adopt risk level as a basis for safety assessment, safety criteria and operational procedures. This is particularly applicable to the risk of capsizing, because majority of factors affectingstability, such as wind and wave forces, sea currents, and centre of mass related to loading condition are obviously of random character.

Safety, understood as safety against accidents concerning ships as well as people and environment is a conception which might be measured in the probabilistic sense.

The probabilistic approach to safety against capsizing was advocated by several authors [8, 9, 10, 11] although all of them drew attention to the basic difficulties involved by this approach.

The most attractive application of this concept would be calculation of the probability of capsizing (or loss of stability) accident during the whole lifetime of the ship. This concept was for the first time proposed probably by Firsov [12]. It was repeated by Sevastyanov [10, 13] and the author [11, 14]. General probability concept with regard to capsizing was given also by Krappinger [15] and Kastner [16].

Not going into detail's of this concept it must be pointed out, however, that the aplication of this method is possible if the function λ_o, which is frequency of accidents in any situation met during the lifetime of the ship is known. Theoretically, this function could be estimated by three different methods [11, 14] :

1. Formulation of the suitable mathematical model of capsizing and performing systematic calculations.

2. Conducting model tests of capsizing in various external conditions.

3. Collecting sufficient statistical data.

Neither of the three above methods is practically available, therefore probability of non-capsizing during the whole lifetime of the ship is not possible at present.

In this context it is necessary to define the capsizing or loss of stability accident. From the point of view of safety under loss of stability accident it is understood not only physical capsizing, i.e. bringing the ship to the upside-down position, but also any excessive heeling leading to flooding, loss of control, shifting of cargo etc., which may be considered dangerous to the ship safety. Such definition of the loss of stability accident was suggested by Abicht [17] and Morrall [18] and was also considered during the discussion at the IInd STAB Conference [19] . We understand after Odabasi [20] capsizing or loss of stability accident as exceeding the angle of roll at which situation dangerous to the ship occurs which disables the ship and makes further operation of the ship impossible.

One way to surmount the difficulty in calculating overall probability of non-capsizing is contained in the proposal of the Polish delegation to IMO in 1978 [21] . Similar idea was pursued in other places [11,12,14]. The essence of this idea is calculation of probability of non-capsizing not in all possible during ship's life situations, but only in few of them, considered most dangerous. This leads to the necessity of making choice of those "dangerous situations" and to adopt possible scenarios of capsizing in those situations. Choice of dangerous situations could be made on the basis of analysis of capsizing accidents on the basis of judgement of experts and on consideration of physical phenomena leading to capsizing. This problem was discussed in several papers e.g. [14, 22, 23], and in this context Working Group on Intact Stability of IMO agreed to consider in further studies two situations [24], namely:

1. Ship in beam seas, taking account of severe wind and waves, and water trapped on deck and other possible external forces.

2. Ship in following waves, taking account of pure loss of stability, parametric resonance and broaching and other possible external forces, e.g. water trapped on deck.

More detailed consideration of dangerous scenarios leading to capsize one might find in a paper by Takaishi, [25].

Basically, the concept of dangerous situations applies to the position of the ship in relation to the direction of wind and waves. In calculation of the probability of capsize, probability of occuring various factors simultaneously has also to be taken into account. Cleary and Letourneau [26] quote 34 possible factors which have to be accounted for. This type of analysis is necessary when programming various scenarios of capsizing.

Although the idea of estimating probability of non-capsizing in few choosen dangerous situations appears to be simpler and more attractive as previously mentioned idea of calculating the overall probability of non--capsizing during the whole period of life of the ship, in practical application it meets quite serious difficulties. The main difficulty is the fact, that function λ_o must be known for the choosen situations and neither of the three already mentioned methods of its estimation could be satisfactorily applied as it is shown in the following

Capsizing in beam seas.

Development of the suitable mathematical models of ship rolling in a seaway which ultimately should lead to the probability of exceeding certain limiting angles of heel attracted many scientists and literature of the subject is immense. We may mention only few selected references [27,28,29,30,31] .

In its simplest form the mathematical model of rolling motion in irregular sea consists of one-degree of freedom linear equation which may be written in a general form:

$$\mathcal{J}(\ddot{\varphi}) + D(\dot{\varphi}) + R(\varphi) = K(t)$$

Where φ is roll angle, $\dot{\varphi}$ is roll velocity, $\ddot{\varphi}$ is roll acceleration, $\mathcal{J}$ is inertia term, D - damping term, R - restoring term and K - excitation term due to waves.The equation of this kind is being studied in frequency domain. The probability of exceeding certain limiting angle of roll considered as loss of stability accident, $\varphi_{crit.}$, could be estimated easily if $m_{o\varphi}$, i.e. variance of the probability density distribution of rolling amplitudes, which is Rayleigh distribution, is known:

$$P\{\varphi_A \geqslant \varphi_{crit.}\} = \exp\left(\frac{\varphi_A^2}{2m_{o\varphi}^2}\right)$$

Linear equation of rolling motion is a good approximation only if amplitudes of roll are small. With increasing rolling amplitudes such as may be dangerous from the point of view of stability this approximation is useless and more appropriate non-linear equation has to be used, which may be written in the form:

$$\mathcal{J}(\ddot{\varphi},t) + D(\dot{\varphi},t) + R(\varphi,t) = K(\varphi,t)$$

Where all terms are functions of time t. This form of equation if neglecting the damping term, could be transformed to the well known Mathieu equation. Prediction of long--term statistics of roll motion has been worked out by Roberts and Standing [32], however this method does not allow prediction of extreeme rolling angles which may cause capsizing. The calculation of the extreme rolling angles requires however, study in the time domain using reliable mathematical model which takes into account of the non-linearities of restoring and damping terms. Specific difficulty arises because of the necessity to approximate properly the righting arm curve.

In the most recent work [33] Nayfeh and Sanchez investigated the equation of rolling in the form:

$$(\mathcal{J}+\delta\mathcal{J})\ddot{\varphi} + D(\dot{\varphi}) + \left[1+\frac{\delta_{GM}}{GM}\cos(\omega_E t+\delta)\right]\cdot R(\varphi) = \mathcal{J}F\,\omega_E^2\cos\omega_E t$$

with the static restoring moment approximated by:

$$R(\varphi) = r_1\varphi + r_3\varphi^3 + r_5\varphi^5 + \cdots$$

and damping moment equal to:

$$D(\dot{\varphi}) = d_1\dot{\varphi} + d_3\dot{\varphi}^3$$

Using digital and analog-computer simulation in the time domain authors have demonstrated the occurence of many complicated rolling responses to regular seas such as competing attractors, period-multiplying bifurcations, chaotic motions, capsizing, and revealed sensivity to initial conditions.

Practical application of such mathematical models and computer simulation is, however, questionable. In any simulation of this kind there are assumed fixed initial conditions and fixed mathematical model. In reality, small changes of coefficients, caused by changes of trim, heel, wave surface and possibly other factors influence also initial conditions as well as the mathematical model therefore singularities revealed by the simulation might not occur. This might be the main difficulty in attempting to calculate extreeme rolling angles, notwithstanding the fact that any calculation method must be transformed into simple calculation procedure if it has to be used as design criterion.

Capsizing in following seas.

Motion of a ship in following seas, is considered to be a dangerous situation. Three specific phenomena may be involved in this situation:

- pure loss of stability
- parametric resonance
- broaching

Pure loss of stability with a ship positioned on the wave crest could be a simple calculation even if the dynamic phenomena are taken into account. Several practical methods of such calculation are known and results were presented by many authors. Apart from the classical work by Paulling [34] we may mention only more recent studies by Mamamoto and Nomoto [35] and Helas [36], the last containing also proposal for a criterion taking this phenomenon into account. But papers mentioned are few of the many considering this problem.

We owe to Grim [37] the first theory of parametric excitation of a ship moving on longtudinal seaway which is due to changes of the stability characteristic in time. The rolling motion equation in this case would be:

$$J\ddot{\varphi} + D\dot{\varphi} + R\,(\overline{GM}_m + \delta\overline{GM}\sin\omega_E t)\varphi = 0$$

which could be easily transformed into Mathieu equation. There were several proposals to utilize this equation for developing stability criterion, the one most advanced being probably proposed by the Abicht [17,38] The method is, however, complicated and gives approximate results because of the simplifications adopted. Broaching in following and quartering seas which often leads to capsizing has been studied inter allia by Boese [39], Motora et al [40] and more recently by Bielanski [41], who proposed a criterion of capsizing based on broaching which, however, could not be used in practice. Summing up, the attempts to develop stabilitycriterion based on phenomena occuring when a ship is moving on longitudinal seaway did not lead to results which can be applied in practice, the more so they do not allow to estimate the function λ_o required to calculate probability of capsizing in this situation.

Model tests.

Further possibility of estimating the function λ_o and of developing stability criteria are model tests of capsizing. Model tests of capsizing which require long runs on realistical irregular sea were conducted in open water. Work done in United States, Germany, Poland and Japan may be mentioned, the results of those investigations being published in many articles. Only a selection of them might be reffered to [42,43,44,45]. Basically, in this method an attempt was made to calculate probability of capsizing for a model in certain wind and sea condition and in relation to ship parameters /GM_o, freeboard etc./. Theoretically this method may provide reliable results but it requires tremendous effort and is extreemely expensive. In authors opinion it might be effectively used only to investigate physics of various modes of capsizing. In this context valuable observations were made as pointed out in [14] and also by Dudziak in [46].

Bearing in mind difficulties in conducting model tests of capsizing in open water, Blume and Hattendorf [47] performed model tests of capsizing in the towing tank in controlled conditions and made an attempt to elaborate on this basis a stability criterion for certain types of ships. This was proposed to IMO, but up to the present time IMO SLF Subcommittee could not reach any conclusive decision in this respect.

Also recently published paper by Grochowalski [48] must be mentioned describing tests of capsizing of captive and free running model of a fishing vessel conducted with the aim of investigating physics of capsizing in quartering waves.

During 17th ITTC [49] proposal was advanced and discussed to adopt standard procedure of model tests of capsizing in order to assess safety against capsizing. This proposal, however, was not pursued further.

Statistical methods.

Stability criteria could be developed using statistical method which could be defined as discrimination analysis. This method is based on stability data for ship capsized and for those, which were operated safely. To some degree this method was used by Benjamin [50] and Rahola [51] and more recently, by IMO. However, rigorous discrimination analysis was used only in the second part of the IMO analysis [52]. This kind of analysis was extended by Krappinger and Sharma to two, three and more parameters systems [53]. Although statistics of casualties might be the most appriopriate method for estimating the function λ_o, in practice sufficient number of statistical data could not be collected. Critical remarks with respect to the present IMO criteria which are based on semi-statistical analysis were included in Chapter 2 of this paper, and not much more could be expected from further analyses at this kind as was shown in [54].

Balance of heeling and restoring moments.

The most attractive approach to the development of capsizing criteria might be the method widely used in other fields of technology which is based on comparing demand and capacity of the system. Both demand and capacity of a system are generally random quantities. In this concept failure of the system is estimated by the probability that the demand - D - of the system exceeds its capacity - C -. Safe condition would be then:

$$P_S = 1 - P[D > C] = P[D \leqslant C]$$

Demand and capacity, being random quantities are defined by density functions.

Comparision of demand and capacity of a system could be done also in deterministic way. In deterministic method mean or characteristic values of demand and capacity are compared and safe condition is determined as:

$$P_S = C^* > D^*$$

Where C* and D* are characteristic values, arbitrarily choosen.

It is virtually impossible to apply the above described method to safety against capsizing, because probability density functions for demand and capability are strongly coupled. The comparision of the characteristic values of demand and capacity /deterministic method/ constituted a basis for majority stability criteria in several requirements and recommendations, national and international. The best example of this is the so called weather criterion which appears in stability regulations adopted by USSR and in some other countries and quite recently by IMO. This approach has some merit, but might be also critisised as being oversimplified, not taking into account the effect of waves and with main parameters arbitralily choosen [55].

Other stability criteria, taking into account crowding passengers on one side, heeling caused by action of the helm, tow tripping of the towing hawser may be dealt with more adequately by deterministic method.

Other concepts and proposals to improve existing criteria.

Some other concepts to develop new or improve the existing criteria were advanced.

An interesting proposal in this respect is a concept of "test track" by Deakins et al [56]. In analogy to the testing car prototypes on the test track they propose to test ships on the route designed to represent series of potentially dangerous situations from the stability point of view. A ship may be qualified safe if it succesfully passes the test track without reaching potentially dangerous angles of roll. This proposal did not induce any significant response.

An elaborated proposal to develop stability criteria was proposed by Kuo et al [57]. This proposal is based on energy balance and takes into account physical phenomena of capsizing under the influence of wind and waves. No practical utilization of the ideas proposed is known. It was already mentioned the proposal of a stability criterion based on model tests advanced by Blume et al [47]. The criterion was intended mainly for large container ships. Also Helas [36] proposed a criterion additional to the existing IMO criteria based on loss of stability on the wave crest. Finally we may mention a criterion which takes into account water trapped on deck which was proposed by Rakhmanin to the IMO Sub-Committee [58]. Also criterion proposed by Bielański [41] based on broaching may be mentioned. This would be almost complete list, if we do not take into account deterministic criteria due to crowding of passengers, towing force, rudder force etc. which appear in some national regulations. Neither of the above mentioned proposals found practical application, although to some of them attracted serious consideration by the IMO Sub-Committee.

IMO work on improved criteria and system of safety.

Bearing in mind critics of the established stability criteria IMO Sub-Committee concerned quite long time ago decided to work on "rational" or "improved" stability criteria. With this purpose a working group was established. The work on these criteria from the very beginning was rather erratic. The working group had difficulties to define what "rational" in the context of criteria really means and how such criteria could be developed. Therefore it was almost impossible to work out a comprehensive work programme aimed at development of such criteria. Nevertheless several attempts were made and finally a long-term programme of work was agreed in 1974 [59]. This programme included basic theoretical studies as well as comparative calculations and model tests. However work according to this programme has never been commenced. This may be understood because solution of problems included in the programme required employment of the resources of many research institutions over many years. Until results of researches were known IMO could hardly do anything in this direction.

Under the pressure of some governments which felt that improvement of the existing IMO criteria is necessary the IMO Sub-Committee concerned adopted more pragmatic approach and a short-term work programme. The result of this short-term programme was development of the "weather" criterion, criteria for some specific ship types and other small improvements.

It was realized at IMO that despite three international conferences on stability and extensive research programmes accomplished in some countries the development of stability criteria based on rigorous physical models is still not much closer than it was twenty years ago. It was also realized that increasing safety against capsizing could be achieved, however, taking other measures. From this consideration the idea of developing a code of stability for all types of ships has arisen. This code is under development at present [60]. Existing Stability criteria are first of all design criteria related to stability parameters of the ship. Satisfying those criteria does not mean that the ship is safe from loss of stability accident. This is because casualty depends not only on stability characteristics, which are very importnant, but also on many other factors, such as operation, construction, environment, information available on board and on human being ability. Therefore considering safety it is necessary to consider the whole system of safety. The idea that the development of the Code should be based on system approach was advanced by the author [61].

Analysis of stability casualties reveals that shifting of cargo, water trapped on deck are important factors. Those factors might

be attributed to operational errors. Importance of operational factors on safety is obvious, although the necessity of including operational requirements in the system of safety was pointed out recently by Dahle and Nedrelid [62] and Kastner [63].
The fact, that design criteria alone could not ensure safety was indirectly recognized long time ago by including into safety regulations requirements concerning stability information, freeing ports, hatches, covers and sills and also concerning some operational procedures. However systematic consideration of four basic elements of the system of safety, namely: external loads, ship, crew and shore service is a new approach. Without doubt safety against capsizing depends on all elements of the system and because of that the recently developed by IMO Code of Stability will include all these elements.

CONCLUSIONS

1. It is generally recognized that present stability criteria are inadequate and in order to increase safety of ships criteria defined as "rational" should be developed.

2. The most appriopriate rational criteria should be based on the calculation of the probability of non-capsizing and risk analysis.

3. All available methods of calculation of accidents frequency function required in the probabilistic approach could not be applied at present and provide adequate results in the foreseable future.

4. As safety against capsizing depends not only on design criteria but also on other factors equally important, system approach to safety seems to be the way to increase safety.

5. As a practical solution which will increase safety against capsizing and at the same time will facilitate work of designers and operators, code of stability based on system approach may be developed.

Actually, such code is under preparation by IMO.

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INSTRUMENTAL CONTROL OF STABILITY AND OPERATIONAL SAFETY OF SEAGOING SHIPS

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The influence of gusty wind and sea waves on seagoing ship is one of the main reasons of her accidents and failures. Oscillating motions caused by these spells result in significant displacements, velocities and accelerations negatively influencing not only the on-board equipment and systems, ship's crew and passengers, but the safety of a ship as a whole as well. There is an obvious necessity to control the above-mentioned dynamic parameters at different 'dangerous' points of a ship to predict or (if it is possible) to prevent hard consequences, including capsizing.

The set of the developed instrumental means is designated for the usage of information concerning ship's motions to control ship's gears, fishprocessing equipment, operational stability system, etc. The instrumental means are devided into three groups: indicators, analysers and advisers. Some results of instruments tests are given.

1. INTRODUCTION

Dynamic stability is the inherent stability of a ship at sea as she reacts to the dynamic combination of forces acting upon she. These forces include those generated by wind and wave motions, current, icing conditions, the speed of the vessel through the water and by shifting cargo, the free surface caused by flooding or from fluids in tanks (such as fuel or water).

Dynamic stability is influenced by vessel design, its possibility to absorb and damp external energy, and the way cargo is loaded, etc. The captains operational or vessel manoeuvering decision are also an important factor.

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There is at present a lack in the theory of capsizal avoidance. Consequently there is a lack of instrumental means and mathematical language in which to express and communicate intuition, which experienced pilots possess, of how to handle a ship in heavy weather.

For most ship types, deciding on the correct capsizal control action is not only the matter of common sense and good seamanship. This complex problem place greater demands on crew and corresponding instrumental means of stability and operational safety control, as the wrong or even late action may often make the situation critical.

Instrumental means for the ship stability and operational safety control are designated not only to provide a seafarer with current 'Safety level' information but to produce also some possible advices. It might be so that the intuitive or practical knowledge of one experienced pilot could have saved another.

Whether or not such instrumental means will be of great use remains to be seen, because to make practically useful quantitative predictions it is still necessary to reach more precize calculus of all mentioned parameters, which influence the level of ship operational safety

2. BACKGROUND

Capsizing with attending human losses do occur more frequently than we should wish [4]. The safety of life at sea problem is extremely actual for small craft accidents. There permanently exists high probability of instant capsizing of small vessels and the letter causes the immense risk of human losses. Limited sizes of small ships do not permit installation of effective active or passive rescue means. Moreover, small craft crew-members as a rule have relatively low qualification and little practice of navigation in storming sea. But even highly skilled shipmasters will have rather different and subjective judgement as to the risk of capsizing, to the dangerous intensity of wave and wind loads, especially in following waves condition.

There exist some instrumental means for static stability control, such as Stalodicator, Loadicator, Wesmar's SC 44 Stability computer, etc. But it should be mentioned that systems of current control of ship storm safety do not exist so far in maritime practice. The developments at Nikolajev Shipbuilding Institute are the result of very extensive efforts of research team headed by doctors V.Nekrasov (theory) and Yu.Zhukov (R&D). The instrumental means of dynamic stability control can be easily incorporated into integrated bridge control systems.

3. THEORETICAL BASIS

Theoretical research of ships seafaring has been intensively carried out at the NSI since mid 70's. By now the foundation of ship sailing safety concept and ships reliability in high seas are practically completed.

The main components of the concept are [5]:

- the theory of transformation of random fields of wind pressure and chaotically disturbed fluids into processes of nonlinear ships' motions;
- the theory of non-local and local stability of ships' motion processes in stormy conditions and under the action of other loads, occuring in ship's operation;
- ship's reliability theory from the stability point of view.

With the help of these theoretical assumptions the set of functional limitations have been formulated for the properties of wave and wind stability, ensuring maximum values of ship's reliability factors under given operation conditions. These functional limitations can be controlled by instrumental means.

First of all, the existing techniques of the probabilistic parameters estimation and stability analysiswere investigated. Statistical mamenta of the 1-st and 2-nd order were obtained on the basis of original nonlinear equations of ship's motions [5]. The equations of the 2-nd order statistical momenta are the balance equations of the various kinds of ship's motion energy. They allow to answer the problem of assessment of extremely possible energy balance with the account of potential resources of the ship , this balance being the criterion of non-local stability.

The relevant analytical technique was developed for the following situatins to be calculated:

- broadside to the wind and 'smooth' waves;
- broadside to the wind and waves with breaking crests;
- on astern seas.

Analytical expressions for the criteria of non-local stability were obtained. A great number of computations and their comparison with the results of seaworthy nautical trials were made [3].

The relevant method of permissible wind and wave conditions evaluation has been approved by National Classification Societies. At present the activity is carried out to develop international standard within the framework of ISO TC/188 'Small Craft'.

To solve the above mentioned problems we have to answer at least the next questions:

(1) What wind and wave load intensity is dangerous from ship's dynamic stability loss point of view?

(2) What information identifies the level of risk of ship dynamic stability loss?

(3) What parts of captain decision-making process can be automated?

Instrumental means for ship dynamic control systems can be discussed afterwards.

4. DANGEROUS WIND-SEA EXCITATIONS

There exist some occasions of ship's dynamic stability loss. Let us primarily pay dirrect attention to the two of them (concerning beam seas) to illustrate the possibility of instrumental ship dynamic stability control:

(1) In the absence of the wind load the ship capsizes only under beam wave (sea) action.

(2) The ship capsizes under wind and beam wave sea action.

Corresponding criteria of non-local stability for both occasions under consideration were obtained on the basis of successive application of the dynamic systems theory and the theory of their movement stability at random disturbances [2].

The first criterion (named " volnostojkost Y " according to [2]) is essentially the criterion of wave (sea) steadiness (Ss). The second one (named " vetrovolnostojkost Kп " according to [2]) is correspodingly the criterion of wind and wave steadiness (Sw). It is evident that under Ss<1.0 the waves capsize the ship. And if Ss>1.0, but Sw<1.0 then the ship is capsized by the wind action.

The above mentioned criteria enables us to assess quantitatively the risk of ship's capsizal in real load and seas conditions, that is to determine dangerous (from the ship's dynamic stability loss point of view) wind and wave parameters .

Let us consider some main correlations between the steadiness criteria and ship's internal load and external excitations parameters [2,3].

In general Ss criterion can be determined by equation:

$$S_s=(D\cdot d_m)/(3\cdot D_{av}\cdot I_{xa}).$$

Correspondingly

$$S_w=(D\cdot l_m\cdot R_d)/M_r ,$$

where Rd is comlex functional of Ss criterion [2]:

$$R_d=(1-0.25\cdot SQR(\frac{1}{S_s}))\cdot SQR(1-SQR(\frac{1}{S_s}))$$

As can be easily seen the value of each criterion depends on values of ship load parameters (D, dm, lm, Ixa) and of parameters Dav and Mr, determined by values of external excitations (h3%, Vw, τ), which can approach the dangerous limits from ship's dynamic stability loss point of view. The subject of this paper gives priority to the analysis of correlation between indicated dangerous limits and ship's motions parameters, which can be measured. Let us illustrate such correlation by one rather simple example [3](see fig.1,2).

As weather condition get worse (h3% increases) maximum relative rolling angle θm increases, but values of Ss and Sw criteria correspodingly decrease. In load condition 1 the ship capsizes under the wave (sea) action at h3%=1.8 m and θm=37°. In load condition 2 the ship capsizes under wind and wave action at h3%=3.0 m and θm=45°.

So we can draw the following preliminary conclusions:

1. The level of ships current safety can be assessed with the help of Ss and Sw calculation and comparison of corresponding θm with that measured.

2. The results of dynamic stability assessment radically depends upon the accuracy and reliability of culculations of mentioned parameters for real ship's load conditions and of measuring of corresponding θm (or other strictly correlated parameters).

3. In general dynamic stability assessment required more complicated calculations and measurings, but it's beyond limits of this very paper.

5. HUMAN FACTORS AND SHIP'S STORM SAFETY

In stormy weather conditions ship's crew and passengers, onboard equipment and systems experience intensive motions influence, which results in discomfort, reduction of capacity for work, loss of attention by human beings, and also in worsening of equipment and electronics operation conditions, reduction of their efficiency, etc. All named negative consequences of intensive motions are followed with errors in navigation and machine operation, traumatism and accidents and so on. And at last the arrorsin captain decision-making process as to assessment of dangerous weather conditions and of ship's dynamic stability loss risk can result in capsizing and human losses.

That is why shipmaster needs some special instrumental means offering objective information concerning ship motions and reliable prediction of possible consequences. Such means can help captain to make for example the following decisions:

- to change the course or the speed of the ship;
- to use the stabilizing tanks or roll damping device;
- to take in or detach ballast;
- to break up fishing and to leave for harbour, port or for open sea (escaping shallow water breaking waves);
- to break some kinds of onboard activities (to prevent possible traumatic consequences);
- to start the rescue operations, etc.

According to the purposes mentioned we can differ three types of corresponding instrumental means:

(1) - simpliest ship's motions indicators (SSMI);

(2) - microprocessor ship's motions analyser (MSMA);

(3) - computer aided shipmaster adviser (CASA).

Evidently the named instrumental means have rather different nomenclature, sensors, hardware and software. Futhermore different means must be installed on the different types of ships. It depends upon the ship's size and parameters, on navigational aids composition, ballast or stabilising tanks, roll damping devices, on number and qualification of the crew, etc.

6. INSTRUMENTAL MEANS NOMENCLATURE

Let us consider some possible structures of named instrumental means for current ship dynamic stability control, which were developed at Small Craft Department of the Nikolajev Shipbuilding Institute.

SSMI "TRITON-I" is used for current control of displacements, velocities and accelerations to ensure the safe crew working conditions, normal equipment operation, preliminary danger of capsizal alarm, etc. It consists of four main blocks:

(1) - block of current initial information sensors and transmitters;

(2) - block of special parameters current values input;

(3) - block of analogue information processing and comparison;

(4) - block of indication.

Initial information consists of values of ship motions parameters. The list of special parameters contains either above mentioned ship load parameters or simply the number of possible ship's load condition (in case it is finite and preliminary determined). In the third block initial information is transformed into the set of slowly changing electronic signals which can be compared with corresponding limited electronic signals determined in the same block on the basis of special parameters current values. The block of indication may contain different number of three-colour scales (green -yellow -red) each for indication of relative motions level at concrete ship's room (bridge, engine room, auxiliary machinery room, fish processing room, etc). But it may consist of only one three-colour scale with special selector of ship's room to be controlled. SSMI "TRITON-I" provide the shipmaster with visualized results of ship motions analysis. Green signal corresponds to completely safe operation conditions, while red one corresponds to dangerous ship's motions level (for example Sw<1.0 or Ss<1.0).

Yellow signal is used to warn and attract captain's attention to approaching danger.

MSMA "TRITON-M" solve more wide list of problems and there exists the set of analyser's modifications. It has correspondingly more complicated structure and the block functions. It is nesessery only to mention here, that it is based on the specific microprocessor with flexible software. The analyser can function autonomously or in man-machine mode, it realizes global current safety level control, controls the operation of active roll damping devices, equipment stabilisers, etc. It has additional channel for initial information input (for example from the data base on Floppy disc). Results are displaid to the special screen. The analyser also solves all problems and tasks peculiar to SSMI "TRITON-I".

CASA "TRITON-A" is based on personal computer usage and is a multy-purpose system, it enables to solve rather complex navigational problems considering the weather forcast and possible ship's routes. In some situations the system is able to realize computer aided control (for example transfer operations on the high seas, operation of onboard launching and retrieval device, etc.). The system contains aditional sensors such as draught, trim and heeling transmitters, etc. The system also solves all tasks and problems peculiar to SSMI "TRITON-I" and MSMA "TRITON-M".

The limitations of Classification Societies (concerning conditions of equipment operation, peculiarities of electronics and so on) must be considered in the course of development of all modifications of named instrumental means.

SSMI "TRITON-I" was tested in real sea sailing conditions on three small vessels of length from 4 to 30 m. Some results of the test are illustrated and can be analysed in present paper (see Fig. 1). Let us consider the indicated area $\theta*$ of maximum relative rolling angles measured in the short time period previouse to the ship's capsizal. It can be easily seen that corresponding calculated angle $\theta_m=37^\circ$ and it is located within the indicated area $\theta*=36\div42^\circ$. Evidently the accuracy of the developed indicator is completely sufficient for practical purposes.

7. CONCLUSIONS

(1) Practical possibility to realize the objective current ship's dynamic stability control is strictly connected with the development of special theory of quantitative assessment and prognosis of ships motions and dangerous wind and wave excitations from the ship's capsizal point of view .

(2) Actual is also the development and perfection of corresponding set of instrumental means.

(3) It is obviously rational to develop the mentioned units and systems on the unified theoretical, technical and element base.

(4) Ship's motions indicator "TRITON-I" proved to be sufficient for practical purposes and can be recomended for use on small vessels as the first step towards the introduction of instrumental means for ship's dynamic stability control.

(5) Further progress in the field under consideration is conditioned by perfection of needed sensors and corresponding software.

8. LIST OF NOTATIONS

D Ship's displacement

d_m Arm of maximum of dynamic stability curve

D_{av} Dispersion of rolling angle velocities

I_{xa} Inertia moment of ship mass and added water mass about the longitudinal axis of the ship

S_s Wave (sea) steadiness criterion

S_w Wind and wave steadiness criterion lm Arm of maximum of righting arm curve

Mr Average wind heeling moment

h3% Wave height of 3% excess probability

Vw Average wind velocity

τm Wave mean period

SQR(·) Square root function of variable (·)

θ_m Maximum relative rolling angle

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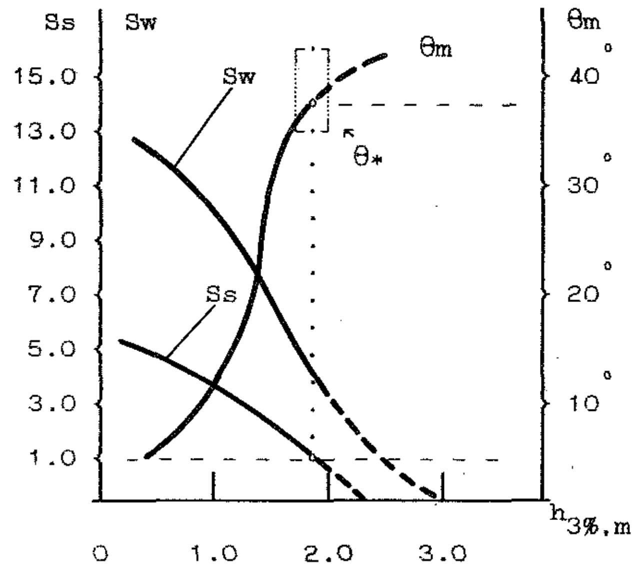

Fig. 1 Dynamic stability parameters of small ship (load condition 1)

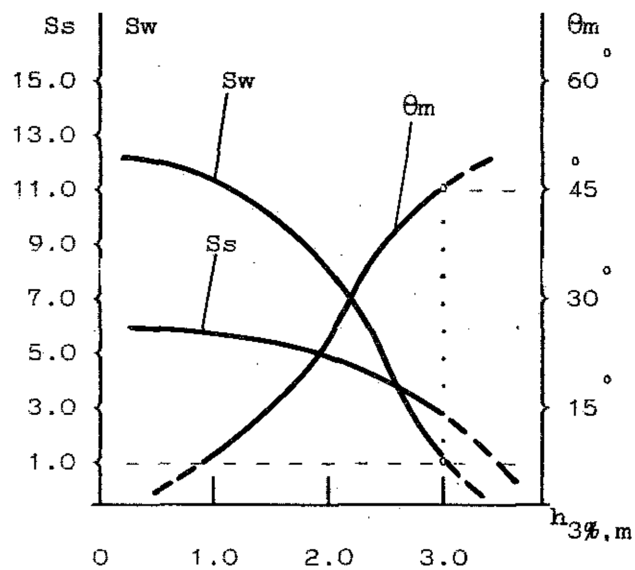

Fig. 2 Dynamic stability parameters of small ship (load condition 2)

AN INVESTIGATION INTO THE COMBINED EFFECTS OF TRANSVERSE AND DIRECTIONAL STABILITIES ON VESSEL SAFETY

D VASSALOS AND K SPYROU

The paper reviews briefly previous attempts to account for the influences of directional stability on transverse stability before putting forward a procedure whereby the two can be treated in parallel and their combined effects on safety investigated. Application of the proposed procedure to a wide beam riverboat is considered and the results are presented and discussed. The investigation undertaken clearly indicates that under certain circumstances directional instability can lead to large heeling angles and possible loss of control, causing a serious threat to vessel safety.

INTRODUCTION

In ship stability studies it is common to investigate the occurrence of loss of stability, which may eventually lead to capsizing, as a single cause-effect relationship. In many cases, however, such an approach is not supported by practical experience, since capsizing can frequently be the final result of combined actions, appearing in a chain of events which gradually reduce the safety margin of the vessel. The HERALD OF FREE ENTERPRISE disaster can be cited as a typical situation where factors such as high speed in shallow water, trim by the bow, loss of control, human error and inadequate subdivision, played a critical role in leading to the capsize.

To be able to define scenarios and then satisfactorily model all the phenomena contributing to loss of transverse stability, together with their interactions, would be the ideal case, but this represents an admittedly immense task at present. For certain situations, however, like the combined effect of transverse and directional stability, a qualitative understanding of the mechanism generating a high risk situation can be obtained with the current state of knowledge.

This paper, focusing on the last point, represents an initial stage of wider scale research on ship stability, ship controllability and their inter-relationship under various environmental conditions. On the basis of such a perspective, single GM-based stability criteria could be subject to re-evaluation. If, for example, "adequate" GM implied also a largely directionally unstable vessel, then the risk of losing control and hence being subjected to unfavourable external excitations possibly leading to significant reductions in GM would be higher. At this level of complexity, to attempt to quantify the overall threat would, of course, be very difficult, but thinking in terms of risk, it could very well be possible that what in the first instance looks safer, could in real life be exactly the opposite.

The investigation presented herein is

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based on a detailed surge-sway-yaw-roll non-linear mathematical model where, by making use of appropriate numerical techniques, time simulation, steady states continuation and analysis of stability properties of each steady state are performed. With these tools the effects of trim by bow, GM and forward speed on both the directional and transverse stability are examined and thereafter the occurrence of dangerous situations as represented by large heeling angles, is identified.

In particular, the study puts emphasis on the non-linear character of the behaviour of the system and its possible consequences for safety.

BACKGROUND

Problems of coupling between planar motions and rolling in ships operating at high speed are well known and have in many cases been the subject of investigation, either from a stability or from a manoeuvrability point of view, e.g. [1], [2], [3], [4]. For this reason, various mathematical models have been developed, usually in three (sway, yaw, roll) or in four (surge, sway, yaw, roll) degrees of freedom. A number of these are linear, e.g. [5], [6], and others non-linear, [2], [7]. The narrower topic of the relationship between transverse and directional stability, however, has not received sufficient attention, with the exception perhaps of the literature dedicated to the broaching-to phenomenon, e.g. [8], [9].

In a recent investigation, [6], regions of instability were identified for the coupled sway-yaw-roll motions of a ship trimmed by the bow (in which condition directional stability is generally poor). In this paper, it was suggested that bow-trim-induced instability can cause large "divergent" or "oscillatory" motions, possibly leading to capsize, even though the metacentric height is positive. It was, however, not possible to gain further insight as the approach was limited by its very nature, trying to explain purely non-linear phenomena with linear theory tools. The local analysis performed allowed only for the identification of the existence of unstable equilibria. It could not, for example, predict the convergence of the system to a neighbouring stable state characterised by a steady rate of turn and heeling angle when an initial perturbation was applied to it at an unstable equilibrium position (see Fig. 1).

Emphasis on identifying these nearby stable states in the phase space and their basins of attraction would be far more important from a safety point of view rather than testing for instability at the upright zero rate of turn position, which is a rather common phenomenon for operating ships.

MATHEMATICAL MODEL

For the purpose of this investigation, a non-linear modular mathematical model was used, with four degrees of freedom, as given below (see also Fig 2):

$$
\begin{aligned}
&\text{surge: } m(\dot{u} - rv - x_G r^2 + z_G p r) &&= X_H + X_R + X_P \\
&\text{sway : } m(\dot{v} + ru + x_G \dot{r} - z_G \dot{p}) &&= Y_H + Y_R \qquad (1) \\
&\text{yaw : } I_z \dot{r} + m x_G (\dot{v} + ru) &&= N_H + N_R \\
&\text{roll : } I_x \dot{p} - m z_G (\dot{v} + ru) &&= K_H + K_R
\end{aligned}
$$

where for the hull forces and moments:

$$X_H = X_{\dot{u}} \dot{u} - Y_{\dot{v}} v r - Y_{\dot{r}} r^2 + X_{vr} v r + \mathrm{Res}(u)$$

$$
\begin{aligned}
Y_H = {} & Y_{\dot{v}} \dot{v} + Y_{\dot{r}} \dot{r} + Y_v v U + Y_r r U + \\
& + Y_{vv} v|v| + Y_{rv} v|r| + Y_{rr} r|r| + \\
& + Y_\varphi \varphi U + Y_{v\varphi} v|\varphi| + Y_{r\varphi} r|\varphi|
\end{aligned}
$$

$$
\begin{aligned}
N_H = {} & N_{\dot{r}} \dot{r} + N_{\dot{v}} \dot{v} + N_r r U + N_v v U + \\
& + N_{rr} r|r| + N_{rrv} (r^2 v / U) + \\
& + N_{vvr} (v^2 r / U) + N_\varphi \varphi U^2 + \\
& + N_{v\varphi} v|\varphi| U + N_{r\varphi} r|\varphi| U
\end{aligned}
$$

$$K_H = K_{\dot{p}}\ddot{\varphi} + C(\dot{\varphi}) + R(\varphi) - z_Y Y_H$$

The manoeuvring coefficients were calculated according to [10] and [11]. The terms accounting for the effect of heeling on the sway and yaw linear coefficients ($N_{v\varphi}$, $N_{r\varphi}$ etc) were taken from [7]. The yaw damping $C(\dot{\varphi})$ was calculated as suggested in [12] while the restoring moment was derived from the hull sections at various heeling angles with a seventh order polynomial being fitted to the results.

In addition, the propeller generated thrust was modelled by :

$$X_P = (1-t)\rho n^2 D^4 K_T$$

with

$$K_T = C_0 + C_1 J_p + C_2 J_p^2$$

For the rudder forces and moments, expressions from [7] were used, e.g.

$$N_R = -(1+a_H) x_R F_N \cos\delta$$

The rudder to hull interaction term a_H was taken from [13].

For the effect of trim on hydrodynamic coefficients, expressions are available from [10] or [14]. Trim correction factors from [10] are more moderate and they were preferred for this study.

STABILITY ANALYSIS

After a number of transformations, equations (1) can be brought in the vector form:

$$\dot{\mathbf{z}} = \mathbf{F}(\mathbf{z},\mathbf{a}) \tag{2}$$

where, $\mathbf{z}$ is the state variables vector, $\mathbf{z} = (u,v,r,p,\varphi)^T$ and $\mathbf{a}$ is a control parameter, varying independently, which can be any of the rudder angle, trim, and propeller rate.

At points of static equilibrium $\dot{\mathbf{z}}$ must be zero and (2) gives:

$$\mathbf{F}(\mathbf{z}(\mathbf{a}),\mathbf{a}) = 0 \tag{3}$$

To derive the dependence $\mathbf{z}(\mathbf{a})$, a technique borrowed from Bifurcation analysis is used, based on [15], which automatically traces the steady states curve, passing without any difficulty over limit points. The technique is based on a differentiation of equations (3) with respect to a parameter l accounting for the length of the solution curve, followed by a transformation into an initial values integration problem. For stability analysis, local linearisation around each solution point $\mathbf{z}_0$ is performed by substituting $\mathbf{z} = \mathbf{z}_0 + \mathbf{b}$ into (2), where $\mathbf{b}$ is the vector of deviations from the solution $\mathbf{z}_0$, yielding, finally:

$$\dot{\mathbf{b}} = \mathbf{C}\ \mathbf{b}$$

where, $\mathbf{C}$ is the jacobian matrix of $\mathbf{F}$. To test for stability, the signs of the real parts of the eigenvalues of $\mathbf{C}$ are examined, the existence of positive real parts signifying instability, either nodal or oscillatory.

APPLICATION

To apply the above procedures, a riverboat was selected, the main particulars of which are given in Table 1. The following effects were investigated:

a) **Effects of Trim by the Bow**

Three conditions were considered:

- even keel
- 0.4m trim by the bow (0.6 degrees)
- 1.0m trim by the bow (1.53 degrees)

In each case the variation of the steady states of the system for different rudder angles was recorded, followed by the identification of regions of instability (e.g. Figs. 3). As shown in Fig. 4, the dimensions of the instability region are increasing significantly with trim, implying also larger heeling angles at the nearby stable states (in the region of zero rudder angle). Therefore, as the vessel becomes more directionally unstable with bow trim, its stable states keep aloof from the unstable upright position, causing possibly considerable heeling. An important observation was that the maximum steady heeling occurs in general at small rudder angles. The maximum transient heeling also increases with trim (Fig. 4). This arises, however, when the rudder is considerably deflected (around 30 degrees). It must be emphasised that the situation would be further aggravated if the reduction of restoring due to bow trim were taken into consideration.

b) **Effects of Metacentric Height, GM**

In deriving Fig. 4 the value of the metacentric height, GM, was kept consistently high at 2.0m. However, since GM is critical in judging stability, its effect was further examined in the following two cases:

In the first case, GM was related to the vertical loading distribution (KG) and as such it was treated as a parameter independent of the hull form. Figs. 5 to 7 show that the instability loop size, and the heeling angle reduce with increasing GM. The effect on the turning performance of the vessel, however, was characterised by a trend towards a larger tactical diameter (Fig. 8).

In the second case, GM was linked to the hull form. Subsequently, a change of its value would imply also a modification to both the manoeuvring and stability coefficients. Under these circumstances, directional instability was eventually eliminated by reducing GM. Heeling angles during turn, on the other hand, were considerably increased (Fig. 9).

c) **Effects of Speed**

The last factor whose effect was investigated was the initial forward speed. Figs. 10 and 11 indicate that, by increasing speed, the instability loop width is reducing slightly, while loop height and maximum inclinations tend to increase. It is interesting to note that in this case linear theory would predict an increase of instability with speed. If, however, loop width is taken as a measure of instability the reverse is actually true.

CONCLUDING REMARKS

A significant finding based on the work presented in the foregoing is that bow trim can generate almost complete loss of control, in the sense that the instability region grows to the extent of occupying most of the control space of the system (for example, directional instability of $\pm$ 17 degrees was noticed at 1.00m bow trim). If such a situation arises, the pilot is likely to attempt to regain control by setting the rudder to a large angle. If this occurred at high speed, low GM and unfavourable external conditions, large heeling could arise with unpredictable implications.

It is further interesting to note that bow trim would allow for the occurrence of relatively large steady heeling angles at around zero rudder setting. Therefore, even in the absence of any effective control action large inclinations could be experienced as the system would be seeking a rather distant stable equilibrium.

Although the analysis was confined at this stage to the calm sea case, the above ideas bear heavily on a waves environment. Induced trim by the bow is then possible, either periodic (large pitch motion) or steady (the ship travelling with the bow "in the wave", in low frequency following seas). In such a condition the slightest disturbance would cause the ship to veer off course and be brought to the dangerous beam waves position, while the pilot would be unable to apply any correcting action. Should this happen, capsizing could be a likely consequence depending on the vessel conditions.

NOMENCLATURE

m	: ship mass
u, v	: surge, sway, velocities resp.
r, p	: yaw, roll, angular velocities resp.
x_G, y_G, z_G	: Co-ordinates of centre of gravity
X_H, Y_H, N_H, K_H	: surge, sway, yaw roll hull forces and moments, resp.
X_p	: longitudinal propeller force
X_R, Y_R, N_R, K_R	: surge, sway, yaw, roll rudder forces and moments, resp.
Res (u)	: resistance
I_x, I_z	: moments of inertia with respect to x and z axis resp.
$C(\dot{\varphi})$	: hull damping moment
$R(\varphi)$	: restoring moment
U	: ship speed
Z_Y	: Z co-ordinate of the point of action of the lateral force, Y_H
Jp	: propeller advance
K_T	: thrust coefficient
t	: thrust deduction
ρ	: water density
n	: propeller rate of rotation
D	: propeller diameter
x_R	: rudder position
a_H	: hull-to-rudder interaction factor
F_N	: rudder normal force
δ	rudder angle
φ	heeling angle

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TABLE 1 : Vessel's Main Dimension

Type	River-Boat		
Length	L (m)	=	37.45
Beam	B (m)	=	7.93
Draught	T (m)	=	1.52
Block Coeff.	C_b	=	0.632
Service Speed	V_s (kn)	=	10
Downflooding Angle	φd	=	12°

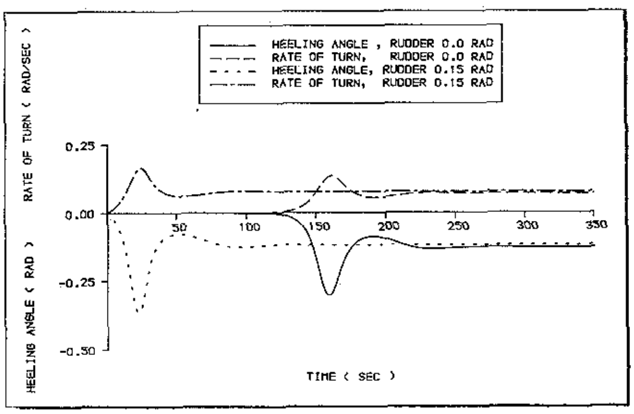


Fig. 1 : Convergence to a nearby stable state : GM = 0.5m

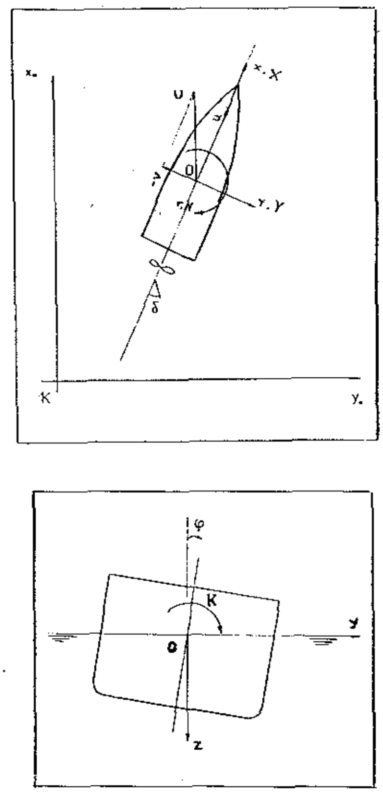
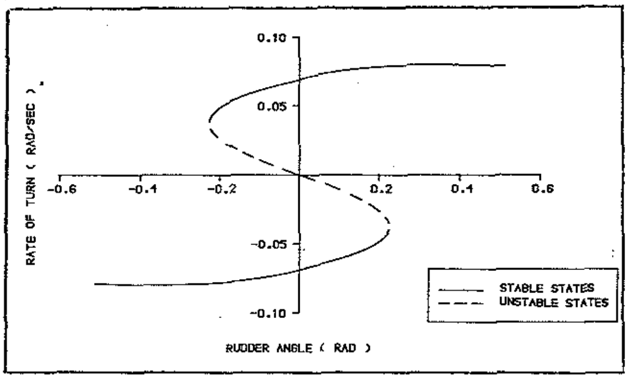
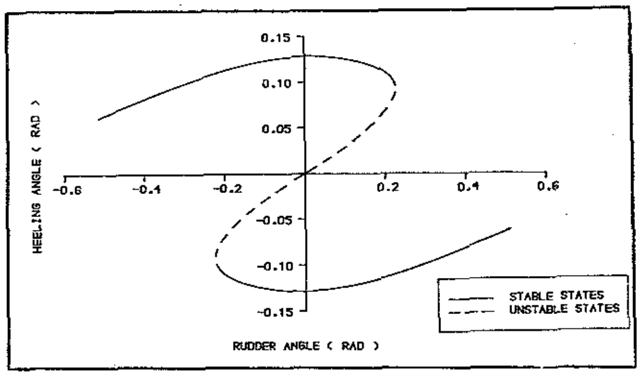


Fig. 2 : Systems of co-ordinates



(3a)



(3b)

Fig. 3 : Region of instability : GM = 0.5m

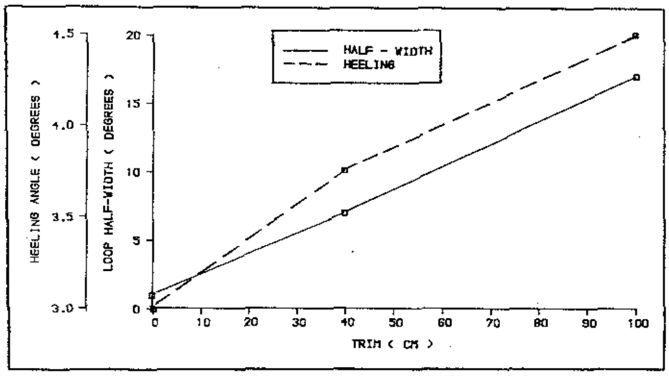
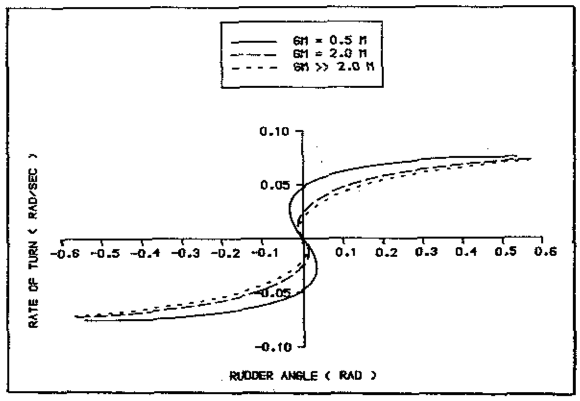
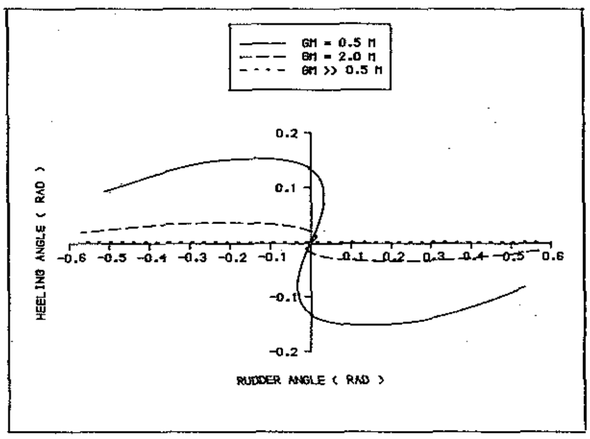


Fig. 4 : Instability loop and transient heeling versus trim : GM = 2.0m



(5a)



(5b)

Fig. 5 : Effect of GM : Trim = 0.0m

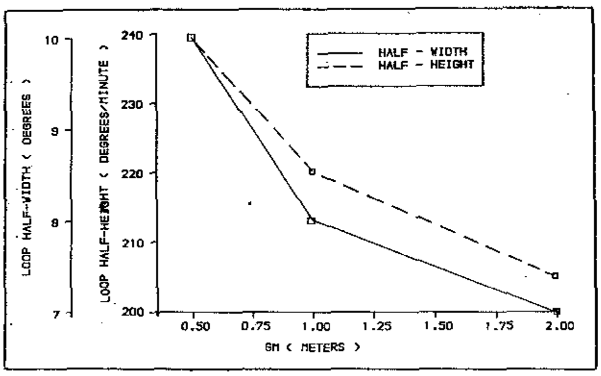


Fig. 6 : Instability loop versus GM : Trim = 0.4m

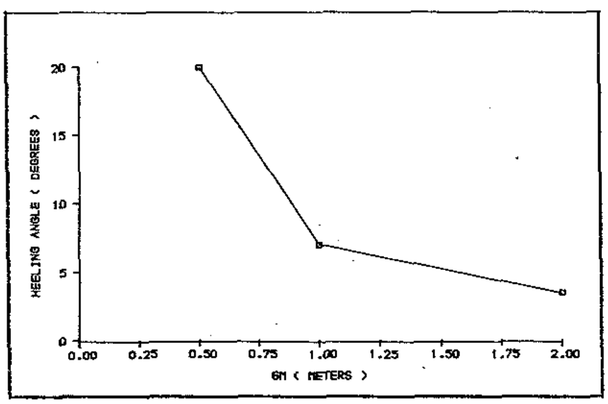
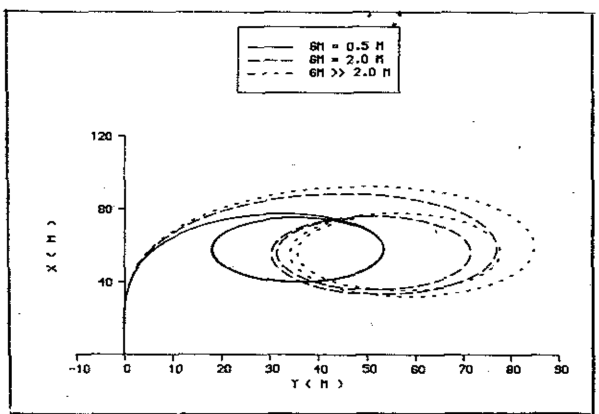
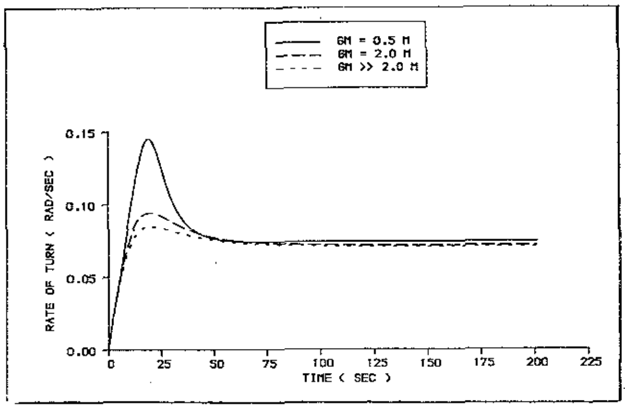


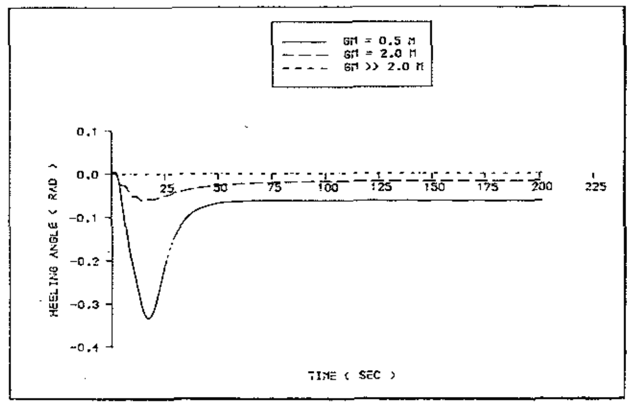
Fig. 7 : Transient heeling versus GM : Trim = 0.4m



(8a)

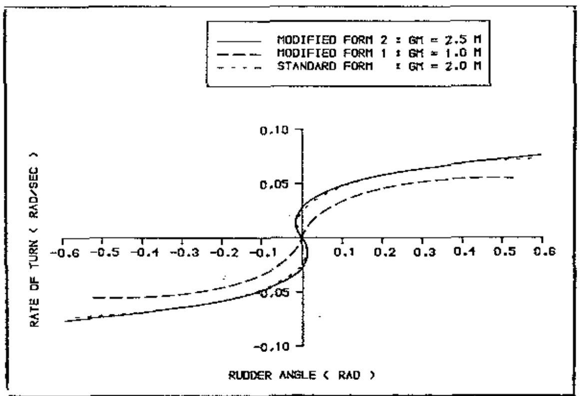


(8b)

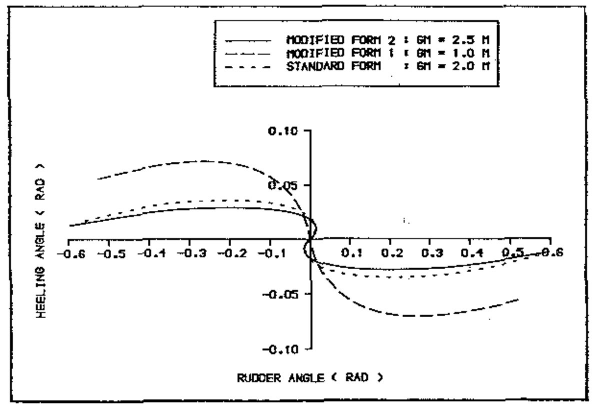


(8c)

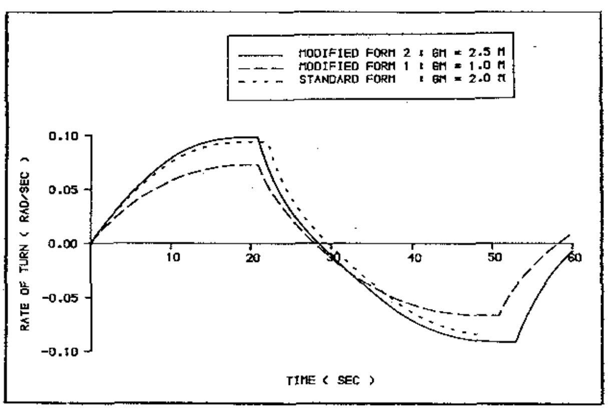
Fig. 8 : Turning manoeuvre



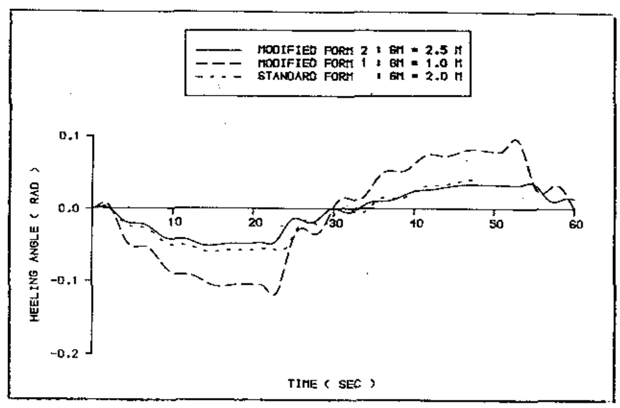
(9a)



(9b)



(9c)



(9d)

Fig. 9 : Effect of GM

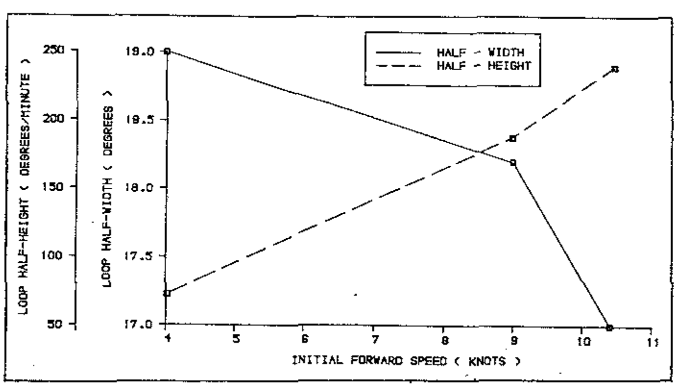


Fig. 10 : Effect of speed on instability region : Trim = 1.0m

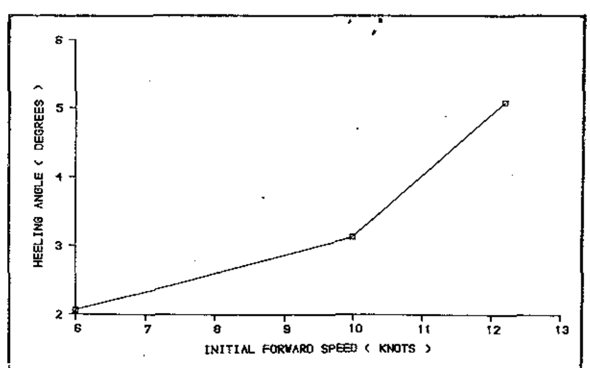


Fig. 11 : Effect of speed on transient inclination : GM = 2.0m

ADVANCES IN THE STABILITY ASSESSMENT OF SEMISUBMERSIBLES

D Vassalos and C Kuo

This paper highlights the co-ordinated research undertaken at the University of Strathclyde on the performance and stability of semisubmersibles, and explains the rationale behind the development of an effective stability assessment procedure for intact and damaged semisubmersibles. The proposed practical stability criteria are outlined and future developments are examined.

1. INTRODUCTION

The semisubmersible has been shown to be one of the most efficient vehicles for offshore operations in hazardous seaways. Operators, however, never cease their attempts to derive even more value from its use, and this inevitably gives rise to questions relating to safety. Safety, in its broadest sense, is a very complex subject but in relation to semisubmersible performance it is stability which requires the major focus of attention. In this respect cost-effectiveness in the exploration and exploitation of offshore resources translates into the rationalisation of semisubmersible stability criteria.

The need to rationalise stability criteria for both intact and damaged conditions, however, demands a more realistic approach to the assessment of semisubmersible stability. In July 1985, in response to this challenge, a research programme entitled Performance Related Efficient Semisubmersible Stability (PRESS) was initiated at Strathclyde University's Marine Technology Centre (SMTC) with support from the Science and Engineering Research Council, the UK Department of Energy, and industrial sponsors. This programme was managed by the Centre and finalised in June 1987.

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It is the objective of this paper to highlight the PRESS programme, and its management, to explain the research work and to discuss the experience gained in integrating the research findings into rational stability criteria for intact and damaged semisubmersibles.

2. THE RESEARCH PROGRAMME

Research Strategy

The stability of a marine vehicle is a very complex matter, involving not only its ability to remain upright in a hazardous environment, but also the ways in which it fulfils its operational requirements without imposing undesirable constraints on other design parameters. Because of this, it has been found that stability research can become a long-term task making little contribution to immediate needs.

Long experience of stability research has led the staff of SMTC to adopt a strategy based on the concept of "Levels of Stability (1). The basic idea of this is to classify the problem of stability assessment into a number of logical assessment blocks of increasing sophistication, each block being related to the quantity and quality of information available and the current state of knowledge. Figure 1 illustrates the concept. The adoption of this strategy allows interim solutions to be available at every stage.

Programme Framework

The overall aim of the PRESS research programme was to carry out fundamental research into factors significantly influencing the performance and stability of semisubmersibles with a view to developing a more rational method of stability assessment for use in practice. The relevant areas of interest were conveniently grouped, as shown in Figure 2, under the headings of "Research", "Interface" and "Output". A brief description of all the PRESS projects is given in Appendix A.

Programme Management

The full complement of researchers and staff comprised the following:

Principal Investigators	-	7
Research Assistants	-	9
Technical Staff	-	2
Computer Programmer	-	1

In addition, two postgraduate students were involved in the programme, having taken certain aspects of semisubmersible stability as research topics for their thesis projects.

The oversight of each research project was in the hands of a principal investigator but responsibility for day-to-day research was the role of his research assistant(s). All the principal investigators and researchers met as a working group at regular intervals to ensure co-ordination of research effort. Individual researchers would make presentations to the group and present concise reports on the progress of each project. Progress was then reported at four-monthly intervals to a Steering Committee. This was composed of representatives from the sponsors, with a chairman selected from amongst its members.

At this level progress was assessed in three ways. The first was to measure technical achievements against a predefined workscope and check them against the agreed milestones. The second was to gauge it against expenditure. Thirdly, researchers made selective presentations to the Committee pertaining to their progress to date.

This approach to project management might appear an obvious one but it is unusual to find such a formal management of projects in a UK academic institution. Further details will be found in Ref. **(2)**.

3. THE RESEARCH APPROACH TO DEVELOPING STABILITY CRITERIA

The approach adopted in developing stability criteria (PRESS 6), is the Energy Balance Aproach, the basic philosophy of which is to derive a quantitative measure of the vessel's stability based on her behaviour in a physically realisable and potentially dangerous situation. It is an approach - better known in ship stability work as the <u>physical approach</u> - that takes explicit cognizance of the influence of such phenomena as wind and waves, but also incorporates some aspects of the vessel dynamics.

To derive a rational, quantitative measure of stability on the basis of a physical approach, it is necessary to consider the following key steps:

a) <u>Potentially Dangerous Situation</u>: There is no dispute over the fact that although the capsize of an intact, upright semisubmersible is highly unlikely, these vehicles <u>can</u> capsize in certain circumstances where a combination of a number of extreme adverse effects is involved.

The research in PRESS 6 was focussed on a free-floating semisubmersible, upright or in a listed state, subject to large rotational motion in an extreme environment represented by steady wind and regular waves.

An energy balance over a critical phase

of the vessel's rotational motions forms the basis of a quantitative assessment of its stability.

b) The Mathematical Model: Following preliminary investigations, see (3), the following single degree of freedom differential equation was considered to be adequate for our purposes:

$$I_v \ddot{\alpha} + c_1 \dot{\alpha} + c_2 \dot{\alpha} |\dot{\alpha}| + R(\alpha, t) = W(\alpha) + M_E(t) \quad (1)$$

Where

I_v = virtual (real + added) rotational inertia

c_1, c_2 = linear and quadratic damping coefficients, respectively

$R(\alpha,t)$ = time-varying restoration/excitation moment

$M_E(t)$ = time-varying external excitation moment

α = angle of inclination

t = time

$\cdot$ = differential w.r.t. time

No claim is being made here that such a mathematical model can describe extreme behaviour over a prolonged period. Extreme behaviour, on the other hand, does not adopt a steady state pattern over prolonged periods either. It is reasonable, then, to assume that extreme behaviour consists of a number of phases, each one of which could be described by the above equation, separated by incidents not described by it which effectively throw the vessel in to a new initial condition.

The solution of (1) cannot be achieved in closed form and a numerical simulation needs to be employed.

Taking the above argument a step further, however, it should be possible to "trigger" a dangerous situation with regard to semisubmersible safety by selecting appropriate initial conditions and thence judge the vehicle's stability based on its ability to survive the said situation. In accordance with the philosophy of the Energy Balance Approach, the motion simulation need cover only the most critical phase of the vessel's extreme behaviour, namely the half-cycle from windward to leeward. Hence, the simulation is localised as shown in Figure 3. The initial conditions and the key parameters describing this potentially most dangerous half-cycle were determined through a parametric investigation, see Table 1 and Ref. (4).

c) Energy Balance: Integrating (1) over the critical half-cycle, beginning with $\dot{\alpha} = 0$ at $\alpha = \alpha_1 < 0$, the energy equation is obtained, yielding:

$$\tfrac{1}{2} I_v \dot{\alpha}^2 + \int_{\alpha_1}^{\alpha} [c_1 \dot{\alpha} + c_2 \dot{\alpha} |\dot{\alpha}| + R(\alpha,t) - W(\alpha) - M_E(t)]\, d\alpha = 0 \quad (2)$$

If the half-cycle ends at $\alpha = \alpha_{max}$ with $\dot{\alpha} = 0$ again, (2) gives:

$$\int_{\alpha_1}^{\alpha_{max}} [c_1 \dot{\alpha} + c_2 \dot{\alpha} |\dot{\alpha}| + R(\alpha,t) - W(\alpha) - M_E(t)]\, d\alpha = 0 \quad (3)$$

The above line integral describes a balance between the energies gained and lost by the vessel over the half-cycle and since the system is non-autonomous the value of the integral will depend on the shape of the motion cycle.

If, now, motion begins at time t_1 and ends at $t_1 + \tau$, (3) becomes:

$$F(\alpha_1, \alpha_{max}) = 0 = \int_{t_1}^{t_1+\tau} [c_1 \dot{\alpha} + c_2 \dot{\alpha} |\dot{\alpha}| + R(\alpha,t) - W(\alpha) - M_E(t)]\, \dot{\alpha}\, dt \quad (4)$$

The value of α_{max} which makes $F(\alpha_1, \alpha_{max}) = 0$ can be compared with a reference angle (for instance, the downflooding angle, α_f) to judge the stability of the vessel.

d) An alternative Approach Based on Ship Experience: An alternative to numerical simulation was adopted in our ship stability research and it proved very effective, see (5), (6). This involved making the assumption that the extreme half-cycle can be modelled by a shape determined from experimental observations, namely a sinusoid. Such an assumption enables the stability criteria derived from an Energy Balance Approach to be presented in a manner similar to the existing stability criteria (i.e., comparison between excitation and restoration areas) and hence facilitates their interpretation.

The sinusoidal function chosen (shown in Figure 4) is as follows:

$$\alpha(t) = \frac{1}{2}(\alpha_1+\alpha_2) + \frac{1}{2}(\alpha_1-\alpha_2)\cos\Omega(t-t_1) \qquad (5)$$

$$t_1 \le t \le t_1 + \frac{\pi}{\Omega}$$

Following the same procedure as in (c), the integral equation (4) now attains the form

$$F(\alpha_1,\alpha_2) = \int_{t_1}^{t_1+\frac{\pi}{\Omega}} \left[C_1\dot{\alpha} + C_2\dot{\alpha}|\dot{\alpha}| + R(\alpha,t) - W(\alpha) - M_E(t)\right]\dot{\alpha}\,dt \qquad (6)$$

Equation (6) represents the Net Area, ie, area under restoring and damping curves minus area under excitation curves. The value of α_2 which makes $F(\alpha_1,\alpha_2)=0$ for $\alpha_2<\alpha_f$ is ensured by demanding, through numerical investigation in each case, that $F(\alpha_1,\alpha_f)>0$ which is similar to the area condition of the existing semisubmersible stability criteria.

e) Proposed Stability Criteria

Taking into account the close link between the stability of a semisubmersible and its operational efficiency, it seems appropriate to suggest that the most suitable presentation of stability criteria would be in the form of maximum allowable KG curves at different draughts and conditions. Furthermore, since the stability assessment procedure does not differentiate between intact and damaged conditions the requirements are essentially the same, the only difference lying in the environmental parameters pertaining to the two conditions.

Using the results of the parametric investigation and on the basis of the arguments outlined above, it is proposed that, in computing the maximum allowable KG values, the following criteria must be adhered to:

Dynamic Stability Assessment (motion simulation over extreme half-cycle)

$$\alpha_{max} \le \alpha_f$$

Quasi-dynamic Stability Assessment (assumed extreme half-cycle):

i) $F(\alpha_1,\alpha_2)=0 \implies \alpha_2 \le \alpha_f$

ii) $F(\alpha_1,\alpha_f)=NA \implies NA \ge 0$

Detailed comparisons, (4), between the two assessments indicated that, in the majority of cases, the dynamic assessment is more stringent than the quasi-dynamic. Both, however, discriminate between the different conditions and designs in the same way. The NA criterion is therefore recommended as being the easiest to compute, understand and apply.

4. TESTING OF THE PROPOSED CRITERIA

The evaluation and interpretation of theoretically derived stability criteria, at whatever level of sophistication, inevitably pose the problem of how to develop confidence in these criteria. The effort directed towards achieving such confidence during this research can be summarised as follows:

Assessment of Representative Semisubmersible Designs

The designs chosen were of the twin pontoon type, the AKER H3, GVA 4000 and SEDCO 700, considered in a number of different conditions, the details of which can be found in (7). The principal findings from these applications were:

a) The proposed criteria were meaningful and discriminate between the various designs.
b) The incorporation of waves and vessel motion in the stability asessment does not yield more restrictive criteria when compared with existing assessments. However, it makes the assessment meaningful and consistent with physical considerations. This, in turn, should

help to eradicate the use of ad-hoc factors in stability criteria.

c) The use of Area Ratios in the existing stability criteria could give misleading results and the Net areas should be used instead as a measure of stability. This is discussed in full in (3).

Comparison with Existing Criteria

A detailed exposition can be found, again in (3). The principal findings from the comparison between the proposed and existing criteria were:

a) Existing intact stability criteria are conservative and damage stability are under-defined when compared with the proposed criteria.

b) With both the existing and the proposed stability criteria the area conditions could, in some cases, be satisfied with zero or even negative GM. Taking into account, however, special phenomena such as tilt, and requirements related to operational procedures and safety, the following points must also be considered:
Intact Stability: Based on the experimental results (PRESS 7), theoretical results (PRESS 1 to 4), and the results of a specially conducted survey (8), it is recommended that GM $\ngtr$ 1m.
Damage Stability: Knowledge of the limiting values of such parameters as the range of positive stability, maximum righting lever and static angle of heel would be of paramount importance. In the current state of knowledge it is not possible to offer meaningful values for these parameters.

Correlation with Model Experiments

The validation of the developed stability assessment procedure was attempted through specially designed model tests and the results presented in (3) provide corroborative evidence. The question of stringency of criteria (old and new), however, still remains.

5. DISCUSSION

The following points deserve consideration:

Organization of Academic Research

Academic research is normally done by small teams with the sole purpose of acquiring knowledge and the transfer of results to practice, if it happens at all, can be a very lengthy process. The approach adopted in this case has several advantages. Firstly, researchers have closer contact with the practical situation and this ensures that there will be a rapid transfer of developed technology from the academic environment to practice. Secondly, external financial contributions allow a number of areas to be tackled simultaneously and this is particularly suitable for a complicated subject such as semisubmsubmersible stability. Thirdly, the organization of the programme is based on the project management techniques used in the offshore industry and thus researchers learn the importance of meeting deadlines, become proficient at in completing tasks within an agreed period, and develop their ability to write concise reports. Such an approach, however, is less effective for work on fundamental problems which requires a longer time span, and unfortunately funding is often available only for short to medium term research/ development projects.

Introducing of Research Advances in Practice

The results obtained here show clearly that for assessing semisubmersible stability it would be more meaningful to employ net area criteria instead of area ratio criteria. The main reason is that more consistent results can be obtained for both intact and damaged conditions by this means. However, the offshore industry is conservative and is likely to be reluctant to adopt new ideas without "more experience" of them. It is therefore difficult to implement any new approach unless organizations such as the International Maritime Organization (IMO) take active steps towards its implementation. IMO, however, will only respond to representations from the various governments involved. It is therefore important for appropriate government departments in each member country to take an active role in promoting more rational stability criteria for semisubmersibles.

Stability as an Aspect of Overall Safety

If rational stability critèria are to be adopted in practice it is important for them to become an integral part of safety assessments. At present, consideration is being widely given to the adoption of a formal safety assessment procedure such as that adopted by the Norwegian Petroleum Directorate. This is an approach that attempts to identify a range of potential failures and seek ways of reducing their occurrence. Clearly, stability assessment must be part of such a procedure and it is hoped that the adoption of a comprehensive formal approach will allow the present advances to be fully incorporated into the total safety framework.

6. CONCLUDING REMARKS

a) Based on an energy balance approach, an effective procedure for assessing the intact and damage stability of semisubmersibles has been developed, which incorporates dynamic information such as regular waves and vessel motion.

b) Deficiencies in the existing methods for assessing stability, such as failure to account for the effect of waves on the restoring moment, lead to inconsistencies when the Area Ratio is used as a quantifiable measure of stability. The Net Area of the excitation/restoration effects appears to overcome such inconsistencies and should be used instead of the Area Ratio.

c) The proposed stability criteria appear to be less stringent in comparison with the existing criteria as regards intact stability, while the reverse is true for damage stability. This is in agreement with a widely-held opinion that the existing intact stability criteria are over-conservative while the damage stability criteria are under-defined.

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APPENDIX A: PRESS PROGRAMME – OBJECTIVES AND EXPECTED OUTPUT

Project 1: **Task – Motions of Listed Semi-submersibles**

Objective: To examine the motion responses of a listed semisubmersible in irregular seaways under the action of wind and currents. The effect of mooring will also be incorporated.

Output: Mathematical analysis of wave loading and a computer program to determine the behaviour of a listed semisubmersible.

Project 2: **Task – Wind Loading Effects**

Objective: To examine interference and shielding effects due to wind loading, using idealised geometrical bodies representing topside structures. Theoretical studies and model experiments will be employed.

Project 3: **Task – Load Variation Effects on Motion Parameters**

Objective: To identify the nature of load variation and the patterns of weight distribution, and to assess the influence of these variations on the metacentric height and mass moments of inertia.

Output: Establishment of bounds on weight-related motion parameters in support of Projects 1, 4 and 6. Provision of load envelopes for operating and survival conditions.

Project 4: **Task – Effects of Large Waves**

Objective: To compute the interaction of large waves and simplified semi-submersible forms in the time domain, to identify the effect on safety of significant parameters

such as the air gap and of non-linear phenomena such as tilt.

Project 5: **Task – Practical Application of Research**

Objective: To integrate the results from the Managed Programme for practical use and to interface in particular with Project 6.

Output: Research results in a format useful for the development of improved stability criteria.

Project 6: **Task – Development of Energy Balance Stability Criteria**

Objective: Using an energy balance approach, to develop and verify a computational procedure for assessing the stability of semisubmersibles, capable of incorporating the effects of wind, regular waves and vessel motions. The findings of Projects 1 to 4 and 7 would be incorporated in the developed assessment procedure with the assistance of Project 5.

Output: Practical stability criteria will be proposed which can be made the basis for improving semisubmersible stability regulations.

Project 7: **Task – Experimental Verification of Stability-related Theoretical Concepts**

Objective: To assist, validate and verify the theoretical calculations and test the proposed stability criteria.

Output: The information gained will be fed back to Projects 1 to 4 and 6 at every stage of the research.

TABLE 1 SUMMARY OF PARAMETRIC INVESTIGATION

PARAMETER		CRITICAL VALUE
VESSEL CONDITION	- Draught	VESSEL CONDITION - Survival
	(a) Dynamic critical axis	No preferred direction
MOTION	(b) Extreme rotat. motion cycle:	
CHARAC-	(i) Extreme angles	5° windwd, dynamic downflooding leeward
TERIS-	(ii) Rotational periods	natural rotation motion period
TICS	(iii) Crit. phase/wave/vessel motion	Max wave slope ~~α~~ as extreme mtn. starts
	(a) Wave length	1.5 x representative semisub length
ENVI-	(b) Wave height	Design storm wave height
RON-	(c) Wave direction	Beam seas
MENT	(d) Wind direction	Quartering
	(e) Combined (wind+wave) direction	Quartering

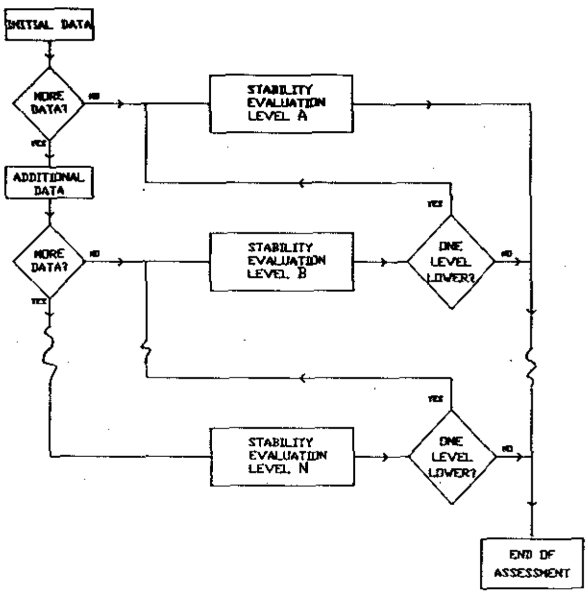

Fig. 1 Concept of "Levels of Stability"

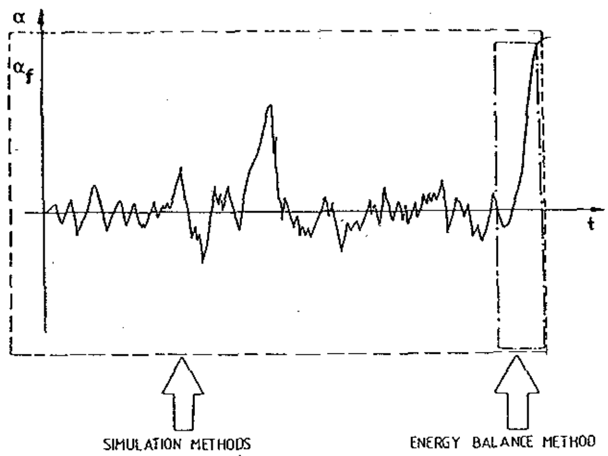

Fig. 3 Energy Balance Philosophy

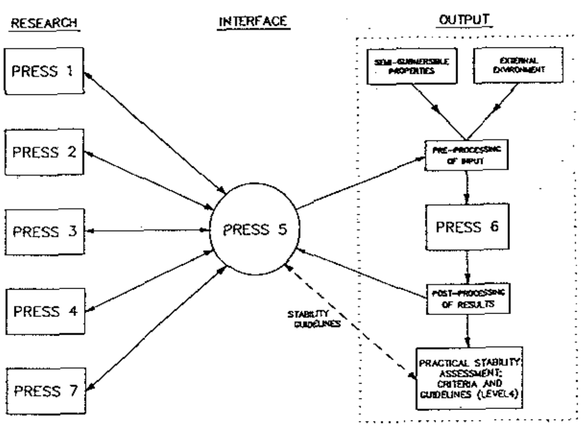

Fig. 2 The PRESS Research Programme

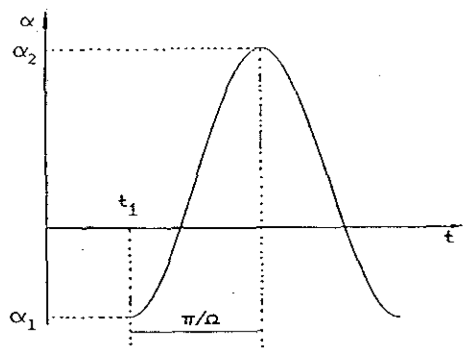

Fig. 4 Sinusoidal Extreme Half-cycle

INTACT STABILITY STANDARDS FOR LARGE SAILING VESSELS

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Mr. Parker E. Marean III[2]

Tall Ships and large sailing vessels (some over 500 gross tons) are being used throughout the world for training, carrying passengers and for public relations work. Many seafarers consider these sail ships to be the most majestic of all vessels, and believe that the love and the lore of the sea, and true seamanship can best be experienced on them. The stability of these vessels are not governed by SOLAS, and in many cases they are not required to meet any national stability criteria, as they are owned and operated as public vessels. However, casualties which have been studied show that criteria are needed to maintain a minimum level of intact stability for all sailing vessels. Because of their configuration, large sailing vessels have vastly different motion characteristics than equally sized motor propelled ships. They are also sailed differently than small sail vessels. The criterion for a tall ship will need to take this into account. This paper proposes a criterion for tall ships and matches it against a historically "good" design. The criterion will take into account the research used in formulating the criterion of the United States Coast Guard and the recently completed research used in formulating the new criterion of the United Kingdom's Department of Transport.

Background

Gone are the days of harbors crowded with tall sailing ships. No longer do you look from the wharfs of Boston or New York City and see hundreds of masts reaching up to the sky as a winter forest without its leaves. To see a tall ship enter a harbor during this century is not a familiar sight and therefore draws much attention. It must be a grand day of celebration to lure more than a dozen tall ships together in one place. Although these statements may have some merit and are able to hold water most of the time there has been renewed interest in sailing large sail ships for pleasure, public relations and especially for performing seamanship training. No matter what the ship is used for, safety and seaworthiness of a tall ship is of utmost importance.

Following the tragic loss of the United Kingdom registered barque-rigged sail training ship *MARQUES*, while competing in the Tall Ships race from Bermuda to Halifax in 1984, an investigation was ordered by the United Kingdom. In the Court's final report[1][3] it was stated "The loss of the *MARQUES* has demonstrated an urgent need to establish a safe and acceptable standard of stability for sail training ships." The search for an acceptable standard for large sail ships has, in general, turned up very little. There were no recommendations by the International Maritime Organization (IMO) after the casualty and no attempt was made to add sail vessels to the international Safety of Life at Sea (SOLAS) and Load Line Conventions (ILLC66). Although the U.S. Coast Guard had a criteria which was derived for sailing passenger ships it had not at that time completed its criteria for sail training ships, and even now the criterion is only for small sail ships of less than 500 gross tons.

Intact stability is one of the most important factors to the overall safety of a sailing vessel, yet it is still the least understood part; which is partly

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[3] Numbers in brackets designate References at end of paper.

due to the subject matter itself. Stability is the very mechanism which gives the sail vessel its ability to move through the water. It is a balance between the heeling forces of the wind (coupled with the lift of the sails) and the righting forces of the ship which exert a vector force to propel the ship. It is our desire to formulate a standard, using the knowledge of sail ship stability which has been acquired thus far, in order to ensure that large oceangoing sail ships ($Lw > 24m$ and Gross Tons > 500) can be designed, built and operated safely. Such a standard may at some future date be appropriate to adopt into the Code of Intact Stability being developed by IMO, so the world's citizens may enjoy a uniform international level of safety for tall ships. We believe the standard should be flexible to allow enough sail for the ship to be maneuverable in light air, while requiring adequate stability for comfortable and safe sailing in steady conditions and a margin of safety to ensure survival during sudden gusts or squalls and storm seas.

Sailing vessel design has clearly evolved over the years; the characteristics of traditional design were based on satisfactory experience and sailing performance. By changing parameters gradually, the designers of the past remained confident of the stability and seaworthiness of the tall ship. However, modern ships are often made of different materials, have different configurations, large accommodation facilities, and most have powerful auxiliary motors. Even sailing vessels which have been "restored" are often fitted with a very different rig and employed in different services from what was originally intended. To varying degrees the limitations of experience in building and operating a large sailing vessel have been surpassed, so that an appropriate level of safety can no longer be assured without the use of extensive naval architecture and engineering calculations, the majority of which can easily be done by electronic computer.

In addition to modern design changes, the tall ship is different from other sail ships. The words "tall ship", when used throughout this paper, include large sailing ships of nearly all rig designs of square sails and stay sails (i.e. Schooner, Brig, Brigantine, Barque, etc.). Ships with square rigged sails in particular need additional crew to set and douse the sails, and many persons must go aloft for the final stowing of the sails. The weather deck must be arranged to provide more working space for sail handling. And most important to the subject of this paper, the stability and seakeeping characteristics of a tall ship are different from those of the smaller sailing vessels. Also, Deakin[2] reports results of research which show that a sail ship under full sail is essentially fully damped so that the wind gust acts as a slowly changing force instead of an impact.

In addition to treating the tall ship differently from a stability point of view, there are further differences to consider since most are involved in training. Tall ships carrying passengers and cargo had traditionally been sailed solely by professional crews who served their apprenticeships at sea and had an intimate knowledge of both the working of the vessel and its limitations. Now most tall ships are sailed largely by trainees supported by a small crew of professional sail instructors. The trainee is functionally different from a passenger (who has paid for a leisurely and safe voyage) because the trainee is willing to work the sails and soon becomes familiar with the safety equipment and rigging. The tall ship involved in training should offer sailing experience in both light air and heavy weather and therefore must be versatile in its operations.

Although the instructors of the trainees may have a fair number of years of experience in sailing they may not be intimately familiar with a particular hull and its sailing rig and most importantly with its overall stability limitations. In order to facilitate the learning of the limitations of a sail vessel it is necessary to provide the Captain and his Sailmaster with a knowledge of the stability of the ship in relation to the amount of sail carried. Because of the need to be versatile it is not sufficient to only provide a maximum allowable sail plan for a tall ship. A maximum allowable sail plan is often used for moderate weather conditions and therefore can be misleading when the weather deteriorates to storm conditions. Providing a comparative chart or set of instructions to be used, based on weather conditions and individual ship characteristics, should be the outcome of a functional standard.

Existing USCG Criteria

The stability standards which have governed sail designs have varied greatly. It is generally accepted that the criteria applied by the U.S. Coast Guard (USCG) is the one of the most comprehensively researched and widely used. The basis of the USCG criteria was a statistical survey by Brooks and Beebe-Center[3] of over 50 ships, in which stability characteristics were calculated and compared. The data was expanded by Long and Marean[4,5] and the standard was modified in 1986 to cover sail training vessels. Minimum acceptable values were set to cover three events of significance. These events reflect the following actual physical sequence which occurs as a sail vessel is rolled through its full range of stability: deck edge immersion, immersion of the downflood point, and knockdown causing capsize. These events are recognizable by both the naval architect and the seaman and are important to both the functionality and the safety of the sailing ship. The USCG standard has formed the basis of the requirements adopted in Australia and those recently proposed in Canada. There are other countries which have used various standards when it was necessary for them to evaluate a ship but the information that we have on the others is not comprehensive and therefore was not used and will not be discussed here.

Criticisms of the USCG criterion have been voiced throughout the years, mostly about the assumptions inherent in the method, but also excessive stability numerals, causing a stiff ship with insufficient sail for light weather. Most of the criticisms of the assumptions inherent in the USCG method have little merit because they are widely used in naval architecture to simplify the hydrostatic model for ease of calculations, and also to simplify the aerodynamic model of the sails. Also, the stability numerals of the USCG criteria are based on comparative statistics, therefore, most criticism of the method are moot. However, there are some criticisms which do have merit and if the criteria were improved to remove each criticism a more useful and functional standard would indeed come forth. The criticisms of significant merit are:

1. The wind is assumed to have a uniform velocity at all elevations.
2. The stability numerals are not in correct units of wind pressure.
3. All types of sails have a force coefficient equal to 1.0.
4. The stability numeral to downflood evaluates the response to a wind gust (increase in wind speed) using an energy balance method, as if it were an instantaneous impact upon the sails, which does not adequately account for the vessel's inertia and sail damping effect.
5. The heeling moment caused by the wind varies with the heel angle as a function of the cosine squared of the heel angle.
6. Overlapping sails are neglected.
7. The maximum heeling moment occurs when the wind is directly on the beam.
8. The standard requires a single sail plan which is too little for light air operation and unsafe in heavy weather.
9. The criteria does not lend itself easily to providing additional knowledge about the sail ship to the master.

Although the USCG criteria has faults, it has been applied successfully to certificate many small sail ships, all of them having safe operations in regards to stability. However, for those which have been turned away by the USCG (not given a certificate) because they were not able to satisfy the criteria, there has been a very noticeable succession of casualties. The *PRIDE OF BALTIMORE*, the most recent capsizing casualty[6], was one of them. It must be noted however, that successful and safe operations are not due solely to the ability to carry a specified amount of sail. The Sail-master must still know the limits of his vessel and be able to control the reduction of sail as the wind increases or the likelihood of severe gusts and squalls exist.

New UK DoT Criteria

Following the loss of the *MARQUES*, the United Kingdom's Department of Transport (UK DoT) commissioned a research program at the Wolfson Unit of the University of Southampton[7] to perform a technical study and to do scientific research on sail ship stability. The study was comprehensive in its scope. It involved scientific research and the gathering of large amounts of information on sail ships both in the UK and internationally. Numerous sail ship designers, owners and operators were interviewed and many sail ships, varying in length from 12 to 45 meters, were evaluated for their stability characteristics. During the scientific research, model tests were performed for both the hydrodynamic and aerodynamic characteristics of various hull forms and sail rigging. During this research many theories were verified and recommendations for improvements were generated.

As an outgrowth of the work performed by the Wolfson Unit a new standard for sail training ships was developed which has recently been adopted by the UK DoT[8]. It is the opinion of the authors that this new criteria is still not comprehensive enough to be used internationally as a criteria for the stability of large sail training ships. Some of the notable shortcomings of the UK standard are:

1. Regulates only the shape of the righting arm curve and not the amount of the actual heeling arms caused by the wind.
2. No incentive to increase the downflood angle to an angle greater than 60 degrees.
3. The Derived Wind Heeling Lever is solely based on the assumption that gusts will not exceed 140% of the mean apparent wind speed (or twice the pressure) and does not provide any limit to the amount of sail carried.
4. Regulates the angle of heel at which a vessel is sailed in steady winds. Although it is recognized that there exists some safe maximum angle, this aspect was reported by the Wolfson Unit[5] as an item which sail masters felt should not be regulated.

Only a few of the shortcomings of the USCG criteria were improved upon by the UK criteria. In fact, the UK criteria appears to ignore many of the important findings of the Wolfson Unit by not evaluating a sail vessel by using the amount of sail it carries, but puts the onus completely on the master to be knowledgeable enough to sail his vessel safely under all conditions while guided by potentially misleading data. We say potentially misleading because the assumption of using twice the pressure of the steady wind has been reported by Deakin[2] to be grossly underestimated when squall conditions exist. It also assumes the vessel is heeling to its maximum amount when determing the amount of sail that may be carried, which is seldom the case. Deakin confirms this clearly in his statement "The maximum possible heeling moment does not therefore correspond to a normal sailing condition."

We believe the work performed by the Wolfson Unit was excellent and that there are many things to be learned from their research. Wolfson's contributions, coupled with the comprehensive research and the years of successful experience of the USCG standard, could dissolve most of the shortcomings of the USCG and UK DoT sail ship standards. A stability standard can be created which would be appropriate for all large sail vessels.

Creating a Hybrid Criteria

The merits and limitations of two independently derived criteria for sailing vessel stability have been discussed. It is central to the purpose of this paper to observe that neither body of regulations were formulated with focus on the large sailing vessel. It is possible, and in fact probable, that the intact stability which is required for safe operations of sail vessels is influenced by factors related to scale. However, scale factors are not fully known at this time so we do not propose an extrapolation of statistics from either source.

We have formulated a proposed standard, attached in the annex. Since this is still preliminary, the values for wind speed and sail coefficients must be considered to be in "square brackets".

We believe this comprehensive standard will be most appropriate for designing, operating and regulating tall ships for the following reasons:

1. There is a method provided for evaluating reduced sail plans for moderate and heavy weather. This allows tall ships to carry enough sail to be maneuverable in light air, which alleviates the difficulty of sailing in a strong current close to shoal water with a light breeze blowing ashore. As in the UK, the sail master should be provided guidance to account for the likelihood of gusts and squalls.
2. Since the measure of stability is a static balance of forces up to downflood or sixty degrees whichever is greater, there is incentive to locate doors and hatches further inboard.
3. The range of stability must be at least 90 degrees, which requires a generous freeboard and virtually eliminates broad shallow hulls that are prone to capsize in heavy weather.
4. The formula to calculate total area of the sails accounts for a wind gradient based on the height

of each sail above the water (See Table 2).

5. The wind heeling arm curve is based on the more realistic function measured at the Wolfson Unit ($\cos^{1.3}\theta$), and uses correct units for wind pressure.

6. The type and shape of each sail is taken into account and an apparent wind angle of 90 degrees for square sails and 60 degrees for fore and aft sails is applied in using Table 1.

7. By using a balance of righting and heeling moments for the Fair and Moderate weather sail plans, sail damping effect is included.

Proposed Criteria for Tall Ships

The proposed criteria rely heavily on naval architecture design calculations. A tall ship must be capable of meeting each portion of the criteria for a range of loading conditions from the full load departure condition to the minimum operating condition, which is usually a 10% consumables arrival condition. The righting arms are calculated taking into account any buoyancy which may be gained from weathertight enclosures above weather deck. The heeling arms are calculated using actual wind induced pressures and the total area of sails which takes into account their type, shape and height above the water.

The proposed criteria is flexible enough to allow the designer to provide enough sail area so the sail master will be able to make headway in light weather, (10 knots with gusts up to 15 knots), without submerging the deck edge or 20 degrees, whichever is less. It also ensures that the most vulnerable sail plan must still be able to safely prevent heeling beyond 60 degrees during a 30 knot squall, which is a typical magnitude experienced during light weather.

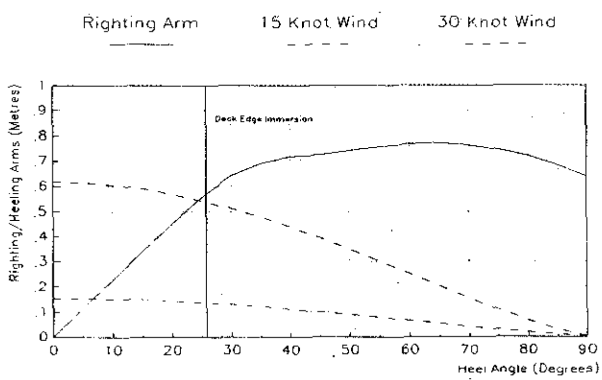
It is believed that as the weather becomes moderate the sail master will have more than ample wind to control the ship so his sail plan options (reduced sail plan) are flexible. However, in order to ensure safe sailing conditions the maximum amount of sail area should be at the Sail master's discretion but not allow heel angles greater than 30 degrees in moderate weather, (15 knots with gusts up to 30 knots), or downflood during a 45 knot squall.

During heavy weather the sail master must reduce his sail significantly in order to maintain a safe vessel. Breaking waves will often accompany heavy weather conditions so this part of the criteria uses a balance of the energy under the righting and heeling arms.

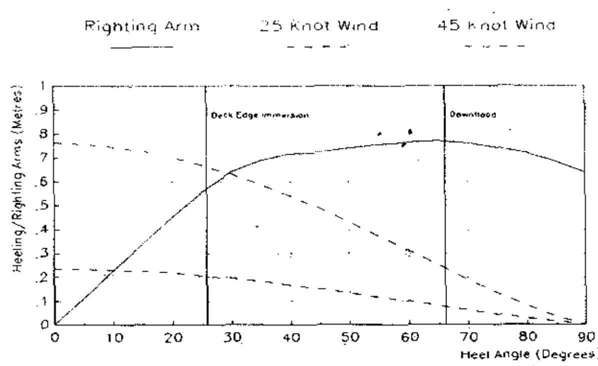
USCG Barque EAGLE

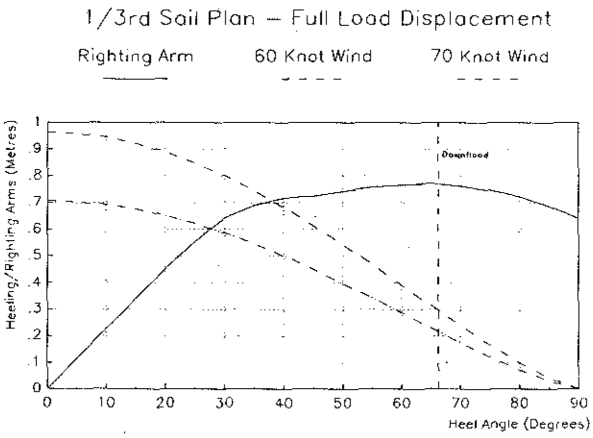
The proposed criteria were applied to the U.S. Coast Guard Training Barque *EAGLE*, which has been reported by Tsai and Haciski[9] to be a "good" design. She was built in 1936 as the *HORST WESSEL* by Blohm & Voss in Germany. There have been numerous articles written about the *EAGLE* detailing how the ship had survived severe squalls and storms. The graphs below show that the *EAGLE* exceeds the proposed requirements in all cases. Several of the calculated wind heeling arm and righting arm curves correlate closely to observed heel angles for corresponding actual conditions reported by Tsai and Haciski.

Maximum Sail Plan – Full Load Displacement



2/3rds Sail Plan – Full Load Displacement





Conclusions

The proposed criteria contains elements of both the current USCG criteria and the research underlying the UK DoT criteria. The proposed criteria is based entirely upon real situations which a master will encounter and is organized in such a way that the designer can provide the master with specific guidance as to the amount of sail carried versus the wind speed. Credit is given for having high downflood angles, the sail type and height are accounted for in calculating the effective wind area and the vessel's response to gusts is taken into consideration. A major premise underlying this criteria is that the designer will provide the master with the sail plans and the wind speeds considered in meeting these criteria for his guidance. Although the designer is encouraged to provide safe limits of heel angles on a wide variety of sails plans there is a minimum of three which cover the range from light to heavy weather. Since the criteria is based upon wind speeds and the vessel's sail plan, further adjustments may be made for a particular vessel which either can not satisfy a portion of the criteria or has a widely varying sail plan. We encourage naval architects to make a critical review of this proposed criteria, to apply it to designs which they may have available to them, and to share their experience.

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Annex

Proposed Criteria for Tall Ships

Naval Architecture design calculations shall be used to show that each oceangoing sail vessel ($L_w > 24$m and Gross Tons > 500) is capable of meeting the following criteria under its range of loading conditions including a full load departure condition and a partial load (10%) arrival condition.

For each loading condition the GZ curve must be positive from 0 to at least 90 degrees, and the downflooding angle must be shown to be greater than 60 degrees using the aggregate area method. (i.e. Angle of downflood is the angle to the lower edge of a critical opening which if submerged would result in an aggregate area in square meters greater than the vessels displacement (tonnes) divided by 1500.)

With all sail furled the vessel must be shown to satisfy the criteria of the IMO Resolution A.167(ES.IV).

While sailing under the following limiting conditions and using the heeling arm curves for each wind speed as defined below, the following criteria must be satisfied.

1. Light Weather Gust Wind - Full Sail Plan

The intersection of the vessel's GZ curve for each loading condition and the wind heeling arm curve for a 15 knot gust wind with the sails set for the most vulnerable sail plan (as much sail area as can be set at any one time), must occur at a heel angle less than 20 degrees or the angle of deck edge immersion whichever is less.

2. Light Weather Squall Wind - Full Sail Plan

The intersection of the vessel's GZ curve for each loading condition and the wind heeling arm curve for a 30 knot wind, using the sail plan from 1. above, must occur at a heel angle less than 45 degrees or an appropriate lesser angle to permit the continued efficient handling of the sails.

3. Moderate Weather Gust Wind - Reduced Sail Plan

The intersection of the vessel's GZ curve for each loading condition and the wind heeling arm curve for a 25 knot wind, using a sail plan with no less than 2/3rds of the sails set, must be less than 30 degrees or the angle of deck edge immersion whichever is less.

4. Moderate Weather Squall Wind - Reduced Sail Plan

The intersection of the vessels GZ curve for each loading condition and the wind heeling arm curve for a 45 knot wind, using the sail plan from 3. above, must be less than 75% of the angle of downflooding to account for uncertainty in the damping effect as sails are reduced.

5. Heavy Weather - Storm Sail Plan - Downflood points not closed

The area under the vessel's GZ curve up to the downflood point for each loading condition must be greater than the area under the wind heeling arm curve for a 60 knot wind, using a sail plan with no less than 1/3rd of the sails set.

6. Heavy Weather - Storm Sail Plan - Downflood points closed.

The area under the vessel's GZ curve up to 90 degrees for each loading condition must be greater than the area under the wind heeling arm curve for a 70 knot wind, using a sail plan with no less than 1/3rd of the sails set.

Other aspects which must be considered by the sail master are:

1. The influence of crew ability, strength and comfort level.
2. Allowance for visibility conditions, day or night, etc.
3. Effect of gear failure on rigging designed for automation or ease of handling.
4. Considerations for carrying passengers instead of trainees.
5. When racing in company with other tall ships the primary mission remains training, education and safety; winning the race is of secondary importance.

The wind heeling moment curve shall be calculated as follows:

$$A_T = \Sigma C_r * C_h * A$$

$$H_A = \frac{K * A_T * L * V^2 * (\cos^{1.3}\theta)}{1000 * \text{Displ.}}$$

Where:

H_A = the wind heeling arm in meters.

K = 0.017, for metric units

C_r = coefficient of sail efficiency - Table 1

C_h = coefficient of wind gradient - Table 2

V = Wind velocity in knots, as specified for each criteria.

A = Area of each sail in square meters.

A_T = The sum of the area of all sails adjusted for C_r and C_h.

L = Distance from the Center of Effort of the sail plan to ½ the draft in meters.

Displ. = Displacement in tonnes

Table 1

Sail Type	Cr
Fore/Aft Sail (not overlapped)	1.5
Fore/Aft Sail (>30% overlapped)	1.2
Square Sail (which cannot be braced around more than 50° from athwartships)	0.8
Square Sail (which can be braced around more than 50° from athwartships)	1.0

Table 2

Height of Sail ϕ Above the Water in meters.	Ch
Under 7.6	0.67
7.6 - 9.1	0.74
9.1 - 10.7	0.81
10.7 - 12.2	0.88
12.2 - 13.7	0.94
13.7 - 15.2	1.00
15.2 - 16.7	1.06
16.7 - 18.3	1.12
18.3 - 19.8	1.17
19.8 - 21.3	1.22
21.3 - 22.9	1.27
22.9 - 24.4	1.32
24.4 - 25.9	1.36
25.9 - 27.4	1.40
27.4 - 28.9	1.44
28.9 - 30.5	1.47
Over 30.5	1.50

STABILITY CRITERIA FOR CONTAINER SHIPS

GÜNTER HELAS

ABSTRACT

The purpose of this paper is to report on the work of a classification society with regard to the assessment of the stability of container ships. Sufficient stability is considered as a precondition for classification. Special attention is paid to container ships, the stability of which is often near the permissible minimum values. The available international stability criteria are discussed, and the need for additional national criteria is stated. The results of some test calculations in form of GM limiting curves are presented. It is shown that further research work is necessary with regard to the stability of container ships in following waves and the determination of the mass and the centre of gravity of containers. New problems for classification societies arise also from the construction of container ships without hatchway covers.

STABILITY - A CONDITION FOR CLASS

Stability is one of the most important factors to be taken into account when designing and when operating a ship. The safety of the ship depends to a high degree on sufficient stability. In the past and also at present many research works, reports and papers were and are dealing with the question what sufficient stability is. An answer to this question which is satisfactory in all respects was not yet found. But supervision authorities and classification societies can not wait for a comprehensive scientific solution, they have to evaluate the stability of ships at present. They do that by means of stability criteria established by the International Maritime Organization (IMO), by governments or by themselves. DSRK has always considered sufficient stability of the ship as a precondition for classification. This is now also the opinion of all classification societies which are members or associated members of the International Association of Classification Societies (IACS). In 1988 IACS has published its Unified Requirement "Intact stability - matter of class". Therefore, the evaluation of stability by a classification society is not only a task carried out on behalf of an administration, it is also its own task, it is a condition for class.

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the maximum number of empty containers.
- The minimum metacentric height is the decisive criterion for loading conditions with medium values of displacement.

- The criteria for the shape of the righting arm curve are the decisive criteria for loading conditions with high displacement. The area under the curve up to 30° is the most important one of these criteria.

IMO has not yet finished its work on intact stability criteria. Therefore, the application of the existing criteria is not considered as fully satisfactory. We think that some additional criteria, in particular for container ships, are necessary. In the opinion of DSRK the further development should include the investigation of the following items:

- metacentric height;
- vanishing stability;
- stability in following waves.

The recommended minimum metacentric height of 0.15 m seems to be very low. We have to take into consideration that a stability calculation after loading a container ship includes many uncertainties caused by wrong or unknown data for the mass of the containers and the height of the centre of gravity of the individual containers. A calculated metacentric height of 0.15 m can be in reality near to 0. Masters have told us that they always try to avoid, whenever possible, to operate the ship with a metacentric height less than 0.50 m.
The IMO criteria allow an angle of flooding of 40° or even less. DSRK is of the opinion that a stricter criterion is necessary.
It is well known that the stability of a ship in following waves, on the crest of a wave, is reduced substantially. This refers especially to container ships which have a stability in smooth water near to the required minimum values.

STABILITY RULES OF DSRK
Like the IMO recommendations, the Stability Rules of DSRK [3] contain criteria for the metacentric height and the shape of the righting lever curve and also a weather criterion. The required metacentric height is 0.20 m.

The criteria for the shape of the righting lever curve and the weather criterion are about equivalent to those of IMO. But there is one exception. DSRK requires an angle of vanishing stability of 60°. Under certain circumstances, this value may be reduced down to 50°, but this is the absolute minimum. We do not permit an angle of flooding, which is to consider as the angle of vanishing stability, below 40° as IMO does. This requirement has a great effect as shown in Figures 1 and 2, where it is represented by curves 4. When the ship operates at her full draught or near to that, the angle of vanishing stability is always the decisive criterion and requires a greater metacentric height. This means a reduction of the container carrying capacity.

The Rules also contain a recommended method for calculation and evaluation of stability in following waves. This method is deseribed in [4] . It was applied to the same two container ships for which the GM limiting curves are given in Figures 1 and 2. The righting lever curves in smooth water and on the wave crest are shown in Figures 3 and 4. The wave length was taken equal to the length of the ship. The wave crest

SPECIAL ROLE OF CONTAINER SHIPS

When evaluating the stability of ships and working out stability regulations, container ships play a special role. The designers of container ships have the possibility to increase the container carrying capacity up to the uppermost limit fixid by the stability critêria, so that no additional reserve stability remains. Because the stability regulations are utilized up to the limits and the permissible minimum values are reached, container ships are endangered with respect to stability to an increased degree. Therefore, it is of special importance to lay down the appropriate level of safety, so that the minimum values are not too low and not to high, in order to avoid stability casualties on the one hand and to restrict the container carrying capacity on the other hand only so far as necessary. There is a danger that the international economic competition leads to the application of unsufficient stability criteria and to unsafe ships. We had in our practice container ships of the same series which were allowed to carry different numbers of containers depending on the regulations applied. Internationally agreed stability criteria are in our opinion the only one solution for that problem.

STABILITY RECOMMENDATIONS OF IMO

Already for a long time IMO is working on intact stability criteria. Some criteria have been developed which have found wide application all over the world. But there are no special criteria for container ships. We have only the usual criteria for all cargo ships which refer to the metacentric height and the shape of the righting lever curve [1] and the weather criterion [2] .

DSRK has carried out some test calculations for existing container ships to show the effect of the IMO criteria. The results are given in the form of GM limiting curves. The lowest value of GM which ensures the fulfillment of the criteria is plotted against the displacement of the ship. Figures 1 and 2 show two examples for ships of following principal dimensions:

Fig. 1 L = 152,40 m
B = 23,05 m
D = 13.40 m
4 tiers of deck containers

Fig. 2 L = 163.37 m
B = 25.40 m
D = 15.90 m
4 tiers of deck containers

The curves represent the following criteria:

Curve 1 - recommended minimum metacentric height.

Curve 2 - criteria for the shape of the righting lever curve (area under the curve up to 30° and 40° and between 30° and 40°, minimum righting lever, position of the maximum of the curve).

Curve 3 - weather criterion.

Curve 4 - angle of vanishing stability required in the DSRK Rules [3]

The following general conclusions can be drawn from the results:

- The weather criterion is the decisive criterion for loading conditions with low displacement, e. g. when the ship is loaded with was assumed to be at midlength. The wave height was obtained from the formula given in [4] and was 4.60 m for ship No. 1 and 4.65 m for ship No. 2. Ship No. 1 is in the loading condition with the maximum number of containers of 20 t and 100 % stores, ship No. 2 in the loading condition with the maximum number of 20 ft containers. In both cases the stability in smooth water is fully sufficient, all criteria are complied with. There is even a certain additional reserve stability. However, the righting lever curves of the ships on the wave crest are very bad. Ship No. 1 has a maximum righting lever of only 0.05 m, ship No. 2 has even a negative metacentric height. Both cases show the danger in which the ships in the assumed seaway are.

STABILITY CONTROL ON BOARD

Stability control on board is very important for container ships. DSRK requires that the ships are equipped with heeling tanks or other approved facilities, by means of which the initial metacentric height of the ship can be checked. Also an on-board computer should be provided for recording the mass and position of all containers. In this connection DSRK has been faced with two problems which need an international exchange of information and experiences. The first problem is the mass of the container which is often higher than stated or allowed. The second problem is the height of the centre of gravity of the containers. Assumptions which are to be taken according to different regulations vary between 0.4 and 0.5 of the height of the containers and have a substantial influence on the theoretical container carrying capacity. A solution for this question could be to carry out inclining tests after loading as much as possible. It would then be possible to calculate a mean value for all containers together. At present DSRK requires to assume the centre of gravity of a container at 0.5 of its height. It might be that this value is too high, but in this case it would be a certain compensation for too low values of container masses included in the stability control.

OPEN-TOP CONTAINER SHIPS

Open-top container ships are a new type of ship which is now under construction in several countries. These are ships with high side erections and without hatchway covers. Figure 5 shows the cross-section of such a ship. It is a design of a shipyard of the German Democratic Republic.

Administrations and classification societies have to decide how to ensure the stability and how to assign the freeboard for such ships where the weathertight closure of the deck as required by the International Load Line Convention is missing. Obviously it is not permitted that the side of the deck immerses up to a certain angle of heel which corresponds with the required range of stability. With regard to the freeboard, DSRK has the intention to use the possibility of equivalents provided in the Convention. It should be shown by model tests that the same effect is obtained by the high sides as by weathertight hatchway covers , namely that the holds remain dry in a seaway. It is now necessary to develop the test conditions for such model tests. This should be also a matter of international co-operation.

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FIGURES

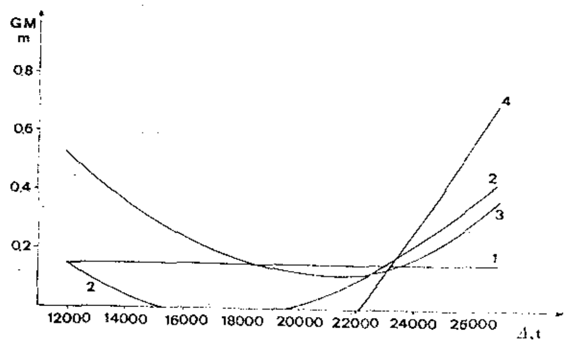


Fig. 1

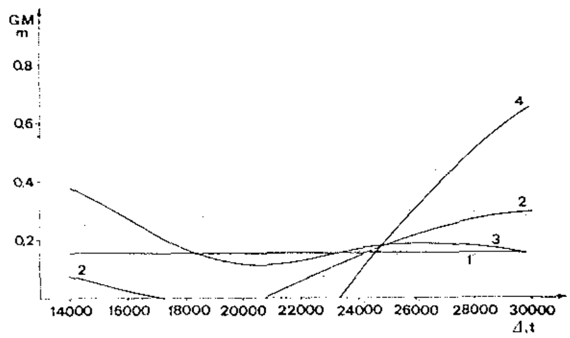


Fig. 2

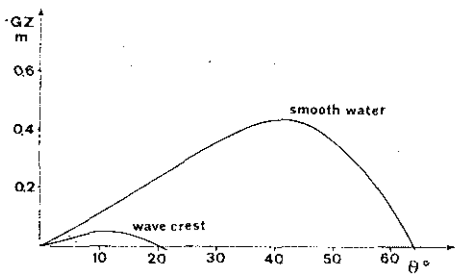


Fig. 3

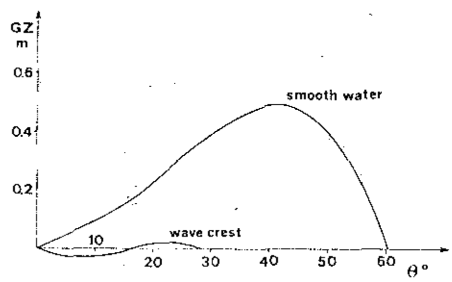
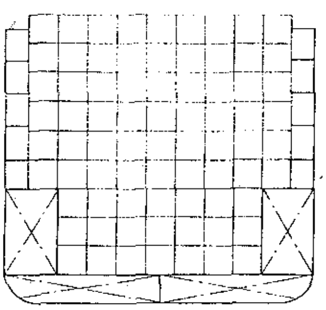


Fig. 4



L = 143.20 m B = 25.60 m
D = 24.10 m d = 10.00 m

Fig. 5

STABILITY OF DRY CARGO SHIPS
STATE OF THE ART, INTACT AND DAMAGE REQUIREMENTS, IMPLEMENTATION IN PRACTICE

Dipl.-Ing. Hartmut Hormann

Abstract

Dry cargo ships are since long internationally subjected to intact stability criteria for which a majority thinks, they provide a sufficient level of safety. Upon scientific considerations based mainly on systematic tests doubts have been casted whether this was generally valid, and proposals are floating around to increase the required minimum values for certain ship categories.

The paper discusses the practical problems, which stand against the application of such higher values, and relates this to the down-to-earth conflict between economy and safety a Master finds himself in.

The IMO decision to require survival capability after damages also for dry cargo ships has changed the situation.

Effects of the survival requirements and possibilities to match them are highlighted, as well as the difficulties to achieve uniform application. Furthermore the question is asked, whether the level of survival capability as represented by the R-formula chosen is adequate.

In linking the areas of intact and damage stability the difference in basic philosophy is discussed : intact stability relates to an every day situation, while matching intact requirements resulting from damage calculations means to cater for a situation, which with high probability will never actually occur.

Introduction

Since a good number of years the question "what is sufficient intact stability" is answered by most of the people in the marine industry by : "what the statutory requirements say". This is fine, but does not hold entirely. The reasoning for this statement is in brief :

- statutory instruments (international or national ones) do not cover all types and sizes of ships
- soon the vast majority of ships will be subjected to survival capability requirements, which might override intact criteria.

The subject of the paper is to reflect on the present scene in the area of the requirements and to discuss their basis and implementation in practice.

Intact criteria - comparing new and old designs

For reasons of clarity, I will refer in this part to the IMO A.167 - criteria, when the existing pattern of requirements is mentioned. I do, of course, not underestimate the value of other internationally agreed sets like those for offshore supply vessels, but they do not represent a new approach compared with A.167, but follow more or less the same pattern.

It is a sufficiently known fact that the types and characteristics of the ships have changed significantly since the present criteria were conceived and agreed. The naval architect in exceeding existing and proven parameters in a design has a genuine duty to ask, whether the level of safety of the new design is sufficiently high, i.e. at least as high as what was acceptable for existing ships. - One of the ways to get a reply to this question is *systematic tank tests; and I am convinced*, despite of all advancement and sophistication of calculation methods, such tests are still the most suitable means. Now, what is the conclusion, if tests in an extreme, but still possible, irregular seaway performed with a model featuring all typical characteristics of a modern containership lead to series of capzisings in following seas while a model of conventional hull type does not capsize at all?

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Quite naturally this will cast doubts into the stability level represented in the model of the new type. The process of comparison of the new hull design with conventional ones is next. In our case it reveals that modern containerships are bigger as the generation of ships for which existing criteria were found acceptable and - most important - their beam-over-depth-ratio (B/D) is much greater. Fig. 1 gives an idea of the differences as evident in the righting lever curves; both curves meet the existing criteria, the alarming difference is represented by the greatly reduced range of stability and - simultaneously - the modern ship for smaller angles of inclination has a characteristic of a traditional full scantling vessel, despite of the fact, that its freeboard-to-draft-ratio (f/d) is of the order of those found in earlier years for so-called shelter deck vessels. - This all is not new, and readers will recall that several colleagues of mine together with me since almost 10 years point to these facts. I suggest that the actions proposed for coping with the problem are well received by those responsible for setting standards :

Means for keeping the level of safety

I will not go into details of the German proposal to introduce a "hull-form-factor" (1), this has been sufficiently publicized, I will in this wider context point to the principles only and for this just quote two colleagues who are entirely independent from the German research project (2) : "The concept using a form factor correction has promise, provided it can be verified against a wide range of parametrically varied ships." The same publication explains very well the concept : "The Federal Republic of Germany's philosophy is that IMO A.167 represents a satisfactory criterion for the hull forms for which it was developed; that is, for shorter vessels (L < 100 m) and conventional hull forms. The form factor is intended to maintain the existing level of safety for these vessels, while providing a more stringent criterion to reach the same level for vessels with hull forms more susceptible to loss of stability in following waves." Here I would just stress : the "more stringent criterion" is necessary to maintain the same level of safety, not an increased one !

Can the advanced criterion be made applicable?

Under the auspices of IMO the international marine community undertakes to work on the improvement of existing intact stability criteria and on the completion of the range of ships covered by agreed criteria. This is extremely difficult and even if a sound solution is developed for certain groups of ships (see below), introducing this for compulsory application is hardly possible; why ? The answer is manifold, the main facets are :

- the subject of intact stability is not accessible for calculating unequivocal and undisputed limiting values; contrary to e.g. the load bearing capability of a steel structure, where you have the physical properties, the basic physical principles, which can be converted into exact formulae and the safety factors against e.g. the breaking strength; and most important : the parameters to be taken into account and having measurable influence are limited in number. Not so in case of the system : buoyant hull of a complex form in an irregular environment causing heeling and uprighting forces extremely varying with the time.
- From this first facet it follows suit that no one is able to directly deduct in figures how close a chosen intact stability level is to the minimum necessary for survival.
- Any method of proving the adequacy of a value chosen fails, because the margin cannot be determined, if a ship survives in a certain environment. And the parameters contributing to the survival (or more positive : to the safe operation) of the ship include human action, i.e. the skill of the nautical personnel to choose the course and to avoid dangerous situations in adverse weather conditions.

The common headline for those three facets is : uncertainty; and because of this the next facets of the whole scope depicting the difficulties arise :

- in particular for containerships, applying a more stringent intact stability requirement means directly loosing container carrying capacity in almost every voyage.
- more stringent requirements will only be universally accepted, if the necessity can be proven. And this cannot be done - see above ! Of course, fatal accidents would change the situation, but nobody will advocate this solution. And if there is a capsize, further difficulties accrue: deducting limit values of stability from an accident requires exact information on what happened and in what condition the ship was; furthermore so-called stability accidents very often are caused by shift of cargo or partial loss of buoyancy, this is not accounted for in our criteria.
- the main argument for not accepting higher requirements is : nothing has happened. However, this is not a proof that the disposition of values was sufficient - not even for a single ship having plied the oceans for many years.

Which level is necessary ?

What I said in the preceding paragraphs implies that the advanced criterion means higher requirements, i.e. less economical service. This is true, the respective trend was evident in the extensive test series performed (3). Comparing the situation now with the one prevailing at the time A.167 was introduced reveals as principle difference : the A.167 level matched with what was already applied for a majority of ships before since long on a more or less empirical basis. Now, however, the A.167 criteria have been applied to ships of types and sizes they were not made for. The statement that those bigger, modern ships have less reserve in stability as traditional smaller ones, which meet A.167, cannot be disputed. This matches with the trend mentioned earlier, as found in the tank tests.

I would emphasize the term trend in this context. The actual values proposed for implementing the concept of the hull-form-factor can by all means be discussed. They were established with all available diligence by first making qualitative deductions, which form parameters had major influence. The figures eventually chosen for the empirical correlation of the major form parameters were derived by relating them to a relatively large but nevertheless limited number of ships in service on the basis of subjective judgement of their stability quality. So, there is room for improvement, and I would solicit efforts in this area. Despite of the difficulties to introduce a requirement curtailing the economy, we should strive for it. I am convinced, the safety margin (regarding intact stability) of the great group of dry cargo ships, multipurpose or containers, is lesser than say 20 years ago. We must not wait for the big "convincing" accident. In order not to be misinterpreted : I do not say, they are unsafe, just less safe than before - but nobody knows, how close we are to the limit.

Before I move to the area of survival capability after damage - the area, which I hope will be the sword to cut the Gordian Knot - I would spend a few thoughts on operational aspects :

Challenge and clash of interest

Every ship, nowadays, has stability information on board, which give limitations for loading the ship, directly or indirectly. The Master is obliged to observe them. That sounds easy, but it is not.

The accuracy with which the necessary data can be assessed is limited and varies, depending on the case and on the means applied. This implies a challenge for the Master to let either safety or economy prevail. (It must be understood that here the Master is meant who obeys under the rules, not the one who knowingly surpasses the limits set for whatever reasons, e.g. good weather forecast, economical pressure, etc.). The Master in the process of establishing whether his ship is in a condition safe to proceed to sea has factual data, he has his decision pattern, and he has parameters influencing his decisions to a varying extent.

The factual data are

light ship data

cargo weight and centre of gravity

consumables, including free surfaces

ballast.

His decision pattern may be either conservative, i.e. tending to make assumptions on the safe side, or tend to the riskier side. It would be normal that a Master combines decisions on the various aspects in a way to arrive at a good average. - He will be influenced in his spate of decisions by parameters like :

- reliability of sources for cargo data
- degree of repitition of voyages of same or similar pattern
- weather expectations
- quality of lashing and securing.

These parameters come to bear when he decides to go either to the conservative or to the risky side. - The vast majority of stability assessments for individual voyages is established this way.

There is, however, the possibility to do away with a good part of the uncertainties : the inclining experiment in service. A number of authors have dealt with this subject (e.g. (4)). Modern container ships need very little extra equipment to be able to perform such test, the evaluation is comparatively simple and quick, the results reliable to an acceptable degree (4). Such a test can only be advocated. The problem is : it can reasonably be performed only after finalization of loading or close to it, and what happens, if the results do not compare favourably with the requirements ? Here is the clash between safety and economy ! The Master has the choice to unload, perhaps re-stow cargo - a very unpopular option - or to proceed to sea hoping that worst will not come to worst.

The naval architect cannot help much in this situation, except in a general way. If we succeed to establish, where the limits between safe and unsafe really are in terms of intact stability requirements, the Master would have solid ground for his decision. With our present knowledge he can still say : I may not quite comply with the required values, but they must embrace a safety margin, which I now make use of.

Damage survival requirements - brief history

Those, who did not follow the respective development closely, will find it not very evident, why the great group of dry cargo ships is not subjected to any survival capability requirements after damage, while may other types of ships - by far not only passenger ships - have to comply with relevant regulations. Damage requirements had their start in passenger ships. It is, of course, hard to explain, why passengers deserve better care for their lifes than professional seafaring people. But there are - as I see it - two main reasons for the original attitude : firstly, also in the early decades of this century desasters like the sinking of a passenger ship triggered public emotions, which required "balancing" actions, i.e. special requirements so that the safety standard of such ships could be proved as being higher than that of ordinary ships; and secondly, without computers the damage calculations required a volume of work, which made it obvious that one could not afford this for every ship. - Our present-day-thinking, of course, categorically suggests that not the question whether there were passengers on board or not should be decisive, but the number of persons, might they be passengers or crew. - It deserves attention that almost all additional damage requirements, which were introduced far later, were not reasoned by the care for the people, but mainly by environment protection considerations and - as a curiosity - in one case for compensation for a reduced intact freeboard !

In advancing the safety requirements in all areas and for many types of ships, in addition to the above, another consideration led in some instances to requirements for survival after damage : if it was comparatively easy to achieve certain survival capabilities for a group of ships, this was introduced. As an example the offshore supply vessels can be mentioned. - This brings us to the genuine reason, why for so long the international recommendation set up during the SOLAS 60 Conference for such a long time was not implemented : the big group of dry cargo ships, since more than 25 years comprises a lot of special designs, which would not be possible, if a compartmentation for survival capability was required using the traditional approach, which is almost exclusively based on transverse bulkheads. The break-through for sound solutions was the introduction of the viable probability concept. This was proposed already in the late fifties, but one must appreciate that considerable time is required for developing the mathematical concept into the form of regulations applicable in practice. I do not think that the twenty-odd years were wasted; on the contrary, the time pressure exerted on the development in IMO for the last two or three years did not only have positive effects. The result would have become much better, if things in the final stage were not done in a hurry.

At this time I wish to bring in another thought, which should facilitate realizing why it is about timely now to introduce requirements for survival after damage (Fig. 2). If one goes back about 5 to 6 decades, one will find under the term dry cargo ship almost exclusively what we would today call break bulk ships, with rather small hatches, often several tweendecks and - above all - very regularly positioned transverse bulkheads, normally every 20 metres. Such design, together with its high L/B-ratio, lead to a rather high survival capability after damage - without any relevant express requirement. Furthermore this survival quality varied very little from ship to ship. Experience during World War II lead at that time, and still for some years later, to slightly higher quality in this respect, while the general features were maintained. With the early sixties the development towards specialized dry cargo ships ensued and soon produced types with very poor survival qualities, e.g. Ro-Ro-vessels, smaller one-cargo-hold-ships. At the same time, ships with "traditional" compartmentation carried on to be built. But, because there was no requirement to have transverse subdivisions watertight and because the naval architects had learned to substitute the transverse bulkhead as construction element by other means as far as strength aspects were concerned, even in such ships the subdivision quality built in voluntarily was lessened. Significant for this period is that the spread of subdivision quality became extremely large. This situation was also highly unsatisfactory from the systematic point of view.

With the new set of regulations, the average level of this quality will be boosted - it is, however, not possible at this moment of time to judge what the level actually resulting will be. A further effect will be a dramatic reduction of the margin between the worst and the best. This is the starting point for my critical attitude.

The requirements for dry cargo ships

Nevertheless what we now have is a solution, adequate to the present state of general knowledge, a solution for the problem to match demand orientated transport modes with the proof of a given level of safety after collision damages. The numerical calculation of survival capabilities performed by using probabilistic methods constitutes the means to objectively compare also extremely different designs as triggered by special transport demands, while at the same time giving the designer the utmost flexibility. Deterministic methods would not qualify for this, e.g. it is practically impossible to numerically evaluate a longitudinal bulkhead. - An indispensible precondition for using the probabilistic approach is assistence as provided by advanced computer programmes in performing a great number of individual flooding calculations with ease. After all, deterministic damage requirements can be applied by calculating the worst case or a few critical ones and covering all others by analogy considerations; the probabilistic method implies the necessity to carry the calculation of all cases to the numerical end result. So, it is about only by now that adequate tools exist.

I shall not embark on contents of the set of regulations for dry cargo ships; at the time of writing this paper the latest version is contained in IMO's MSC Circular 484; the ultimate decision will be taken at the MSC meeting in May, 1990, there will be slight amendments to the text of MSC Circular 484, but they will not change the principle nor have an effect on my comments.

Looking at the regulations as they stand now, gives all kinds of opportunities to argue. Before this is started, however, one should always recall that the result, the attained subdivision index A, is not a physically correct value but only a comparative measure - despite of the "correct" basis, i.e. statistical evidence, physical and mathematical laws. By this token simplifications in fixing numerical values like permeabilities and others are of very little significance, important is to fix the data in a suitable way and then apply them throughout. The result is a value for the quality of subdivision in relative figures, not in absolute ones.

Assessment of the level set

The required subdivision index R is calculated by a simple equation, which does not differentiate between any ship types. This is logical, produces problems, however, because the term dry cargo ship resembles a great variety of special types with very different survival properties. Of course, ultimately the levelling of all ships should be achieved, but a more pragmatic way would have reduced the surprises significantly, which we still can expect. A pragmatic approach would have been to set as a first step ship-type-dependent levels of R, which were just achievable by a "good" design, while allowing the principle design features to be kept. There would have been good reasons for going this way. They become evident, if we recall that the all-over level now set is also arbitrary as an average figure (Fig. 3). And even more important, it was set on the basis of some 50 exisiting ships, for which sample calculations were performed in the correctness of which I have serious doubts.

This situation now leads to the different ship types having different problems in conforming with the new regulations. The Ro-Ro-type and similar, such as self-unloaders will have great difficulties - despite of the possibility to have contributions to the A-value by calculating the effect of damages, which occur only below a watertight deck, i.e. the lower Ro-Ro-deck. For ships designed to carry long objects the designer has on principle the option of improving the A-value by shifting longitudinal bulkheads inwards. The question then is, how to use the outward spaces; their designation as tanks has its economic limits.

Application to container ships

Also in case of containerships compliance with the new regulations is not as easy as it appears in the first place. On principle watertight subdivisions matching with 40' containers would result in sufficient survival capability, if there were not the under-deck-passageways which normally extend openly over the whole cargo area length and which give open access to the cargo holds. So, even if all smaller contributions to A by spaces or groups of spaces formed by longitudinal and also horizontal subdivisions are accounted for, R might not be reached for present day containership designs. To improve the situation the designers can go for a transverse subdivision of the passageways, this solution will certainly not be favoured to often because of the obvious difficulties. - The ultimate possibility to increase the A-value is an increase in the intact stability values, i.e. the voluntary shifting of the minimum-GM-curve away from A.167-compliance to higher values. This, however, would effect directly the economy of the vessel in a negative way. Such a mode might also be undesirable out of safety considerations because higher GM values mean higher accelleration forces e.g. on the lashing equipment.

As already pointed out, internal openings leading to progressive flooding play a key role in the assessment of the A-value. At this point I would make a strong plea for worldwide uniformity from the onset, in particular with respect to the treatment of any internal openings. The very limited experience existing through comparative test calculations indicates the danger of heavily varying end results depending on the attitude taken regarding such openings and - if any - their closing facilities.

Operation and damage requirements

Let me at last return to operational aspects. It was already explained that many ships will have to be subjected to noticeable limitations in their loading flexibility because the subdivision index attained is not high enough, if the ship is loaded to the intact stability limits. This constitutes a temptation for the Master, which must not be underestimated : He has learned that his ship is safe as long as he adheres to the intact stabiltiy requirements. And now he should observe higher intact values, i.e. he should stop loading before his ship's capabilities are used up, just to be able to cope with the unlikely event of a collision ? He will certainly in the argument with himself bring in his own skill in saying : I can and I shall do much to safeguard that my ship is not involved in a collision. - This temptation is great the more, because the possibilities for an Administration to check compliance with stability requirements are very limited.

One could hold against that a similar situation does exist for every Master of a passenger ship. However, there are differences in as much as the purpose of a passenger ship is to carry passengers, of course; and economic aspects are hardly ever involved here; there is not much room or even temptation to exceed the damage stability governed service limitation. To the contrary, to say it again, the slightest deviation from the limits to the lower side for a containership means, a few containers more will be carried!

Conclusion

In theory, I am sure, for a good part of ships to come after the damage survival requirements take effect (1.2.92) the question is solved, how to implement higher intact stability requirements, which in my view are necessary to keep the level of safety represented for smaller, older ships by A.167; the Gordian Knot is cut. In practice, however, we are left with the problem of control over the application. I do not generally advocate enforcement of regulaitons through rigorous control, but here, I am afraid, at least during the first years routine checks by flag or port authorities appear to be the only means. Such check requires particular skill on the side of the authorities and has as prerequisite sufficient documentation on board. International efforts are required for defining the minimum of such stability documentations for the individual voyage. Also in this area, the computers which are by now a normal equipment on all new ships will be a valuable help.

In the second part of this paper I have made rather critical remarks regarding the level of the required subdivision index. Naval architects will have severe problems with it for a number of designs. The big advantage is, however, that the marine world will become used to increased intact stability requirements. Consequently the problem of introducing what research reveals to be necessary for an adequate intact stability level of safety might and hopefully will vanish.

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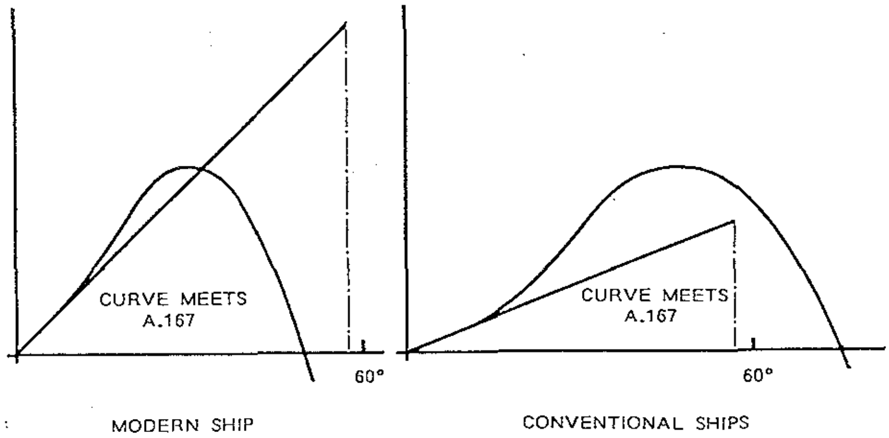

Fig. 1

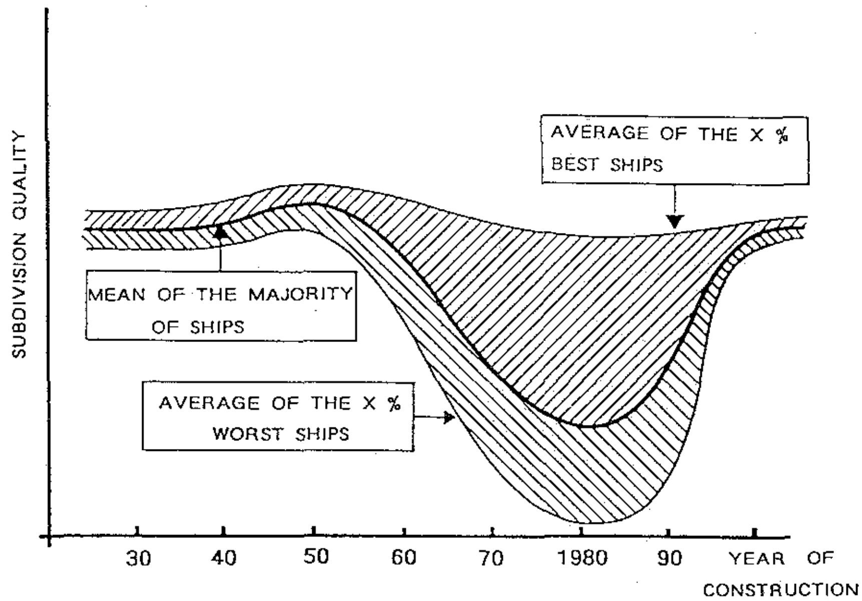

Fig. 2

MSC-57 Multi-national Collation

Includes all ships; sorted by Ship Subdivision Length

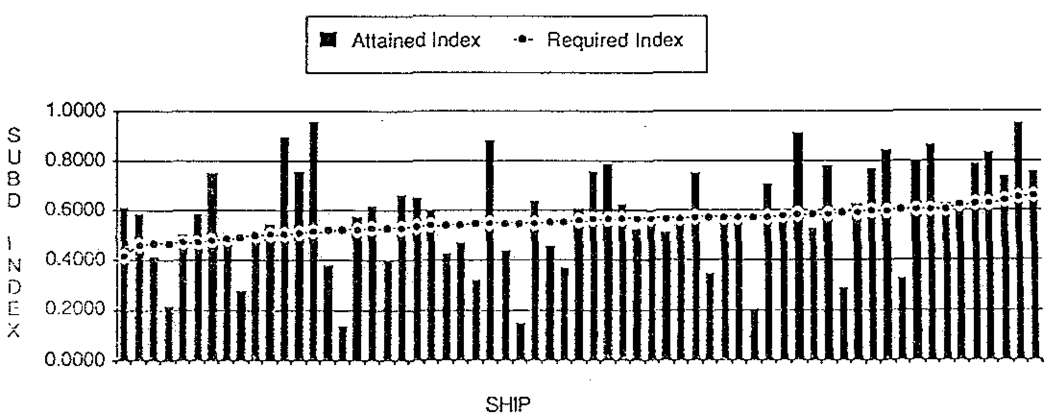

Fig. 3

RIDGELY-NEVITT FISHING VESSEL SERIES STABILITY IN LONGITUDINAL WAVES

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ABSTRACT

The paper deals with a quite simple procedure to evaluate the transverse stability curve and some statical and dynamical stability indices, both in still water and in longitudinal wave, of a fishing vessel derived according to the well-known Ridgely-Nevitt series.

The procedure allows to evaluate the stability indices during the preliminary design, before producing the lines plan or the table of offsets.

The calculations have been also carried out in following sea because it is well-known that in such condition the hull experiences a strong reduction of transverse stability which should be taken into account by designers.

The influence of wave height, of ship parameters considered in the series and of ship size on stability indices has been investigated as well.

INTRODUCTION

The assessment of the transverse stability of small fishing vessels is of the utmost importance in order to ensure an adequate level of safety during operations. Small ships, and in particular fishing vessels, can suffer from insufficient stability due to their size and form, as pointed out by casualty records.

The stability assessment should be carried out in the preliminary design with a suitable accuracy, in order to save time and money.

Deriving the hull form from a standard series allows to evaluate the propulsive power to be installed onboard without expensive model tests; unfortunately data on the stability characteristics of the hull of the series are very often not available.

This means that the table of offsets must be deduced during the preliminary stage of the project, using tentative main dimensions and form coefficients of the hull, in order to calculate stability indices; if stability criteria are not fulfilled, calculations must be iterated. Even if computer programs shorten the time needed for such tentative calculations, the authors feel that a set of analitycal relationships which give stability indices as a function of main dimensions and form coefficients of the hulls derived according to a given standard series is the most effective tool for the designer. Such relationships can be easily implemented in a optimizing CAD procedure, being restraints in the selection of the optimum hull.

The sistematic calculation of stability indices of hulls derived from a standard series, carried out covering all the parameter ranges investigated by the series, that is, the stability analysis of a series, can point out that some set of parameters (like B/T, $L/V^{1/3}$, C_B) for a given displacement volume, associated with reasonable center of gravity height and freeboard can lead to hull with lack of stability. Therefore such analysis should be also carried out during the preliminary stage of the series design, when the parent hull form and the parameter ranges to be tested are chosen, in order to avoid to test hull forms which, in practice, will not be utilized by designers.

Usually, the series hulls are derived on the basis of geometrical similitude. As shown in papers (1), (2), the characteristics of the immersed volume of an actual ship can be obtained by a proper transformation of corresponding data of a similar hull. Such method reduces the calculations needed to perform the stability analysis of a series, and provides a simple presentation of results. Moreover, the method applies as well in the case of a ship statically poised on a longitudinal sinusoidal wave (1), case in which the ship experiences a strong reduction of stability.

The present paper deals with the stability analysis of the well-known Ridgely-Nevitt series of fishing vessels.

Stability indices according to I.M.O. criteria have been extensively calculated covering the form parameter ranges explored by the series; results, put in non dimensional form according to the geometrical similitude theory, are presented as a set of equations which have been deduced using regression technique.

Particularly attention has been devoted to determine the minimum displacement volume at which a Nevitt hull still fulfills the stability criteria.

Finally, the case of the ship travelling in a following seaway has been analysed, confirming the strong dependence of the stability indices on the wave height to ship length ratio.

This paper completes the Ridgely-Nevitt series analysis reported in previous papers (3), (4), (5); in particular in reference (4) a set of equations which represents in analytical form the table of offsets of any hull derived from the series is given.

HULL SIMILITUDE THEORY

The theoretical background of hull geometrical similitude and some applications are reported in (1), (2); some definitions and results are briefly summarised hereafter.

Let Λ a hull having length L, beam B, design draught T and depth D, referred to the frame K X Y Z shown in fig.1; the XY plane is parallel to the design water plane. A hull Λ' is defined similar to if the co-ordinates X', Y', Z' of a generic point P' of Λ' are obtained from the co-ordinates X, Y, Z of a corresponding point P of Λ by a linear change, i.e.:

$$X' = X\,l;\; Y' = Y\,b;\; Z' = Z\,d$$

where l, b, d are positive constants called similitude ratios.

In such transformation the form coefficients, in particular the block coefficient and the depth to design draught ratio, do not change; the main dimensions of Λ' are:

$$L' = L\,l;\; B' = B\,b;\; D' = D\,d;\; T' = T\,d$$

The geometrical similitude theory still hold in presence of sinusoidal wave. The buoyancy center co-ordinates of a family of similar hulls can be expressed in the form:

$$\begin{aligned} X_B/L &= X_B/L\,(C_V,\, C_\phi,\, C_k) \\ 2Y_B/B &= 2Y_B/B\,(C_V,\, C_\phi,\, C_k) \qquad (1) \\ Z_B/T &= Z_B/T\,(C_V,\, C_\phi,\, C_k) \end{aligned}$$

the non-dimensional quantities C_V, C_ϕ, C_k are defined as follows:

$$C_V = V/V_o,\quad C_\phi = (B/T)\tan\phi,\quad C_k = h_w/(T\cos\phi)$$

where V and V_o are the actual and the design displacement volume, ϕ is the angle of heel at level trim, h_w is the wave height; the wave length is set equal to the ship length L, and the wave crest is amidship.

The above functions (1) can be evaluated for only one hull of the family and therefore such procedure is particularly useful in analysing the stability of a standard series which is constituted by some families of similar hulls.

Finally, as regards to the transversal moment of inertia of the water plane area, it holds:

$$I_{xx}/(L\,B^3) = I_{xx}/(L\,B^3)\,(C_V,\, C_k)$$

THE RIDGELY-NEVITT STANDARD SERIES

The Ridgely-Nevitt series of fishing vessels was developed in the 50's (6). After several comparative tank tests, a parent hull was chosen the main characteristics of which were (fig.1):

$$\Delta/(0.01L^3) = 300 \text{ tons/cuft} \quad L/V^{1/3} = 4.567$$

$C_p = .650$	$C_x = .760$	$L/B = 4.529$
$C_B = .494$	$C_w = .779$	$B/T = 2.30$

Three hulls were derived from the parent one with C_p equal to .554, .597, .700; the variation in C_p was not obtained by shifting the transverse sections, but re-drawing the hulls varying gradually the most significant geometric characteristics. From these four hulls, twelve hulls, three for each C_p, with $L/V^{1/3}$ equal to 3.852, 4.149 and 5.227 were derived keeping B/T constant; these derived hulls are similar to the parent ones. Therefore the series is constituted by four families of similar hulls, one for each C_p considered.

The sixteen hulls were tested in towing tank and results given as plot of C_R as a function of $V/\sqrt{L}$, C_p and $L/V^{1/3}$.

It is note-worthy that it was tested only one value of B/T; as suggested by Nevitt, results can be extended to the B/T range 2 ÷ 3.5 by adopting the correction factors relevant to the B.S.R.A. series.

STABILITY INDICES AND CRITERIA

Reference is made to I.M.O. recommendations on intact stability of fishing vessels (7) which are summarised in the following:

- GM > .35 m, where GM is the initial metacentrc height
- ϕ_x > 25° (pref. 30°), where ϕ_x is the angle of heel at which the righting lever GZ is maximum
- GZ_{30} > .20 m, where GZ_{30} is the rigthing lever at an angle of heel equal or greater than 30°
- E_{30} > .055 m rad
- E_{40} > .090 m rad
- $E_{40} - E_{30}$ > .030 m rad, where E_{40} and E_{30} are the restoring energies at an angle of heel of 40 resp. 30 degrees; if the downflooding angle ϕ_d is less than 40°, the $E_{\phi d}$ value must be considered.

The above criteria must be fulfilled for all conditions of loading.

In the following the expressions of the stability indices referred to in the above criteria are re-written in a form particularly suitable for the stability analysis of a standard series.

Because righting arms do not change varying longitudinal scale factor, being both immersed volume and its statical moment proportional to the longitudinal hull dimension, the stability indices are normalized by a transverse dimension, in particular the beam B. Such non-dimensional indices do not depend in still water on the $L/V^{1/3}$ ratio, so that the following relationship holds:

$$I/B = f(C_B, B/T, f/B, Z_G/D, C_V) \qquad (2)$$

where I is the generic index, f the freeboard at amidship, Z_G the vertical co-ordinate of the center of gravity and C_V the non-dimensional ratio defined above. The dependence of I on $L/V^{1/3}$ and on the design displacement volume V is explicited in the relation:

$$B = (V^{1/3}/(L/V^{1/3})^{1/2}) \, ((B/T)/C_B)^{1/2} \qquad (3)$$

With reference to fig.1, the metacentric height is:

$$GM = I_{xx}/V + Z_B - Z_G$$

and the ratio of GM to B is:

$$GM/B = (I_{xx}/(C_V C_x C_p \, LB^3)) \, (B/T) + \qquad (4)$$
$$+ (Z_B/T)/(B/T) - (f/B + 1/(B/T)) \, Z_G/D$$

The righting arm GZ at a given angle of heel ϕ is:

$$GZ = Y_B \cos\phi + Z_B \sin\phi - Z_G \sin\phi$$

and the ratio of GZ to B is:

$$GZ/B = (2Y_B/B)/2 \cos\phi + (Z_B/T)/(B/T) \sin\phi +$$
$$- (f/B + 1/(B/T)) \, Z_G/D \sin\phi$$

in the above relation $2Y_B/B$ and Z_G/D can be expressed in still water (C_k=0) as a function of C_V and C_ϕ according to (1); being:

$$\phi = \arctan(C_\phi/(B/T))$$

it can be rewritten in the form (2); moreover, it can be splitted as follows:

$$GZ/B = f_{GZ}(C_V, C_\phi, C_B, f/B, B/T) + \qquad (5)$$
$$- (f/B + 1/(B/T)) \, Z_G/D \sin\phi$$

where f_{GZ} is the buoyancy contribution which does not depend on Z_G/D and the second term is the weight contribution which does not depend on the hull form but only on its dimensionless parameters f/B and B/T.

The righting energies up to an heeling angle ϕ can be expressed in an analogue way; the relation:

$$E_\phi = \int_0^\phi GZ \, d\phi$$

can be written as follows:

$$E_\phi = f_\phi(C_V, C_\phi, C_B, f/B, B/T) + \qquad (6)$$
$$- (f/B + 1/(B/T)) \, (Z_G/D) \, (1 - \cos\phi)$$

The considerations about eq.(5) apply to eq. (6) as well.

As regards to the angle ϕ_x at which GZ is maximum, from the relationship:

$$GZ = B \, (GZ/B)$$

being B given by eq. (3) and (GZ/B) given by eq.(5), it can be deduced that for similar hulls varying V or $L/V^{1/3}$ the GZ values are all multiplied by a constant factor; this means that the characteristic angles of stability diagram (i.e. angle ϕ_x, angle of downflooding, angle of capsizing) do not depend on the design displacement volume V nor on the length to displacement ratio. As a conseguence of above, while it is possible to met the other stability criteria by increasing V or decreasing $L/V^{1/3}$, the requirement on ϕ_x cannot be fulfilled varying these design parameters; therefore, the stability criterion which fixes a minimum value for ϕ_x can be the most severe one.

As regards to the functional expression of ϕ_x, it holds:

$$C_{\phi x} = f(C_V, C_B, f/B, B/T, Z_G/D) \qquad (7)$$

where, as usual, is: $C_{\phi x} = (B/T) \tan(\phi_x)$

Finally, as regars to the maximum righting arm GZ_x, the subdivision adopted in eq.s (5) and (6) is no longer useful, because also the former term, which represent the buoyancy contribution, depends on Z_G/D being ϕ_x given by eq.(7).

When the hull is statically poised on a longitudinal sinusoidal wave whose length is equal to the ship length, the stability indices depend also on the ratio of wave height h_w to the ship length L; moreover, the indices to beam ratios depend on $L/V^{1/3}$ as well, even if in a moderate manner, because h_w/T increases as $L/V^{1/3}$ increases at a given wave slope h_w/L.

APPLICATION TO THE RIDGELY-NEVITT SERIES

The Ridgely-Nevitt series has been analysed from the point of view of stability as outlined above.

The $2Y_B/B$ and Z_B/T functions have been calculated for twelve hulls, obtained associating to the four values of C_p of the parent forms, namely .554, .597, .650, .700 three values of the depth to draught ratio D/T: 1.1, 1.3, 1.5. Calculations have been carried out for twenty values of C_V, fourteen values of C_ϕ, twelve values of C_k in order to allow calculation of stability diagram for the $L/V^{1/3}$, B/T parameter ranges covered by the series and for the wave height to ship length ratio range 0 ÷ .15 with a suitable accuracy. Then, righting arms have been calculated in still water considering three values of B/T, namely 2., 2.3 and 3.5, and two loading conditions, full loading and light, corresponding to the C_V values of 1 and .62 respectively. The latter value has been chosen on the base of statistical data taking into account that the series hulls have been designed for a full loading displacement volume ranging approximately from 100 to 1000 c.m. The righting arms have been calculated considering six values of Z_G/D ranging from .65 to .80.

Righting arms have been evaluated in sinusoidal wave also at full loading condition, considering only the value .650 of C_p, three values of B/T, 2.3, 3., 3.5, three values of $L/V^{1/3}$, 3.852, 4.567, 5.227 and the value of Z_G/D of .70 in order to make a comparison with the corresponding still water results.

Finally, seven stability indices, namely GM, GZ_{30}, E_{30}, E_{40}, $E_{40}-E_{30}$, ϕ_x, GZ_x have been calculated and results put in non-dimensional form as explained above.

Plots of non-dimensional values of considered indices in still water are reported in fig.s from 2 to 5 to show the stability trends with respect to examined variables.

According to relationships (4), (5) and (6) the first five indices to beam ratios decrease linearly as Z_G/D increases; GZ_x and $C_{\phi x}$ decrease with Z_G/D as well.

The plots reveals that all indices increase moderately as prismatic coefficient increases, as well as the functions $I_{xx}/(C_V C_x C_p LB^3)$, f_{30}, f_{40} and f_{GZ30}.

As regards to the B/T ratio, it is well known its marked influence on stability; fig.6 confirms this result and shows that all indices vary almost linearly with this parameter. Fig.7 shows the influence of h_w/L: it is evident that in presence of wave the indices decrease markedly: for instance GZ_{30}/B at h_w/L equal to 0.05 is an half than in still water. As said above, in presence of wave $L/V^{1/3}$ also influences the indices, see fig.8.

The influence of the freeboard to beam ratio should be analysed in conjunction with that one of the Z_G/D ratio. At a given Z_G/D dynamic stability, i.e. GZ_{30}, E_{30} and E_{40} increase as f/B increases up to a value of f/B behind which the unfavoreble effect of center of gravity raising prevailes. This typical pattern of the dynamic indices is evident in the reported figures.

The influence of f/B and Z_G/D on ϕ_x is particularly interesting. The quantity (B/T) $\tan\phi_x$ is plotted in fig.5 versus f/B; each curve is relevant to a constant Z_G/D while the B/T and C_p values considered are 2.3 and .554 resp. The plot reveals that the minimum ϕ_x angle requested by I.M.O. criterion can be achieved, at a given f/B, only up to a maximum allowable value of Z_G/D. A curve of these values can be obtained intersecting the plot in fig.5 with an horizontal line drawn at $C_{\phi x}=1.073$. Such curve is reported in fig.9, where the curves relevant to C_p .700 and B/T 3.5 are plotted as well.

The diagram in fig.9 shows that the maximum allowable value of Z_G/D, for B/T 2.3 and reasonable values of f/B are quite low. For instance, for f/B=.11, C_p=.700, Z_G/D must be lower than .66 to met the ϕ_x criterion. Of course, increasing B/T improves the ϕ_x index: from the diagram it can be deduced that the typical values .70 for Z_G/D and .11 for f/B fulfill the criterion if the B/T ratio is greater than 3.0.

Because the resistance data are available only for the 2.3 value of B/T, the analysis continues for such value, even if data reported in the following allow to perform similar analysis for different values.

In particular in the following the maximum Z_G/D value deduced from fig.9 is associated to each f/B value considered, being this the most unfavourable case. Table 1 reports some readings made in fig.9 which have been used in the following calculations.

TABLE 1

f/B min. values vs. Z_G/D at B/T 2.3

C_p \ Z_G/D	.70	.72	.73	.75
.554	.130	.143	.152	.172
.700	.128	.134	.146	.161

Fig.10 reports the diagram of the initial metacentric height versus the design volume; each curve is relevant to a given value of the parameters $L/V^{1/3}$, f/B and Z_G/D, being the last two parameters associated as explained above. The $L/V^{1/3}$ values considered, 3.852 and 5.227 are the minimum and the maximum value for which resistance data are given.

Reading the intersections of an horizontal line corresponding to GM equal to .35 m with the plotted curves the diagram in fig.11 can be drawn, which represents the maximum allowable length to displacement ratio versus design volume, for a given minimum metacentric height and again at constant Z_G/D or f/B.

The above diagram can be plotted for the other indices as well: these diagrams, see fig.s from 12 to 15, together with that one in fig.9, allow to assess if a given choice of $L/V^{1/3}$, Z_G/D, C_p, f/B and V fulfills the stability criteria at full loading condition.

In practice, the curves represent the equation:

$$(L/V^{1/3})_{max}=(V^{2/3}/I^2_{min})((B/T)/(C_x C_p))(I/B)^2$$

The above analysis can be repeated for different loading conditions: for instance, diagrams in fig.s 16, 17 and 18 are relevant to a displacement volume of 62% of full loading volume. The curves are plotted versus the design volume, and the ratios Z_G/D and f/B are associated according to table 1. This means that it has been assumed that the center of gravity height at light condition is the same as at full loading condition. Of course such diagram can be drawn, using the equation set given in the following, assuming different center of gravity behaviour.

It is note-worthy that at light condition the indices increase for C_p equal to .700 and decrease for C_p equal to .554; therefore for this aspect also low values of C_p are unfavoureable.

It is interesting to determine the minimum design volume at which the stability criteria are met. The stability indices increase as the length to displacement ratio decreases; therefore the minimum value of this parameter is to be considered. Reading the above diagrams at $L/V^{1/3}$ equal to 3.852 leads to the curves plotted in fig.19 where the abscissa is the Z_G/D ratio and each curve represents the equation:

$$V_{min}=((L/V^{1/3})/((B/T)/(C_x C_p)))^{3/2}(I/(I/B)_{min})^3$$

where Z_G/D and f/B are related as usual.

Using the above equation a comparison among the criteria can be made. For C_p .700 the most severe one is GZ_{30} at C_V 1 for Z_G/D up to about .72, and $E_{40}-E_{30}$ at C_V 1 for higher values of Z_G/D; for C_p .554 the values of V_{min} are higher than in the previous case, while the most severe criteria are GZ_{30} at C_V 1 up to Z_G/D about .72, $E_{40}-E_{30}$ at C_V 1 for Z_G/D from .72 to .735 and E_{30} at light condition for higher values of center of gravity height.

It is note-worthy that the design displacement volume of 100 c.m does not fulfills the criteria if Z_G/D is equal to or greater than .70.

Analytical expressions of stability indices

Approximate expressions of the non-dimensional stability indices have been derived using regression technique. According to relationships (4), (5), (6) and (7), at given B/T and C_V, the functions can be divided into three groups: $I_{xx}/(C_x C_p LB^3)$, Z_B/T which depend on C_p; f_{GZ30}, f_{30} and f_{40} which depend on C_p and f/B, and GZ_x, $ø_x$ which depend on C_p, f/B and Z_G/D. All the indices depend quite linearly on C_p; the dependence of GZ_x and $ø_x$ on Z_G/D is also quite linear, while the dependence of all indices on f/B can be assumed as quadratic. Therefore polynomial expressions have been used, containing the variables Z_G/D and C_p raised to the first power and the variable f/B raised to the second power. Higher order terms have not been considered in order to limit the number of terms; at the same time the achieved accuracy is felt suitable for the purpose.

As regards to the first group, two sets of equations are given, the former relevant to full loading condition, i.e. C_V=1, the latter to light loading condition, i.e. C_V=.62.

As regards to the second group, four sets of equations are given, relevant to the above C_V values and to two values of B/T, namely 2.3 and 3.5. Results relevant to B/T equal to 2.0 are not reported because the series shows a lack of stability for such low value of this parameter.

The generic equation is in the form:

$$f = C + \Sigma\, a_i\, Y^{exp}\, Z^{exp}$$

where is $Y=C_p$ and Z=f/B; as regards to the third group, four sets of equations are given as well, being each of them in the form:

$$f = C + \Sigma\, a_i\, X^{exp}\, Y^{exp}\, Z^{exp}$$

where is $X=C_p$, $Y=Z_G/D$ and Z=f/B.

CONCLUSIONS

The stability criteria are restraints which can influence drammatically the choice of main form parameters of a small fishing vessel hull.

The stability analysis must be carried out since in the preliminary stage of the project, taking into account all criteria and not only the GM one; in particular the $ø_x$ criterion limits the maximum allowable center of gravity height at a given freeboard irrespective of the design hull volume.

Hence the need to furnish the designer with approximate expressions of stability indices, which can be implemented in a CAD procedure.

When designing a standard series, a preliminary stability analysis is of the utmost importance, because otherwise the hull tested could not be useful for the designer.

As regards to the Nevitt series, the B/T value tested leads to low values of stability indices and cannot be used for small displacement volume hulls.

The provided approximate analitycal expressions of stability indices can be used to easily and quickly judge the stability of a Nevitt hull, and moreover can be also applied to hulls not derived from the Nevitt series but of similar form.

ACKNOWLEDGEMENTS

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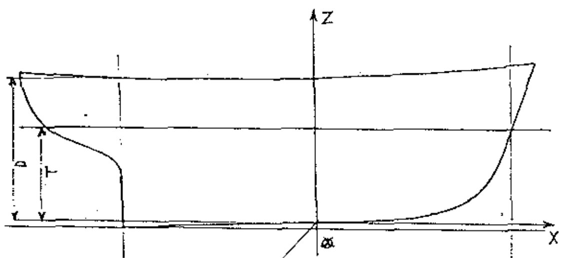


Fig. 1

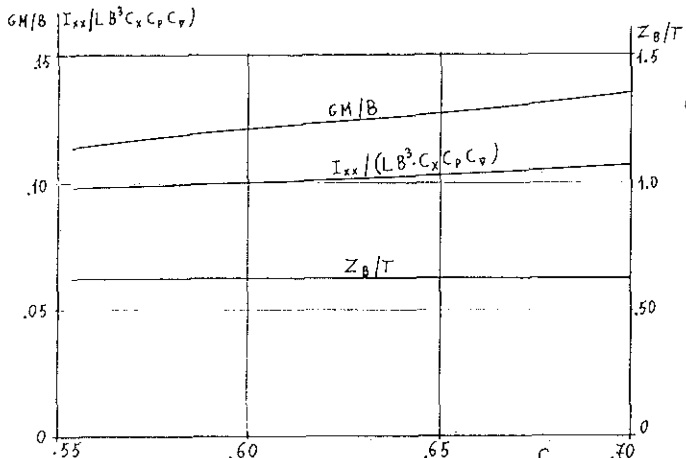


Fig. 2

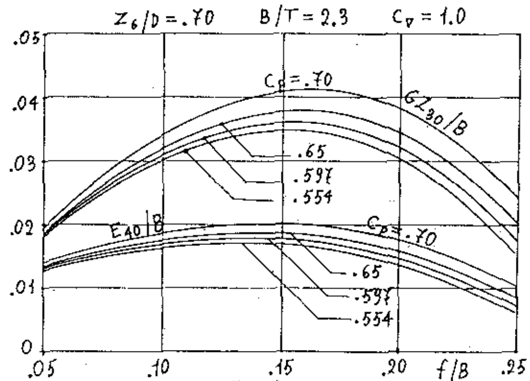
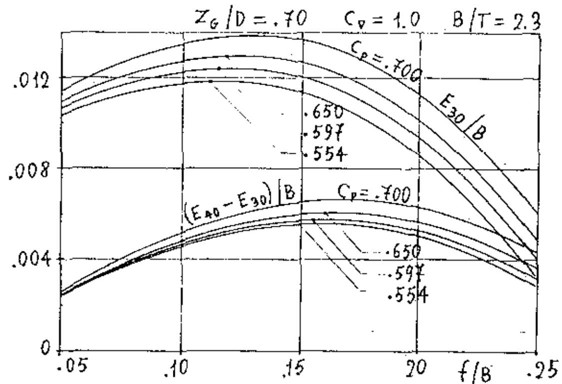


Fig. 3



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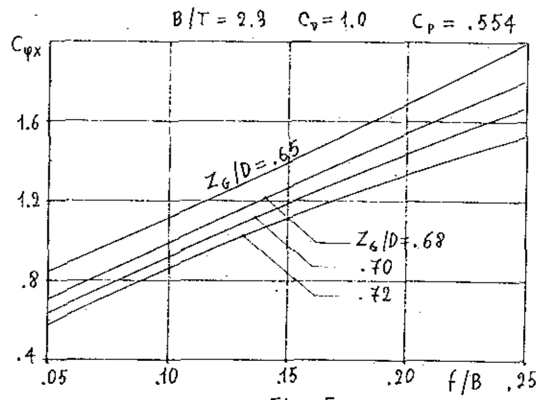

Fig. 5

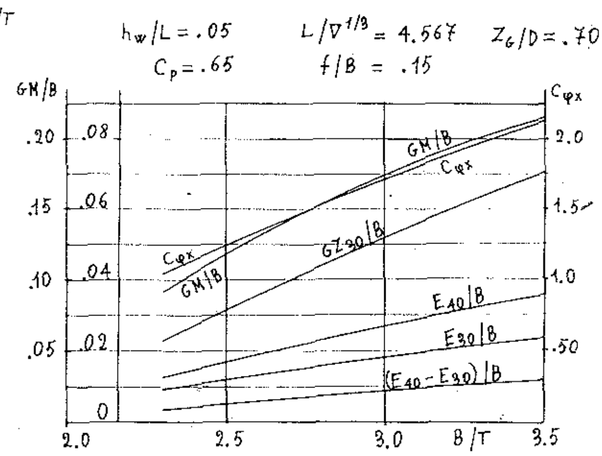

Fig. 6

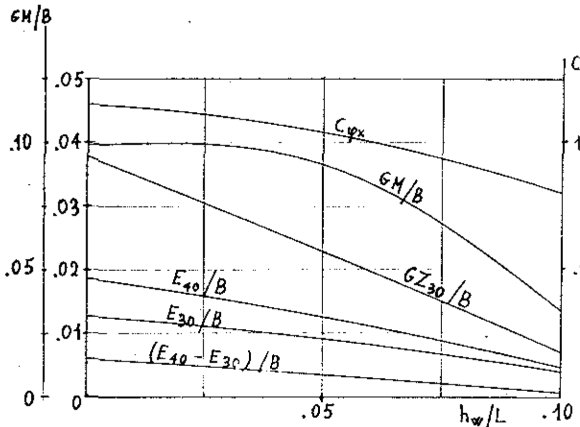

Fig. 7

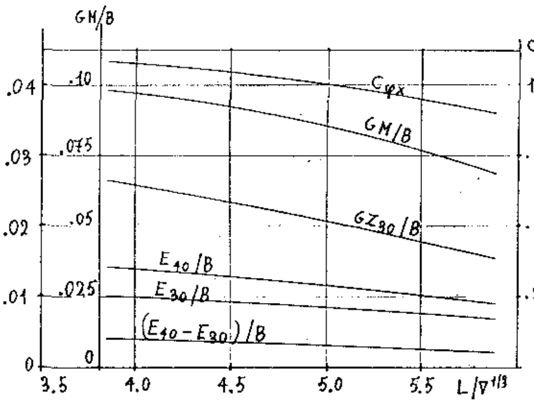

Fig. 8

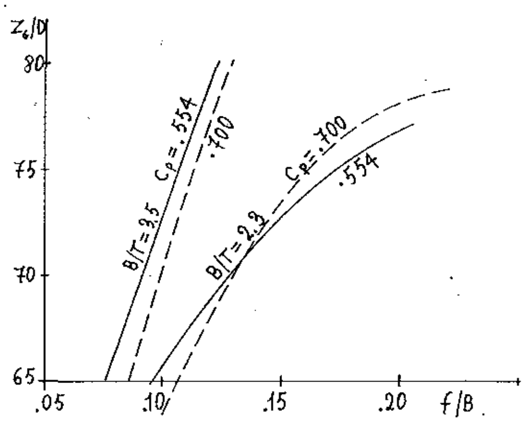

Fig. 9

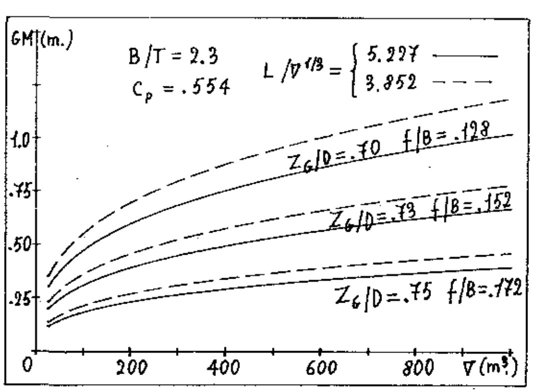

Fig. 10

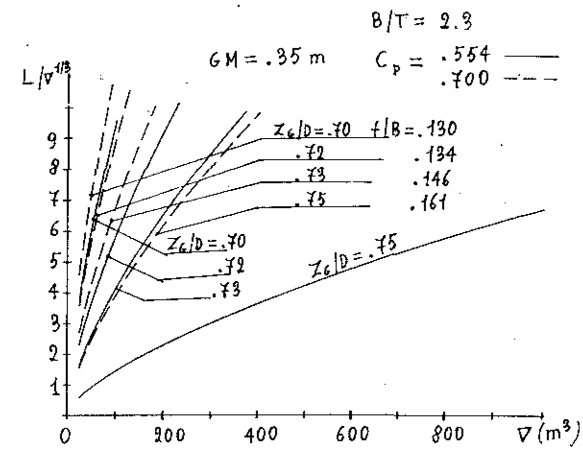

Fig. 11

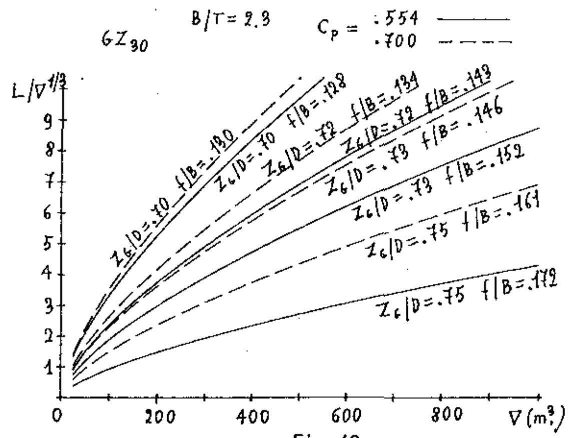

Fig. 12

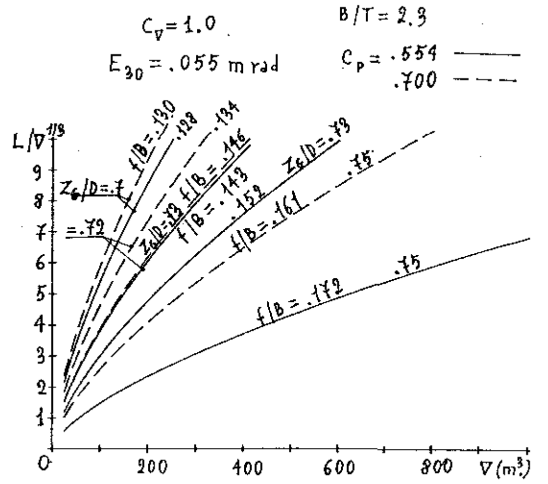

Fig. 13

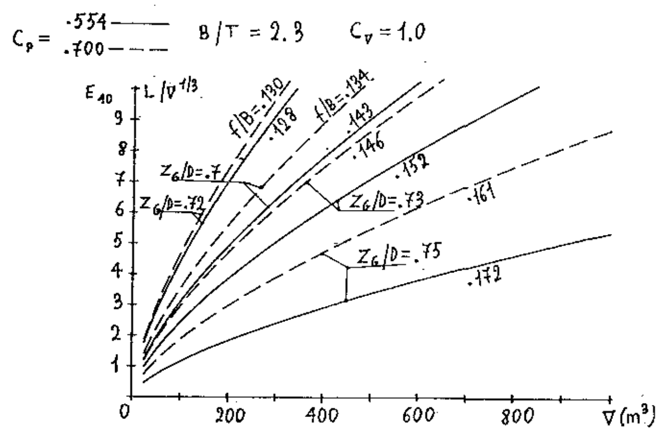

Fig. 14

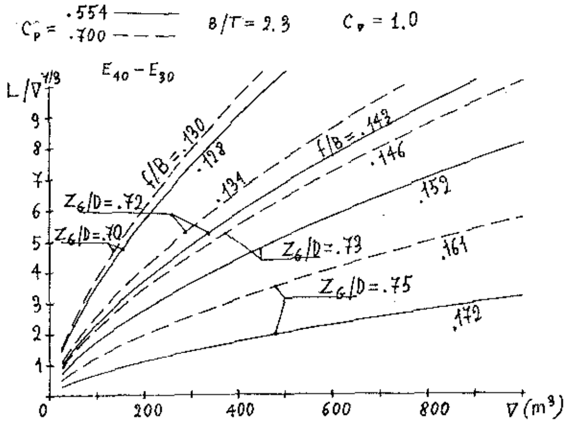

Fig. 15

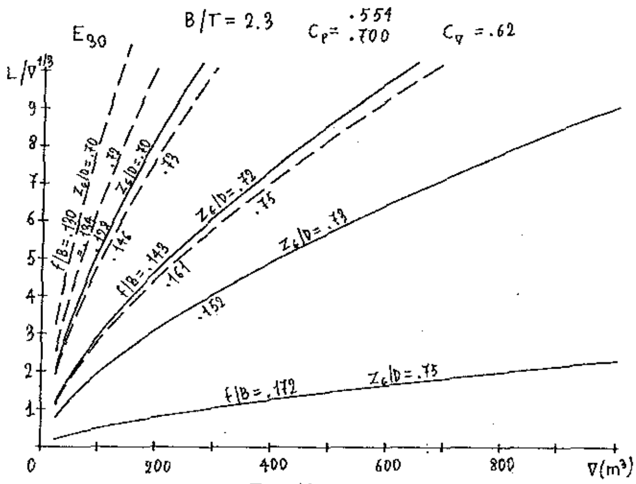

Fig. 18

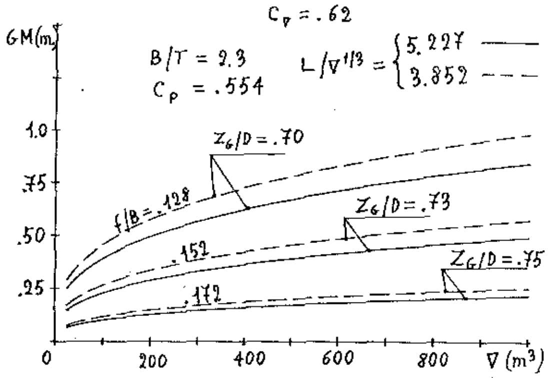

Fig. 16

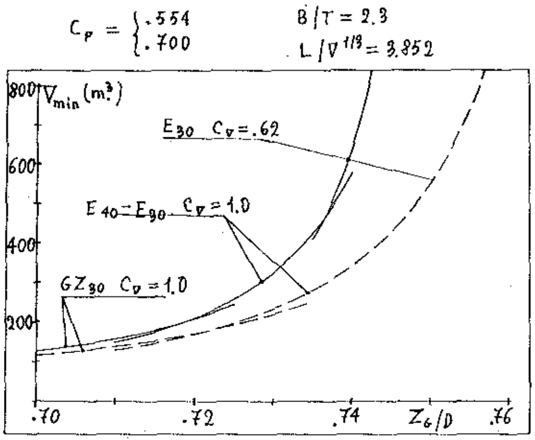

Fig. 19

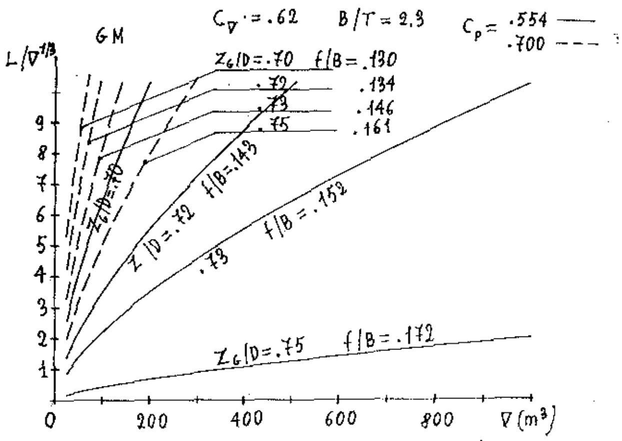

Fig. 17

$C_V = 1$

$I_{xx}/(C_x C_p) = .0562 + .0733\ C_p$ $\qquad Z_B/T = .624$

B/T = 2.3

f 30

C= +4.676431E-02

VAR.	COEFF.
Y^ 1.00 Z^ 0.00	+8.657376E-04
Y^ 0.00 Z^ 1.00	+1.168402E-01
Y^ 1.00 Z^ 1.00	+1.364133E-01
Y^ 0.00 Z^ 2.00	-3.032093E-01
Y^ 1.00 Z^ 2.00	-2.598148E-01

f 40

C= +6.906531E-02

VAR.	COEFF.
Y^ 1.00 Z^ 0.00	+1.075139E-02
Y^ 0.00 Z^ 1.00	+3.467723E-01
Y^ 1.00 Z^ 1.00	+4.882727E-02
Y^ 0.00 Z^ 2.00	-8.592824E-01
Y^ 1.00 Z^ 2.00	+9.249747E-02

f 40-30

C= +3.031557E-02

VAR.	COEFF.
Y^ 1.00 Z^ 0.00	-2.172154E-03
Y^ 0.00 Z^ 1.00	+1.176214E-01
Y^ 1.00 Z^ 1.00	+7.248697E-02
Y^ 0.00 Z^ 2.00	-2.328274E-01
Y^ 1.00 Z^ 2.00	-9.512140E-02

GZx

C= +9.818937E-02

VAR.	COEFF.
X^ 1.00 Y^ 0.00 Z^ 0.00	-8.539510E-02
X^ 0.00 Y^ 1.00 Z^ 0.00	-1.240074E-01
X^ 0.00 Y^ 0.00 Z^ 1.00	+1.253119E+00
X^ 0.00 Y^ 1.00 Z^ 1.00	-1.568432E+00
X^ 1.00 Y^ 0.00 Z^ 1.00	+2.765905E-01
X^ 1.00 Y^ 1.00 Z^ 0.00	+1.189819E-01
X^ 1.00 Y^ 1.00 Z^ 1.00	+1.816539E-01
X^ 0.00 Y^ 0.00 Z^ 2.00	-1.272635E+00
X^ 1.00 Y^ 0.00 Z^ 2.00	-6.332553E-01
X^ 0.00 Y^ 1.00 Z^ 2.00	+3.016613E-01

f GZ 30

C= +1.554840E-01

VAR.	COEFF.
Y^ 1.00 Z^ 0.00	-1.233377E-02
Y^ 0.00 Z^ 1.00	+6.125134E-01
Y^ 1.00 Z^ 1.00	+4.583738E-01
Y^ 0.00 Z^ 2.00	-1.399296E+00
Y^ 1.00 Z^ 2.00	-6.706314E-01

C phix

C= +4.130384E+00

VAR.	COEFF.
X^ 1.00 Y^ 0.00 Z^ 0.00	-2.078387E+00
X^ 0.00 Y^ 1.00 Z^ 0.00	-5.189787E+00
X^ 0.00 Y^ 0.00 Z^ 1.00	+1.977727E+01
X^ 0.00 Y^ 1.00 Z^ 1.00	-2.279493E+01
X^ 1.00 Y^ 0.00 Z^ 1.00	-4.183435E+01
X^ 1.00 Y^ 1.00 Z^ 0.00	+2.605729E+00
X^ 1.00 Y^ 1.00 Z^ 1.00	+6.554008E+01
X^ 0.00 Y^ 0.00 Z^ 2.00	+7.291685E+00
X^ 1.00 Y^ 0.00 Z^ 2.00	+1.089474E+02
X^ 1.00 Y^ 1.00 Z^ 2.00	-1.837374E+02

B/T = 3.5

f 30

C= +3.490388E-02

VAR.	COEFF.
Y^ 1.00 Z^ 0.00	-1.929945E-03
Y^ 0.00 Z^ 1.00	+2.853763E-01
Y^ 1.00 Z^ 1.00	+2.600811E-01
Y^ 0.00 Z^ 2.00	-9.715722E-01
Y^ 1.00 Z^ 2.00	-6.936388E-01

f 40

C= +5.591778E-02

VAR.	COEFF.
Y^ 1.00 Z^ 0.00	-3.836129E-03
Y^ 0.00 Z^ 1.00	+4.790016E-01
Y^ 1.00 Z^ 1.00	+3.455194E-01
Y^ 0.00 Z^ 2.00	-1.384020E+00
Y^ 1.00 Z^ 2.00	-8.702985E-01

f 40-30

C= +2.101278E-02

VAR.	COEFF.
Y^ 1.00 Z^ 0.00	-1.906324E-03
Y^ 0.00 Z^ 1.00	+1.936888E-01
Y^ 1.00 Z^ 1.00	+8.536207E-02
Y^ 0.00 Z^ 2.00	-4.127868E-01
Y^ 1.00 Z^ 2.00	-1.762022E-01

f GZ 30

C= +1.102931E-01

VAR.	COEFF.
Y^ 1.00 Z^ 0.00	-1.448565E-02
Y^ 0.00 Z^ 1.00	+1.022437E+00
Y^ 1.00 Z^ 1.00	+6.556387E-01
Y^ 0.00 Z^ 2.00	-2.428278E+00
Y^ 1.00 Z^ 2.00	-1.640931E+00

GZx

C= +5.705907E-02

VAR.	COEFF.
X^ 1.00 Y^ 0.00 Z^ 0.00	-1.775243E-02
X^ 0.00 Y^ 1.00 Z^ 0.00	-5.915463E-02
X^ 0.00 Y^ 0.00 Z^ 1.00	+1.418398E+00
X^ 0.00 Y^ 1.00 Z^ 1.00	-1.230172E+00
X^ 1.00 Y^ 0.00 Z^ 1.00	+3.714162E-01
X^ 1.00 Y^ 1.00 Z^ 0.00	+1.454849E-02
X^ 1.00 Y^ 1.00 Z^ 1.00	+4.011057E-01
X^ 0.00 Y^ 0.00 Z^ 2.00	-1.872601E+00
X^ 1.00 Y^ 0.00 Z^ 2.00	-9.305001E-01
X^ 1.00 Y^ 1.00 Z^ 2.00	-1.415004E+00

C phix

C= +4.992752E+00

VAR.	COEFF.
X^ 1.00 Y^ 0.00 Z^ 0.00	-2.526938E+00
X^ 0.00 Y^ 1.00 Z^ 0.00	-5.652576E+00
X^ 0.00 Y^ 0.00 Z^ 1.00	-6.763562E-01
X^ 0.00 Y^ 1.00 Z^ 1.00	+1.428304E+01
X^ 1.00 Y^ 0.00 Z^ 1.00	-9.736236E+00
X^ 1.00 Y^ 1.00 Z^ 0.00	+2.843989E+00
X^ 1.00 Y^ 1.00 Z^ 1.00	+1.686629E+01
X^ 1.00 Y^ 0.00 Z^ 2.00	+1.180748E+02
X^ 0.00 Y^ 1.00 Z^ 2.00	+8.563095E-01
X^ 1.00 Y^ 1.00 Z^ 2.00	-1.900634E+02

$C_V = .62$ $\qquad I_{xx}/(C_x C_p) = .0200 + .0960\ C_p$ $\qquad Z_B/T = .471$

B/T = 2.3

f 30

C= +4.467360E-02

VAR.	COEFF.
Y^ 1.00 Z^ 0.00	+3.232653E-02
Y^ 0.00 Z^ 1.00	+1.394599E-02
Y^ 1.00 Z^ 1.00	+1.000366E-02
Y^ 0.00 Z^ 2.00	-3.665148E-02
Y^ 1.00 Z^ 2.00	-1.961710E-02

f 40

C= +7.643033E-02

VAR.	COEFF.
Y^ 1.00 Z^ 0.00	+4.824347E-02
Y^ 0.00 Z^ 1.00	+7.048283E-02
Y^ 1.00 Z^ 1.00	+3.713184E-02
Y^ 0.00 Z^ 2.00	-1.597673E-01
Y^ 1.00 Z^ 2.00	-6.919099E-02

f 40-30

C= +3.178406E-02

VAR.	COEFF.
Y^ 1.00 Z^ 0.00	+1.586377E-02
Y^ 0.00 Z^ 1.00	+5.616520E-02
Y^ 1.00 Z^ 1.00	+2.784533E-02
Y^ 0.00 Z^ 2.00	-1.220511E-01
Y^ 1.00 Z^ 2.00	-5.162694E-02

f GZ 30

C= +1.647248E-01

VAR.	COEFF.
Y^ 1.00 Z^ 0.00	+9.364208E-02
Y^ 0.00 Z^ 1.00	+1.820634E-01
Y^ 1.00 Z^ 1.00	+1.289968E-01
Y^ 0.00 Z^ 2.00	-4.592924E-01
Y^ 1.00 Z^ 2.00	-2.476142E-01

GZx

C= +1.679964E-01

VAR.	COEFF.
X^ 1.00 Y^ 0.00 Z^ 0.00	+4.674655E-02
X^ 0.00 Y^ 1.00 Z^ 0.00	-2.240152E-01
X^ 0.00 Y^ 0.00 Z^ 1.00	+1.460627E+00
X^ 0.00 Y^ 1.00 Z^ 1.00	-2.363547E+00
X^ 1.00 Y^ 0.00 Z^ 1.00	-1.029392E+00
X^ 1.00 Y^ 1.00 Z^ 0.00	+6.325663E-02
X^ 1.00 Y^ 1.00 Z^ 1.00	+1.861531E+00
X^ 1.00 Y^ 0.00 Z^ 2.00	-3.119424E-01
X^ 0.00 Y^ 1.00 Z^ 2.00	-8.482983E-02
X^ 1.00 Y^ 1.00 Z^ 2.00	-9.147765E-01

C phix

C= +1.587693E+00

VAR.	COEFF.
X^ 1.00 Y^ 0.00 Z^ 0.00	+9.538187E-01
X^ 0.00 Y^ 1.00 Z^ 0.00	-2.578002E-01
X^ 0.00 Y^ 0.00 Z^ 1.00	+3.007552E+01
X^ 0.00 Y^ 1.00 Z^ 1.00	-4.832483E+01
X^ 1.00 Y^ 0.00 Z^ 1.00	-2.873536E+01
X^ 1.00 Y^ 1.00 Z^ 0.00	-1.995671E+00
X^ 1.00 Y^ 1.00 Z^ 1.00	+5.957419E+01
X^ 0.00 Y^ 0.00 Z^ 2.00	+1.102796E+02
X^ 1.00 Y^ 0.00 Z^ 2.00	-9.247500E+01
X^ 0.00 Y^ 1.00 Z^ 2.00	-6.544049E+01

B/T = 3.5

f 40-30

C= +2.589103E-02

VAR.	COEFF.
Y^ 1.00 Z^ 0.00	+1.352435E-02
Y^ 0.00 Z^ 1.00	+1.402584E-01
Y^ 1.00 Z^ 1.00	+5.776851E-02
Y^ 0.00 Z^ 2.00	-3.237608E-01
Y^ 1.00 Z^ 2.00	-2.076608E-01

f 30

C= +4.101508E-02

VAR.	COEFF.
Y^ 1.00 Z^ 0.00	+3.727995E-02
Y^ 0.00 Z^ 1.00	+1.073958E-01
Y^ 1.00 Z^ 1.00	+3.075387E-02
Y^ 0.00 Z^ 2.00	-3.760525E-01
Y^ 1.00 Z^ 2.00	-4.813657E-02

f 40

C= +6.690481E-02

VAR.	COEFF.
Y^ 1.00 Z^ 0.00	+5.080856E-02
Y^ 0.00 Z^ 1.00	+2.476733E-01
Y^ 1.00 Z^ 1.00	+8.845740E-02
Y^ 0.00 Z^ 2.00	-6.998792E-01
Y^ 1.00 Z^ 2.00	-2.555636E-01

f GZ 30

C= +1.348502E-01

VAR.	COEFF.
Y^ 1.00 Z^ 0.00	+8.704828E-02
Y^ 0.00 Z^ 1.00	+7.379961E-01
Y^ 1.00 Z^ 1.00	+2.124995E-01
Y^ 0.00 Z^ 2.00	-2.161293E+00
Y^ 1.00 Z^ 2.00	-5.084427E-01

GZx

C= +9.467458E-02

VAR.	COEFF.
X^ 1.00 Y^ 0.00 Z^ 0.00	+9.949129E-02
X^ 0.00 Y^ 1.00 Z^ 0.00	-8.643051E-02
X^ 0.00 Y^ 0.00 Z^ 1.00	+1.433185E+00
X^ 0.00 Y^ 1.00 Z^ 1.00	-1.546548E+00
X^ 1.00 Y^ 1.00 Z^ 0.00	-1.130480E-02
X^ 1.00 Y^ 1.00 Z^ 1.00	+2.703004E-01
X^ 0.00 Y^ 0.00 Z^ 2.00	-1.787013E+00
X^ 1.00 Y^ 0.00 Z^ 2.00	-2.415963E+00
X^ 0.00 Y^ 1.00 Z^ 2.00	+6.196170E-01
X^ 1.00 Y^ 1.00 Z^ 2.00	+1.964235E+00

C phix

C= +6.794538E+00

VAR.	COEFF.
X^ 1.00 Y^ 0.00 Z^ 0.00	-5.558897E+00
X^ 0.00 Y^ 1.00 Z^ 0.00	-7.119511E+00
X^ 0.00 Y^ 0.00 Z^ 1.00	-8.488039E+01
X^ 0.00 Y^ 1.00 Z^ 1.00	+1.267901E+02
X^ 1.00 Y^ 0.00 Z^ 1.00	+1.275472E+02
X^ 1.00 Y^ 1.00 Z^ 0.00	+7.046967E+00
X^ 1.00 Y^ 1.00 Z^ 1.00	-1.649004E+02
X^ 0.00 Y^ 0.00 Z^ 2.00	+5.823340E+02
X^ 1.00 Y^ 0.00 Z^ 2.00	-7.826049E+02
X^ 0.00 Y^ 1.00 Z^ 2.00	-7.718434E+02
X^ 1.00 Y^ 1.00 Z^ 2.00	+9.763297E+02

Application of Modern Geometric Methods for Dynamical Systems to the Problem of Vessel Capsizing with Water-on-deck

Jeffrey M Falzarano* Armin W Troesch†

Abstract

The ability to resist capsizing is a fundamental requirement in ship design. Vessel capsizing is a large amplitude dynamic phenomena requiring the consideration of nonlinear dynamics and hydrodynamics. However, traditional ship stability analysis is based upon nonlinear hydrostatics while ship motions analysis is based upon linear dynamics. In lieu of numerical simulation or approximate local analysis (near a single equilibria), the approach of this paper is to apply modern geometric methods in analyzing vessel stability. These modern methods are not limited to the size of the nonlinearities and are capable of analyzing the global (trajectories near one or more singular points) system behavior. Specifically, the periodically forced roll equation of motion for a typical fishing vessel as effected by a reduced righting arm due to water-on deck, increased damping due to bilge keels, and various regular wave amplitudes and frequencies is studied.

Introduction

Traditional ship stability criteria are based solely upon the analysis of the static roll restoring moment curve. Although, this curve was found to be an important vessel characteristic in assuring vessel safety (Falzarano, 1988), other vessel characteristics are also significant. These include the amount of hydrodynamic and viscous roll damping as effected by the size of bilge keels, the frequency and magnitude of the wave exciting force, initial conditions, and the presence of water-on-deck. In this work, the effect of important design parameters and the various modeling approximations on the resulting dynamics are studied.

Background

Traditional Ship Stability Analysis

An early analysis of undamped, unforced vessel dynamics was undertaken by Mosley in 1850 (Mosley, (1850)). Mosley's work forms the theoretical basis of modern ship stability criteria. In his paper, Mosley studied the unforced, undamped, single degree of freedom roll equation of motion. He equated the overturning energy and the restoring energy in order to judge vessel safety. He arbitrarily choose his initial conditions so that his energy integral would not include integration constants.

Rahola (1939), in his doctoral dissertation, analyzed the righting arms of a number of capsized Finish fishing vessels to determine what are the important external forces and what should be the required righting arm curve to assure vessel safety.

Mosley's analysis is the basis for the modern ship stability weather criteria. Rahola's work is the justification for empirically required GM's which are determined by analyzing vessel casualty data. These two methods form the rationale for state-of-the-art ship stability analysis.

The Role of Simulation

Because the exact dynamics problem of a vessel rolling near capsizing is highly nonlinear, it is impossible to solve the problem exactly in closed form. In order to make real progress, approximations must be made. Even with these approximate models, numerical simulation is often used. Due to size and speed limitations of computers, though, numerical simulation is limited in problem size and simulation time. In a previous work, one of the authors (Falzarano, (1988)) developed a simulation code which combined a nonlinear time domain motion simulation code (Paulling (1974)) with the simultaneous solution of the water-on-deck hydraulics problem (Dillingham and Falzarano, (1986)). The usefulness of the program was limited by its computational complexity. Generally, when simulation is required, optimizing the speed and efficiency of the computer calculations is valuable (King,

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(1990)). Since only a finite number of simulations can be performed, guidance on the system's sensitivity to initial conditions and parameter values is crucial. In the remainder of this paper, techniques are described which can be used by themselves or in conjunction with simulators to predict vessel capsizing.

Alternative Approaches

One method to assess boundedness of motion (i.e., non-capsizing) is to apply Lyapunov's direct method. Much work has been done using this approach (e.g., Odabassi, (1979)). However, Lyapunov techniques yield an approximate bound on the motion and the efficiency (accuracy) of the bound is often difficult to assess.

Other techniques include approximate deterministic criteria (Virgin, (1989)) and studying how the resulting amplitude of the motion is effected by changing important parameters (e.g., Cardo, Francescutto, and Nabergoj, (1982), Nayfeh and Sanchez, (1988), and Falzarano, Stiendl, Troesch and Troger, (1990)).

The Ship Dynamics Problem

The intent of this section is to obtain a single degree of freedom roll equation of motion from the complete six degree of freedom equations. Although these assumptions restrict the generality of the results they are systematic and consistent and not usually explained in other works. Basically, these approximations assume small motions in all modes of motion but roll and the existence of a coordinate system origin (i.e., roll center) which approximately decouples the roll motion from sway. As a result, these approximations are most accurate in beam seas.

Determining the general (nonlinear) force vector[1] of the six degree of freedom equations of motion is difficult. For general body motion, it has not yet been done. The determination of these forces requires solving the nonlinear three-dimensional hydrodynamics problem of a body floating on the free surface.

Following Vugts, (1970), all velocities, except perhaps the roll velocity, are considered to be small relative to some reference quantity. This is justified for a typical ship with a choice of coordinate origin near the roll center since the terms containing the roll velocity squared in Euler's nonlinear equations are small quantities. Euler's nonlinear equations written in a body fixed axis system then become the following linearized equations of motion:

$$X = m[\dot{u} + z_G \dot{q}] \qquad (1)$$

$$Y = m[\dot{v} + x_G \dot{r} - z_G \dot{p}] \qquad (2)$$

$$Z = m[\dot{w} - x_G \dot{q}] \qquad (3)$$

$$K = I_{44}\dot{p} - I_{64}\dot{r} - m z_G \dot{v} \qquad (4)$$

$$M = I_{55}\dot{q} + m(z_G \dot{u} - x_G \dot{w}) \qquad (5)$$

[1]Generalized forces and motions, are forces and moments, and translational and rotational motions (displacements, velocities, or accelerations) respectively.

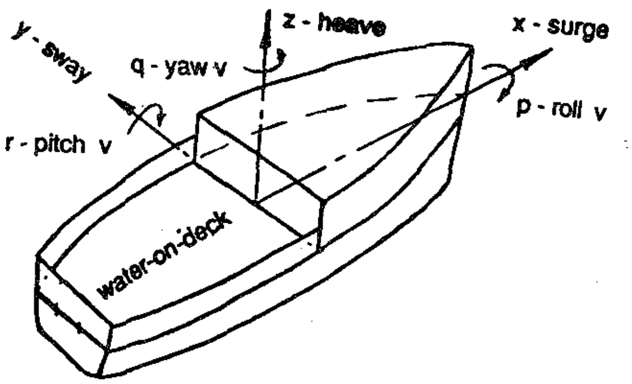

Figure 1: Ship coordinate system

$$N = I_{66}\dot{r} - I_{64}\dot{p} + m x_G \dot{v}. \qquad (6)$$

Considering the standard ship Euler angle rotation order (i.e., yaw, sway, and roll), and small rotation amplitudes for yaw and pitch, only roll requires special consideration. To first order, the rotation about a fixed or moving axis system is identical, so the Euler angle kinematics associated with finite rotations can be ignored conditional on roll being taken last.

The hydrostatics and hydrodynamics are included in the general force vector $\underline{X}$. These forces are obtained by integrating the pressure over the body surface. First order linear terms proportional to unit body motion (displacement, velocity and acceleration) and incident wave amplitude are considered separately. The force proportional to unit body displacement is the linear hydrostatic force **C**, the force proportional to unit body acceleration is the linear added mass **A**, and the force proportional to unit body velocity is linear damping **B**. The linear forces due to the incident wave, the wave exciting forces, are given as $\underline{F}(t) = \underline{F}_o e^{i\omega t}$. For this frequency domain representation, the matrices **A** and **B** are constant. The matrix **M** represents the physical mass or inertias about a specific axis plus the inertial and coordinate coupling. The subscripts refer to the mode of motion and are: 1=surge, 2=sway, 3=heave, 4=roll, 5=pitch and 6=yaw. See Figure 1.

In order to include important nonlinear roll effects, the linear hydrodynamics are supplemented with an empirically derived linear and quadratic viscous roll damping and a calculated nonlinear roll restoring moment curve. The viscous damping is calculated component-wise using the methods described by Himeno, (1981) and the nonlinear roll restoring moment curve is calculated using a ship hydrostatics computer program (InterCAD, (1983)). The calculation of the linear hydrodynamic forces is performed using a linear frequency domain ship motions computer program (Beck and Troesch, (1989)).

Considering only these forces, the frequency domain representation of the seakeeping equations including nonlinear roll forces, $\underline{g}(\underline{x})$ are obtained,

$$(\mathbf{M} + \mathbf{A})\ddot{\underline{x}} + \mathbf{B}\dot{\underline{x}} + \mathbf{C}\underline{x} = \underline{F}(t) + \underline{g}(\underline{x}). \qquad (7)$$

The equation system (7) is expressed in Cauchy standard form by multiplying through by the inverse of the mass and added mass matrix $(\mathbf{M} + \mathbf{A})^{-1}$ and defining $\underline{y} = \dot{\underline{x}}$:

$$\dot{\underline{y}} = -\mathbf{B}^*\underline{y} - \mathbf{C}^*\underline{x} + \underline{F}^*(t) + \underline{g}^*(\underline{x}) \quad (8)$$

$$\dot{\underline{x}} = \underline{y}. \quad (9)$$

where the matrices $\mathbf{B}^*$ and $\mathbf{C}^*$, and the vectors $\underline{F}^*(t)$ and $\underline{g}^*(\underline{x})$, have already been multiplied by the inverse of the mass plus added mass matrix.

In the linearized equations (equations 1-6), no inertial coupling occurs between the symmetric (surge, heave and pitch) and the asymmetric (sway, roll and yaw) modes of motion. Moreover, owing to port-starboard symmetry of the underwater body of a typical ship, there is no linear hydrodynamic coupling either. Consequently, for a typical ship the coupling between yaw, and sway and roll is small and for a proper choice of coordinates (i.e., a roll center) the coupling between sway and roll can also be minimized. The net result of the above systematic although heuristic procedure is to separate the roll equation of motion from the other five degrees of freedom. The only phase variables of interest are roll velocity and roll displacement. Equations (8) and (9) then represent a two dimensional[2], one degree of freedom system.

The techniques described in this paper are general enough for n-dimensional problems, though their interpretation to higher dimensions requires an understanding of the phase space topology. Consideration of these aspects will appear in a paper to be published. In a companion paper Falzarano, Stiendl, Troesch and Troger, (1990), consider the effect of the other asymmetric degrees of freedom (sway and yaw) on the nonlinear roll motion. In addition, they evaluate the effect of the frequency dependence of the hydrodynamic coefficients.

The Water-on-deck Approximation

Falzarano (1988) developed a simulation model that simultaneously solved the ship motion and water-on-deck problem. In that work, the ship motion was modeled using the linear time-domain with an approximate nonlinear model for the time-varying hydrostatics. The water-on-deck forces are determined by solving the shallow water hydraulics problem (Dillingham and Falzarano, (1986)) at each time step. As with any general simulation model, initial conditions and equation parameters were observed to have a significant effect on the eventual motion.

The present state-of-the-art in dynamical systems theory makes it impractical to analyze general nonlinear time-varying systems analytically. In order to model the water-on-deck force, it is necessary to solve a nonlinear system of partial differential equations at each time step. The water-on-deck force is a not a simple function of time. Therefore, the study of an analytically tractable explicit model exhibiting similar dynamics is desirable. Here explicit refers to a differential equation which is expressible in terms of simple functions of the phase variables and time.

[2]Dimension refers to the number of independent variables needed to uniquely specify the state of a dynamical system.

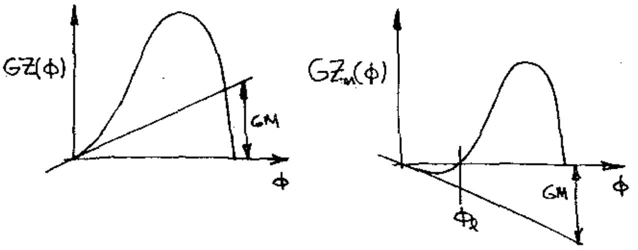

Figure 2: Roll restoring moment curve unmodified and modified

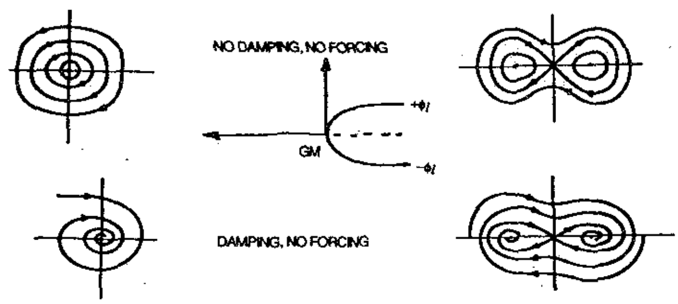

Figure 3: Pitchfork bifurcation diagram and phase portraits (with and without damping)

Caglayan, (1985) suggests that the dominant dynamics of the water-on-deck problem can be approximated by a fixed weight to achieve the same pseudo-static heel angle (loll angle). Following Caglayan's suggestion, a simplified model is formulated below.

The equation of motion describing the single degree of freedom vessel roll motion ϕ with water-on-deck is as follows,

$$\begin{aligned}(I_{44} + A_{44})\ddot{\phi} + B_{44}\dot{\phi} + B_{44q}\dot{\phi} \mid \dot{\phi} \mid + \\ \Delta GZ_m(\phi) = F_{sea}\cos(\omega t + \gamma_4).\end{aligned} \quad (10)$$

where I_{44} is the moment of inertia (in air) about the roll axis, A_{44} is the linear roll hydrodynamic added mass coefficient, B_{44} is the linear roll damping coefficient, B_{44q} is the quadratic viscous damping coefficient, Δ is the vessel displacement, and $GZ_m(\phi)$ is a polynomial approximation to the nonlinear roll restoring moment arm curve including both the intact stability curve $GZ(\phi)$ and the approximate water-on-deck effects. See Figure 2. The single frequency external wave exciting force has amplitude F_{sea} and phase angle γ_4 (with respect to wave crest amidships).

It can be shown by writing the above equation in Cauchy standard form $\dot{\underline{x}} = f(\underline{x}, t)$ and setting $\dot{\underline{x}} = 0$, that the equilibria for the unforced equation ($F_{sea} = 0$), correspond to roots of the function $GZ(\phi) = 0$. When the GM, the slope of the righting arm curve at the origin, is reduced through zero (from no water-on-deck to the static effect of water-on-deck), the stable upright equilibrium, $(\phi, \dot{\phi}) = (0,0)$, bifurcates into an unstable upright equilibrium and two stable equilibria at the positive and negative loll angles, $\pm\phi_l$. Locally, this bifurcation is the classical pitchfork bifurcation and is illustrated below in its bifurcation diagram and corresponding phase portraits (Figure 3).

For the bifurcated (as a result of the loll angle) system with one unstable and two stable equilibria (without damping or forcing), the phase portrait around

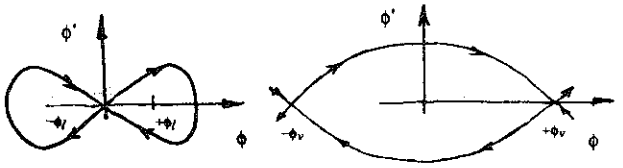

Figure 4: Phase portraits with a) homoclinic and b) heteroclinic connections

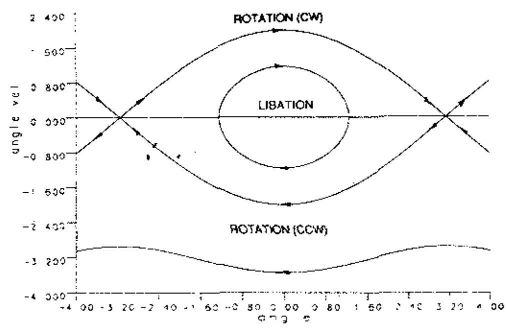

Figure 5: Phase portrait of undamped unforced pendulum

the loll angle will have two homoclinic orbits, each connecting the unstable saddle at the origin to itself (Figure 4a). A typical ship[3], with and without the water-on-deck approximation, will have an undamped unforced phase portrait with a heteroclinic connection connecting the saddles at the positive and negative angle of vanishing stability to one another and vice versa (Figure 4b).

Dynamical Systems Theory

Invariant manifolds

Since the energy is constant along an orbit, the orbits of a two-dimensional conservative *autonomous* (time independent) system are easily calculated and expressible in closed form by solving the energy equation for the velocity in terms of the displacement. A well-known example is the trajectories of the simple pendulum as illustrated in Figure 5. The undamped and unforced ship dynamics are similar except that the pendulum saddles at $\pm 180°$ are replaced by saddles at the positive and negative angles of vanishing stability (i.e., $\pm\phi_v$) for the ship.

Referring to Figure 5, the orbit connecting the two saddles is important since it separates the simple harmonic motion (inside) from the rotational motion (outside). This special curve is called a *separatrix.*

If the system is not conservative, the trajectories can still be determined but this is usually done numerically. Even with damping, these trajectories are still invariant with respect to time and need only be calculated once for all time. The separatrix of the undamped system is replaced by invariant manifolds; one set that originates from infinity and approaches the saddle; and another set that originates at the saddle and approaches the stable node. The manifold approaching the saddle is called its *stable manifold*

[3]The special case of self-righting lifeboats which have an angle of vanishing stability at $\phi_v = 180$ degrees is not considered.

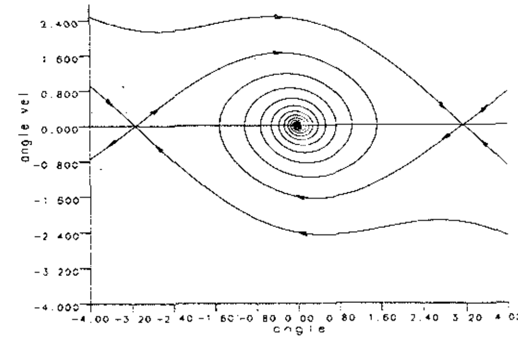

Figure 6: Phase portrait of unforced damped pendulum

and the manifold starting from the saddle its *unstable manifold* (Figure 6).

Periodically Time-varying Forces and Poincaré Maps

It is common when analyzing a *nonautonomous* (time dependent) system to define time as an additional phase variable and to study the dynamics in the resulting extended phase space. If the time-dependence of the differential equation is periodic, then the resulting dynamics may also be periodic. It is reasonable to convert this extended phase space dynamical system to a Poincaré map which samples states of the extended phase space once per period of the forcing.

Two important issues immediately arise - the selection of the phase at which to sample the dynamical system, and the behavior of non periodic response. The first issue is easily reconciled by knowing that for a periodically time-varying differential system any two Poincaré maps sampled at different phases are C^r conjugate (Wiggins, (1988)). Practically speaking, this means that Poincaré maps at different phases are qualitatively similar although they may have been rotated and deformed. The answer to the second issue is exactly why the Poincaré mapping technique is used to analyze periodically forced differential equations. The Poincaré mapping technique allows the presence and type of periodicity and possible lack of periodicity (e.g., the presence of aperiodic or chaotic response) of the response to be assessed.

The procedure for converting a n-dimensional non-autonomous differential equation,

$$\dot{\underline{x}} = \underline{f}(\underline{x}, t). \tag{11}$$

into a Poincaré map involves explicitly adding time as an additional autonomous phase variable to obtain an n+1-dimensional system (Wiggins, 1988).

If the time dependence is periodic, the the equation repeats itself every period of the forcing; so the system may be written as follows,

$$\dot{\underline{x}} = \underline{f}(\underline{x}, \theta) \text{ with } \underline{x} \in \mathbf{R}^n \tag{12}$$

$$\dot{\theta} = 1 \text{ with } \theta \in \mathbf{S}^1. \tag{13}$$

Therefore, the phase space is in the product space $(\mathbf{R}^n \times \mathbf{S}^1)$ of the n-dimensional euclidian space (i.e.,

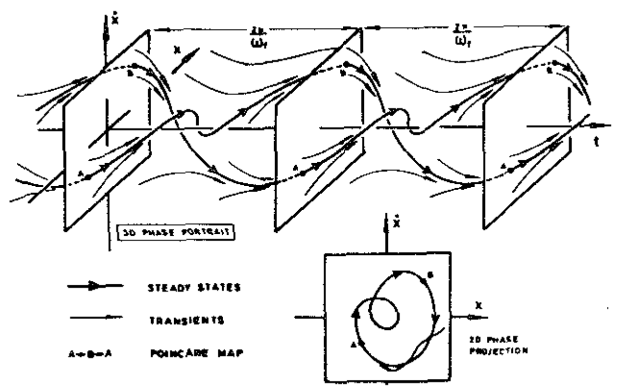

Figure 7: The Poincaré section in the extended phase space (Thompson and Stewart, (1986))

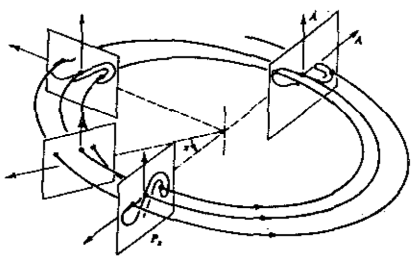

Figure 8: The Poincaré section in the cylindrical phase space (Moon, (1989))

$\mathbf{R}^n$) and the periodic time is a one torous (i.e., $\mathbf{S}^1$). The Poincaré cross-section $\sum$ is defined by the fixed time phase t_0, which is contained in the open interval from zero to T,

$$\sum = \{(\underline{x}, t) | t = t_0 \in (0, T]\}. \tag{14}$$

The map P of the cross-section $\sum$ onto itself (i.e., $\sum$) is defined as follows,

$$P : \sum \rightarrow \sum. \tag{15}$$

This is equivalent to a point $\underline{x}$ with time t_0 mapped to another point $\underline{x}$ with time $t_0 + T$,

$$(\underline{x}(t_0)) \rightarrow (\underline{x}(t_0 + T)). \tag{16}$$

The map so defined is called a Poincaré map or a stroboscopic sampling. One can consider the extended phase space $(\underline{x}, t) \in \mathbf{R}^{n+1}$ (Figure 7) or in order to visualize the periodicity one may consider the phase space to wrap around on itself $(\underline{x}, \theta) \in \mathbf{R}^n \times \mathbf{S}^1$ (Figure 8).

In this work, the map is never explicitly solved for; instead the differential equations are integrated for integer periods of the forcing $T = 2\pi/\omega$ (i.e., T, 2T, 3T...), to obtain successive iterates of the map.

Invariant Manifolds as Boundaries of Behavior

Invariant manifold analysis as described above is of practical use in that the manifolds separate distinct types of behavior. For the ship dynamical system, the stable invariant manifolds that originate from the angle of vanishing stability separate safe (non-capsizing) behavior from unsafe (capsizing) behavior. See Figure

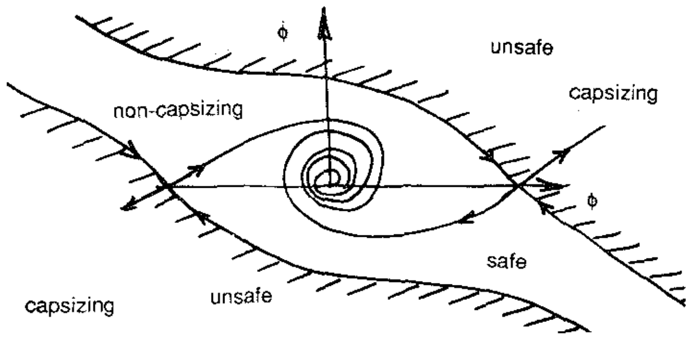

Figure 9: Boundary between safe and unsafe region

9. Specifically, the non-intersecting manifolds in the Poincaré map form a crisp basin boundary between the various oscillatory or growing steady-state solutions that can occur. Just as in the unforced equation, an initial condition $(\phi, \dot{\phi}, t = t_p)$ of roll displacement, roll velocity taken at some fixed time phase (recall time is mod (2π)), will never cross a nonintersecting invariant manifold *at that phase for all future time*[4].

Prediction of Intersections of Invariant Manifolds and Chaos

If the invariant manifolds do not intersect, Poincaré maps clearly show the eventual state of the motion of a given initial state. However, once the manifolds have intersected, the boundary between non-capsizing (bounded motion) and capsizing (unbounded motion) is no longer simple. Prior to the intersection of the manifolds, the stable manifolds of the saddles formed the safety boundary. Following intersection, the stable manifold moves inside the unstable manifold. The boundary between safety and danger is no longer just a line, but an intersected region. Outside this boundary is still unsafe while inside remains safe. However the eventual state of motion starting inside the region formed by the intersection requires further study, (Falzarano,(1990)).

It is clear that predicting under what conditions the manifolds intersect can be of great practical value. The intersection can be predicted in terms of the parameters of the system using Melnikov's method (Guckenheimer and Holmes, (1986)). Application of the Melnikov method begins with an unperturbed system (no damping, no forcing) where the trajectories are known for all time (Figure 10a). These trajectories are used to determine the characteristics of the perturbed system (damping and forcing). See Figure 10c. The unperturbed system is usually a time-invariant nonlinear system.

The Melnikov function for the heteroclinic intersections of equations (12) has been analytically determined (Falzarano, (1990)). The scaled differential equation and heteroclinic Melnikov function, $M(\tau_0)$, for a cubic approximation to the righting arm curve near the angle of vanishing stability are as follows,

$$\ddot{x} + \epsilon\delta\dot{x} + \epsilon\delta_q\dot{x} \mid \dot{x} \mid + x - kx^3 = \epsilon\gamma\cos(\omega\tau) \tag{17}$$

[4]The invariant manifolds are distorted and rotated as they evolve through one period and therefore successive *iterates* of the points will not cross the manifolds.

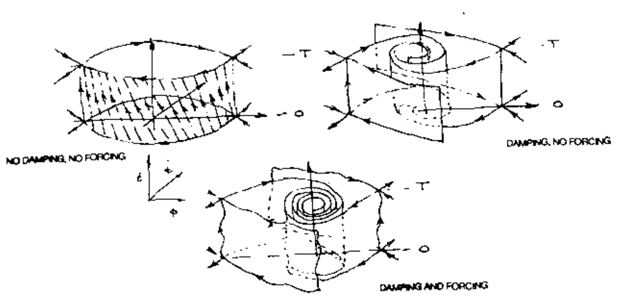

Figure 10: Invariant manifolds of a) unperturbed, b) damped, and c) damped forced system

$$M(\tau_0) = -\frac{\gamma\pi\omega\sqrt{2}}{\sqrt{k}}\text{csch}(\frac{\omega\pi}{\sqrt{2}})\cos(\omega\tau_0) + \frac{2\sqrt{2}\delta}{3k} + \frac{8\delta_q}{15k^{3/2}}. \quad (18)$$

The zeroes of the Melnikov function correspond to transverse intersections of the stable and unstable manifolds. The minimum required wave amplitude corresponds to setting $M(\tau_0) = 0$ and solving for gamma, the amplitude of the wave exciting force.

Although the manifold intersections can be predicted using Melnikov's method, more specialized techniques are required to analyze the resulting dynamics following intersections. An alternative approach used to analyze the erosion of the safe basin following intersections is provided by Thompson, (1990). In his paper, Thompson considers the manifolds intersections as the first of many important events which eventually result in the total erosion of the safe basin. Although Thompson is able to study the basin up to its final erosion, his technique requires a fine grid to be integrated for various parameter values. In lieu of integrating a grid of initial conditions Falzarano (1990) calculates the manifolds which represent the basin boundaries.

Application of Geometric Methods

In order to gain a clearer understanding of methods described in the previous sections, these techniques are applied to the twice capsized clam dredge *Patti-B*. The *Patti-B* was previously studied using traditional stability analysis, linearized stability analysis, and simulation. This analysis and vessel characteristics can be found in Falzarano, (1988).

The *Patti-B* has the dubious distinction of having capsized *twice*! Her first capsizing (NTSB, (1979)) occurred nearshore off of Ocean City, Maryland and she was salvaged. Approximately two years later, *Patti-B* was far offshore (USCG, (1979)) when she capsized again. *Patti-B*'s two capsizings are especially disturbing since she is typical of a large number of similar sized and designed fishing vessels, and apparently met all the then existing recommendations (Falzarano, (1988)). The fact that she had a disproportionately large after-deck prone to trapping water may have adversely contributed to her unfortunate safety record.

In this section, modern geometric methods for analyzing dynamical systems are used to analyze the clam dredge *Patti-B*'s capsizing mechanisms. As stated earlier, the effect of water-on-deck or damage is approximated by modifying the hydrostatic curve, the result which may or may not include a loll angle.

In order to demonstrate the importance of various design characteristics, the *Patti-B* with a reduced righting curve is studied as designed and with bilge keels fitted. Figure 11 shows the phase plane for *Patti-B* with a reduced righting arm curve (GM=.1197 ft) and the corresponding saddle trajectories or separatrices for the unperturbed system (no damping or forcing). Inside the unperturbed saddle trajectories are bounded simple harmonic motions and outside are unbounded rotations. This is the extent of the phase plane information that can be derived from solely considering the righting arm curve and with out including damping or forcing in the analysis.

Figures 12-14 show invariant manifolds (in the upper half plane) in the Poincaré map for the *Patti-B* for various dampings and regular wave amplitudes. Figure 12 is for the *Patti-B* without bilge keels (as she was designed) forced by a .45 radians/second (period 14 seconds) one foot high wave. Figure 13 shows how the manifolds are effected by the increased damping that would result from modest sized bilge keels being fitted onto the *Patti-B*. The stable manifold associated with the positive saddle moves outward and the unstable manifold associated with the negative saddle is pulled inward. Figure 14 shows how the Poincaré map changes as the same frequency (ω_o = .45 radians/second) wave is increased to a height of five feet. Both the stable and unstable manifolds expand outward. Since the outer stable manifold does not expand as much, the manifolds move closer. As the wave amplitude increases, they eventually intersect. Figure 15 shows simulations of two sets of two initial conditions on either side of the safety boundary. The two initial conditions inside the safe region eventually converge to the same bounded steady state solution (indicated by a circle in Figure 14). While the two inside the unsafe region quickly capsize. The upper unsafe initial condition capsizes to starboard (clockwise rotation), while the lower unsafe initial condition capsizes to port (counter-clockwise rotation).

Figure 16 shows the Melnikov prediction of the manifold intersections for the GM considered in the previous figures (GM=.1197 ft) with and without bilge keels and a slightly negative GM (GM=-.0985 ft) such as would result from a large amount of water trapped on deck and causing a loll angle (ϕ_l = .225 radians), (c.f., Figures 2, 3 and 4). The angle of vanishing stability is .675 radians for the loll angle case versus .81 radians for the positive GM case. Although the righting arm curve effects the required wave amplitude for manifold intersections, the damping is more important in this example. Further research is required to quantify the effect of intersecting manifolds on capsizing.

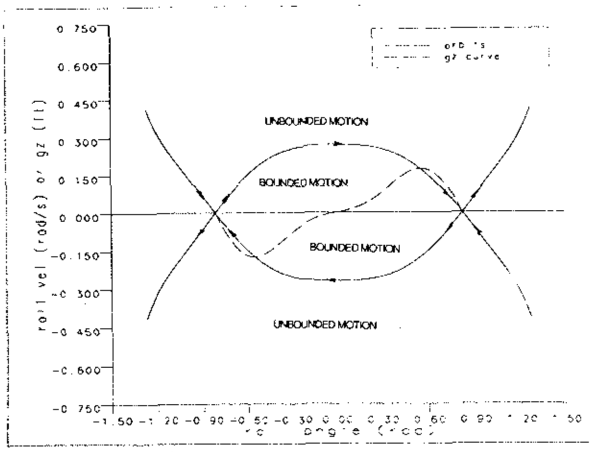

Figure 11: Righting arm curve considered and unperturbed orbits for *Patti-B*

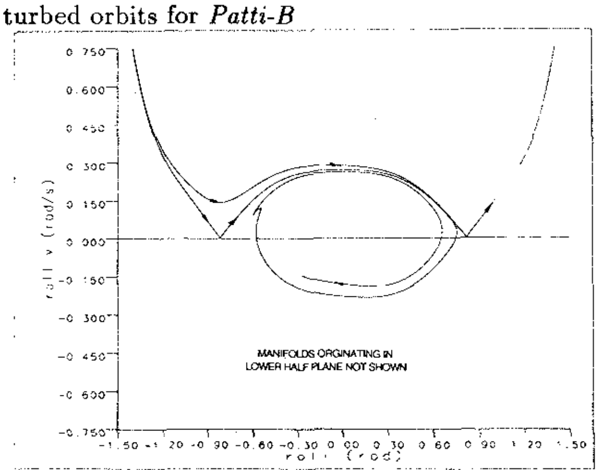

Figure 12: *Patti-B* with reduced righting arm curve and no bilge keels

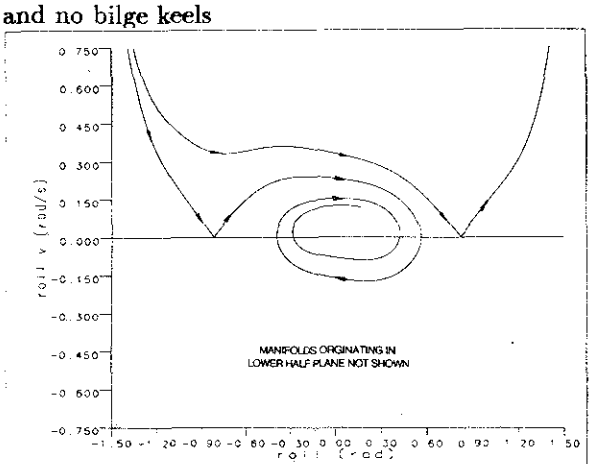

Figure 13: *Patti-B* with reduced righting arm curve and bilge keels

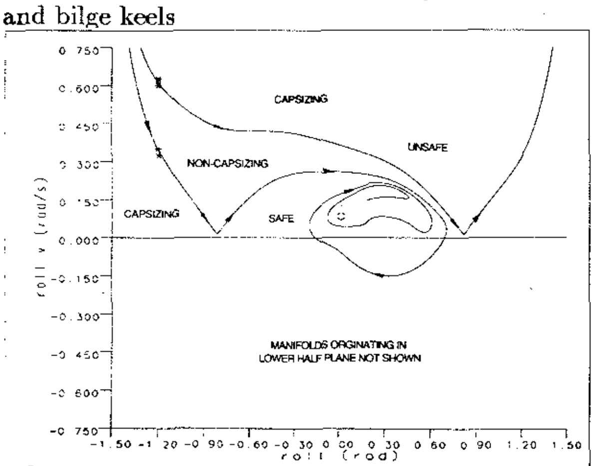

Figure 14: *Patti-B* with reduced righting arm curve and bilge keels

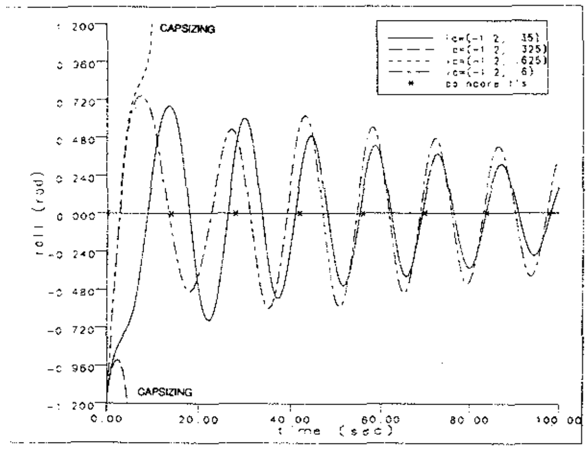

Figure 15: *Patti-B* simulations

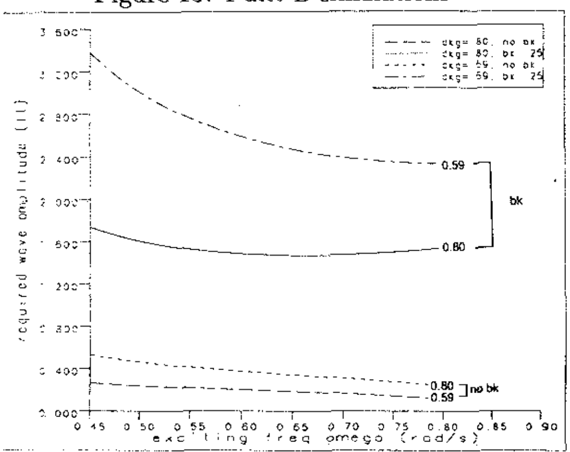

Figure 16: Melnikov Function for *Patti-B*

Conclusions

The techniques described in this paper should give greater insight into the effect of important ship and environmental characteristics on unbounded motion (capsizing) of the periodically forced nonlinear roll equation of motion. The important vessel characteristics include the nonlinear $GZ(\phi)$, as effected by water-on-deck, damage, icing, etc.; and damping, as effected by the size and presence of bilge keels. The most important environmental characteristic studied was the size and frequency of the regular wave forcing. Finally initial conditions due to non-steady-state effects (transients) are crucial and may eventually determine safety. In addition these techniques should provide guidance prior to doing simulation, so that crucial behavior will not be missed.

Acknowledgements

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FIRE AND STABILITY

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SUMMARY

Stability is one of the most important factor to define the assurance ratio of the vessels and to measure the ship's safety range.

Therefore, and it is essential, a ship should be designed with adequate stability and this main requisite should be kept in all operative conditions.

Fire is one other very feared accident that could compromise personnel and ship's safety. It could cause precarious stability conditions, due to fire fighting systems which give rise to accumulation of big quantities of water (sprinklers, utilization of hoses etc.).

This water could cause a swift loss of stability coming from the sudden increase of weight in the higher decks and by the result of free water surfaces.

The report shows accidents occurred on board some ships, and, recalling the rules actually in force, deals with technical subjects joining the solution of the problem.

Particularly, effects of fire fighting efforts made from the personnel of the vessel and from means of ashore, when the ship is mooring in the harbour, are considered.

INTRODUCTION

According to the last conception the "Level of safety at sea", more than a definition, is the result of several factors such as regulations and detailed rules, technical standars, established practices derived from common sense, social tradition, training, education and human factors. All these factors are motivated from different aspects of safety at sea.

In fire-fighting and stability, National and International Organizations such as National Government and I.M.O. (International Maritime Organization) have sought ever increasing standards of ship safety.

Fires on board continue to take their toll of human life. However, better fire safety will continue to improve through international agreement.

The common material used in fire-fighting efforts are CO2, foam, halon and water. The use of such materials is conditioned each by quantity and the toxic effects due to their chemical nature. For the water is the implicit risk related to the loss of stability and, at last, to damage the goods.

Up today, it doesn't result the existence of any rules controlling the use of water in fire-fighting

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efforts and the related stability risk.

The stability rules specify a minimum standard of stability for a vessel in any sea-going condition. The majority of vessels operate with an adequate reserve above this minimum so that if a fire-fighting emergency should occur there should be no immediate stability risk and the major efforts can then be exclusively directed to fire-fighting alone. Some other vessels, in particular smaller ones, operate with a stability standard not far in excess of the required minimum. In such cases stability considerations may well run concurrent with the emergency since the minimum standard required by the rules gives little room for any further deterioration. Moreover, not specific requirements of dewatering systems are recommended in upper decks. Furthermore, emergencies rarely occur at a convenient time and it is doubtful if a stability assessment later than departure from the previous port would be available. So, most decisions and assessments are likely to be made on the basis of personal expertise.

The paper presents risks that may occur with fire-fighting efforts that could limit and in some cases impair the full safety of the ship during the fire related to the stability. The risk that the ship will list because of water flooded during fire-fighting efforts is examined and discussed. Two examples of well-known fire on board are studied as validation of the risk and its related complications. Some recommendations and guidelines for the development of National and International standards are briefly outlined.

BACKGROUND

During the International Symposium "Fire Safety of Ship" held in Athens during May 1989, an high concern come out about the loss of stability. It was suggested that more studies about stability criteria during firefighting should be done and the ship's risks under such influence should be considered.

In this paper we refer to two salvage operations as the most relevent cases of firefighting in presence of water flooding.

M.S."SCANDINAVIAN SEA"

The passenger vessel Scandinavian Sea was destroiyed by a fire which began at sea off the Florida coast on March 9, 1984. The firefighting were temporarely suspended because the list exceeded 10 degrees.

The stability of the vessel during various time of the firefigthing efforts was calculeted and discussed.

There was no injuries or loss of life. The vessel was declared a constructive total loss. At the time was valued at 16 million USD.

Vessel information

Lenght overall	149 m
Beam	20 ,,
Draft	6.7 ,,
Gross tonnage mark submerged	10,736.48 tons
Gross tonnage mark not submerged	9,588.52 ,,
Depth to main deck	8.8 m
Deadweight in tons	3156/956

Three decks below the main deck were designated A,B and C deck, and four decks above the main deck were designated as the upper, lounge, boat and sun deck.

Firefighting began at sea with the crew using the firemain system. Water was introduced into deck "A". The vessel arrived at the pier with a list of one to two degrees. After the arrival, shoreside firefighting support introduced large amount of water also onto the boat deck, the lounge deck and the main deck in order to cool off all the risk areas. Little if any water was removed from the ship throughout these efforts. As fire-fighting continued, the vessel continued to list to starboard. The list was about 10.8 degrees at the conclusion of the effort there was speculation as to the capsizing risk to the vessel due to the introduction of firefighting water. The on-scene commanders attempted to assess the potential for capsizing but little information was available for making such as assessment. clearly the ability for an on-scene commander to determine the effects of firefighting efforts on the stability, at pierside, is desiderable.

CALCULATION AND OBSERVATION

The stability of the vessel during firefighting effort has been evaluated for the following conditions (see figure):

Condition A: Arrival condition according to the Chief Mate.

Condition B: about 526 T of free water on the lounge deck, accordingly to the report of the NTSB (National Transportation Safety Board, USA) investigators.

Condition C: about 1630 T of free water, this is an hypothetical condition to assess the effects of additional firefighting water.

From the official report of the U.S. Coast Guard it is evident that the vessel was in little danger of capsizing and the firefighting efforts were stopped due to excessive list. Moreover, there were some portlights on the same deck "A" wich were almost submerged at time of firefighting efforts were stopped. This could have been an additional potential source of flooding and they were closely monitored. The decision of stopping the firefighting efforts seems a wise one for the results of the calculation. Calculation show the angle of equilibrium to be 5.5 degrees. The maximum righting arms was about 110 cm at approximately 52.5 degrees. In addition, the available righting energy to this angle was about 48 cm degrees. The angle of vanishing stability was 87 degrees, leaving a residual range of stability of 81.5 degrees (figure, condition B). From the calculation seems evident that the vessel was not in immediate danger of capsizing, so long as the port lights mentioned remained intact. However the vessel would have listed further if had additional firefighting occurred without dewatering. The maximum righting arm was 58 cm at approximately 52.5 degrees. The available righting energy was about 18.5 cm degrees. The angle of vanishing stability was about 79 degrees, leaving a residual range of stability of about 50.5 degrees.

M/f "DELEDDA"

Vessel information:

Built in 1978 at Castellammare di Stabia Shipyards, Naples, Italy.

Length overall	130,95	m
Length by p.p.	118,00	,,
Beam at main deck	20,00	,,
Height at main deck at 1/2 L.b.p.	7,20	,,
Draft	5,56	,,
Displacement	7790	t

The vessel is equipped with a stern port to the main deck.

Fire began at pier-side and it was localized in the crew quarter zone, in the aft area of the upper deck, over the garage. The zone was equipped with Sprinklers that started to flood water as soon as the fire began. The Sprinklers flooded 2.2 t of water per minute over the full 370 mq firefighting zone.

Before the firefighting efforts began the vessel had the following stability parameters:

Displacement	6.978 t
Draft	5,17 m
Coordinates of center of displacement	2,96 ,,
Radius of transv. metacenter	6,90 ,,
Height of transverse metacenter	1,55 ,,

Before the firefighting efforts, the vessel had even transverse trim condition and she didn't have a list.

In addition, shoreside firefighting support introduce large amount of water.

At the time the fire was extinguished, about 720 t of free water resulted onboard. The water center of gravity was at 5,52 m from the construction line and 31,79 m from the aft perpendicular (such distribution is presented in tab.1). The vessel had a list of only 11.5 degrees because limited by the constraints of one anchor and 8 moorings ropes. Because the Deledda developed a list during the firefighting efforts, a study was made of the vessel's stability condition at the maximum observed list. The liquid loading of the vessel, including fuel oil, fresh water and ballast water, combined with the measuerd amounts of water trapped in the compartments were applied to the hydrostatic proprieties of the vessel. From the study, done for the ship-owning company, the results indicated that the vessel without constraints had have a list of about 27,5 degrees.

After flooding, the following elements were calculated:

- the displacement increased of 720 t and finally was 7698 t
- the draft increased of 0,38 m
- the center of gravity was lower of 0,27 m
- the radius of transverse meta-center became 5.36 m
- height of the transverse meta-center decreased to 0.51 m
- the trim was 1.41 m.

From this calculation, the following conclusion are drawn:

1) when the vessel arrived at a list of 16 degrees, the right side low angle of the stern port could touch the sea level. In such an event, additional free water could be flooding the main deck. Because of a wider free water surface, the height of the transverse metacenter could have been decreased with a further reduction of the right arm.

2) the fire leftover closed some de-watering system increasing the water accumulation.

3) because of the emergency of the de-watering the decks some on-scene holes were realized on the hull,

4) without the external firefighting efforts, the only sprinklers' system

would flood the decks with 70 t per 30 minutes. Such situation would generate a list of 25 degrees at free vessel.

5) in the case that one mooring rope would have been broken, the others wouldn't have been resisted to the resulting dynamic action. The resulting dynamic energy of such event could have broken the other mooring ropes creating serious concerns on the safety of the vessel.

Moreover, the calculations indicated that the ability of the vessel to right itself after external heeling force has been applied would have been reduced of a great percentage from the condition of the vessel before the firefighting efforts.

ON BOARD AND IN HARBOUR

It is wide known that the fire is one of the most dangerous and surely the most visible risk on board.

Master and crew have to rely completely on their professionality when facing a fire on board. The professional expertise of mariners in case of fire is done of well codified procedures and interventions. Among several procedures the most used ones are the methods of limiting and extinguishing the fire. Quite often in a fire emergency, outside halon, CO_2 or foam self-protected areas, the water is largely used in the fire-fighting efforts. In case of large flooded water without dewatering, particularly in upper decks, the ship lists because of the lateral movement the free water surface. After the ship is listed, a risk for the vessel to lyes on her beam ends exists. If this risky situation occurs the capsizing of the vessel may happen. On ferry-boats such risks can be generated or amplified by the movements of the cargo. The cargo is static when the ship trim condition are regular, but it may become easily dynamic under a strong variation of transverse trim that will generate an inclining momentum sum of all the weights and liquids and eventually wind and waves. Moreover, a very recent paper support the concept of loss of stability due to water on deck. Some papers, show even if only for one ship model, that in the case of accumulated water on upper decks generates risk of capsizing in waves.

From the usual stability tests, the Public Administration gets the fundamental data to inform the Master "to give the means to evaluate the stability characteristics in all the working conditions with an easy procedure". In fact SOLAS 74 (83) Chapter II-1/Part B Reg.22.1 and Reg.22.2 states that "Every passenger ship regardless of size and every cargo ship having a lenght, as defined in the International Convention on Load Lines in force, of 24 m and upwards, shall be inclined upon its completion and the elements of its stability determined. The master shall be supplied with such information satisfactory the Administration as is necessary to enable him by rapid and simple processes to obtain accurate guidance as to the stability of the ship under varying conditions of service." As it is easy to see the Regulation is referred only to the stability of the ship under varying conditions of service. On this base some ships now have computer programmes for routine stability, some not being so convenient for non routine use.

The argument is also very important to the Port Authority for the duties and responsabilities that he has. In Italy, such duties and responsabilities are ruled by the National Law of May 13, 1940, n.690 "Organizzazione e funzionamento del servizio antincendio nei porti - Organization and Management of fire fight service in harbors." The Port Authority has the responsibility to control and to direct the fire fight operations of merchant ships in harbours.

A fire event will force the Port Authority to be knowledgeable of the general ship conditions at the time of the emergency, including stability and trim for different working conditions and the fire-fighting efforts risk of capsizing because of free water on upper decks. In this respect it is evident that the solution of the problem should have to be the result of International Agreement.

REMARKS

Some years ago in Naples, during the transformation of two Italian cargo ferry ships to passengers ferry boats the problem of accumulated water on upper decks due to fire-fighting efforts was conceived. Concern on the risk to reducing the righting arms, due to fire-fighting efforts, with catastrofic effects on the surviving stability of the ship were established. In that occasion, the study of national and international rules, technical standards, etc., concluded that the rules, etc., did not include neither specific requirements for decks upper the bulkhead deck neither any requirements of dewatering systems.

In case of a fire, the sprinklers would easily flood the upper decks during the emergency. In fact the SOLAS 74 (83), Chapter II-2/Part A Reg.4, states that "Every ship shall be provided with fire pumps, fire mains, hydrants and hoses complying as applicable with the requirements of this regulation. The required fire pumps shall be capable of delivering for fire-fighting purpose a quantity of water, at pressure specified in paragraph 4, as follows: pumps in passengers ships, not less than two thirds of the quantity required to be dealt with by the bilge pumps when employed for bilge pumping."

The water flooding of upper decks will directly reduce the height of transverse metacentre. Such reduction will reduce the righting arm that will effect the ship stability and the overall safety of the ship. Such concern is obvious from the previous calculation where it is evident that the free surface of the water or any other liquid used on upper decks will reduce the righting energy of the vessel. The fire aboard the M.S. Scandinavian Sea on March 9, 1984 has been taken as an example of our concern and it is a good validation of the risk. In addition, from the M.S. Scandinavian Sea and Deledda calculation come out that also in optimal sailing condition it is necessary only a surface of water, of few centimeters high, function of the size and shape of the vessel, in a large area without dewatering systems and watertight doors, such as restaurant, lounge, bar, etc, to reduce the transverse righting energy of the vessel.

The installation of dewatering systems such as scappers, on decks above the bulkhead deck, is required just as additional provisions applicable only to special category spaces, such as garages, by Reg.37.2.2.1 of SOLAS 74 (83) Chapter II-2/Part B. The above Regulation states that " In view of the serious loss of stability which could arise due to large quantities of water accumulating on the deck or decks consequent on the operation of the fixed pressure water-spraying system, scuppers shall be fitted so as to ensure that such water is rapidly discharged directly overboard."

CONCLUSIONS

Although the free surface for a given compartments is indipendent of the volume of water in it, the heel angle is not. The limiting factor for heel in this case (as in most passenger ships) is the downflooding which occurs when the "water interface" moves transverserly across a stairway.

Penetration in superstructure envelope, mode during the course of firefighting to allow water to drain off the vessel, could minimize stability problems resulting from firefighting efforts. This could significantly reduce the heeling moments.

Some recommendations, could be drawn to minimize the degradation of stability as the efforts progresses. If there is a choice, the fire should be fought as low in the ship as possible. This as the effects of lowering the center of gravity or at least minimizing the rise. The extent of longitudinal flooding should be limited as much as practicable. The degradation of stability due to free water depends on several factors. Among them the lenght of the flooded compartments, the angle of heel of the vessel, the height of the compartments above base line and the percentage to which the compartments is full (i.e. 10%, 20%, etc.). This percentage effects the equilibrium heel but not the free surface correction. The rules don't include the calculation of the transverse metacentric height after flooded water in upper deck closed area. It seems that the rules don't give to the Master the knowledge of the ship limit influencing the ship trim and stability in non routine conditions. However, such limit should be recalculated because on it there is a great influence of the safety of the ship and of the people. The use of computer programmes including non routine conditions could be very useful.

Scuppers could be realized easily in some cases. Their application could be required on passenger ships in upper decks with wide areas such as restaurants, lounges ,bars, discos, casinos, etc. Such solution can apparently seem simple, however it would not be so in the case of service and accommodation areas. In any case a supplementary rule should control the application in different ship types and cases.

The water volume of the fire pumps is not related to the necessary amount of water to extinguish the fire but is related to the volume rate of bilge pumps.

It is under study, in Sub-Committee on Fire Protection the eventual relationship between the fire rising in a zone and the necessary amount of water to extinguish it.

We hope in the attention that naval architects, masters and Maritime Organizations will pay on the subject showed in this paper for limiting the effect of free water or for a previous assessment of stability and heeling effect owing to fire-fighting.

ACKNOWLEDGMENTS

We would like to thank Dr. C.G Biancardi for his helpful cooperation.

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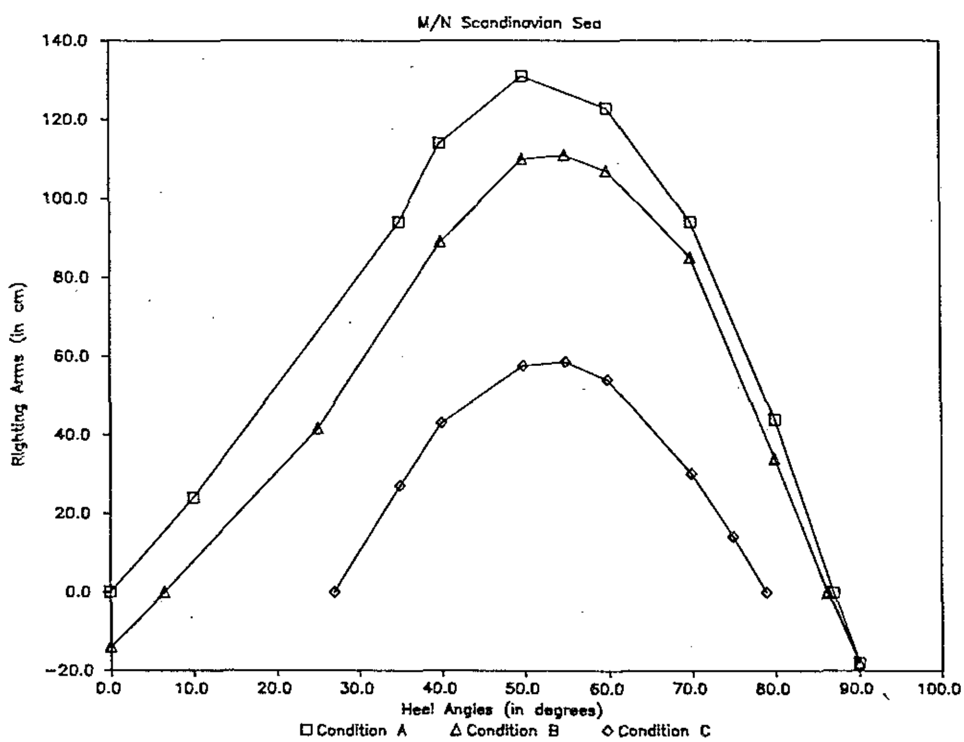

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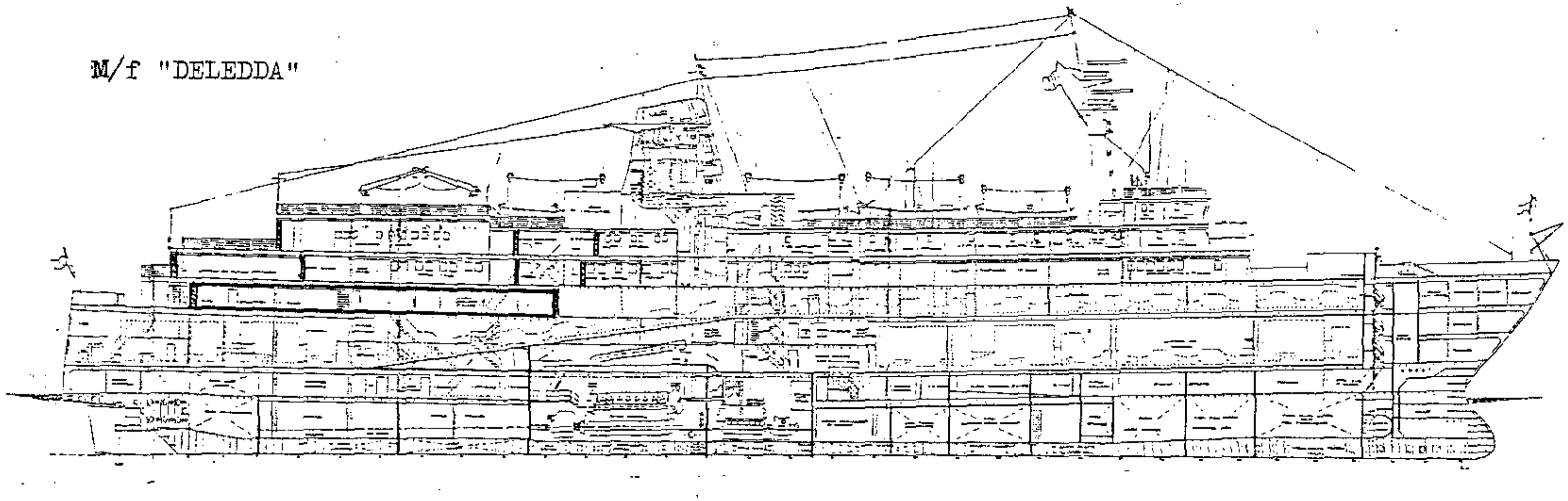

WATERTIGHT BULKHEAD FIREPROOF BULKHEAD CREW ACCOMODATIONS - FIRE AREA

N	LOCATION OF THE WATER ADDED ON BOARD	Q (t)	Yg (m)	Zg (m)	l (m)	b (m)	h (m)	l * b**3 (m**4)	Xg (m)	Q * Xg (t * m)
1	MAIN DECK	40	25	7.5	40	3.2	0.32	1300	9.0	360
2	UPPER DECK	70	17	12.5	35	4.5	0.45	3200	8.5	595
3	LOWER DECK	80	81	5.5	11	8.5	1.12	6700	4.3	344
4	STORE ROOM	150	32	5.7	8	10.0	1.87	8000	5.0	750
5	CREW'S DINING ROOM	180	19	5.9	12	17.5	0.85	65000	0.0	0.00
6	AUXILIARY ROOM	200	30	2.2	11	13.5	1.35	27000	0.0	0.00
	TOTAL	720						111200		2050

Q : weight of water added
Xg; Yg; Zg : position of center gravity of the water added
l; b; h : dimensions (lenght, beam, height) of the water added

SENSITIVITY OF SHIP MOTION PREDICTIONS TO WAVE CLIMATE DESCRIPTIONS

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ABSTRACT

Long-term predictions of most probable maximum values of motion amplitudes are required for design purposes, so as to guarantee adequade living standards onboard and the safety of the ships against capsizing. This work will present the current methodology of deriving the long-term probability distributions of ship responses and the different sources of wave data. Long-term distributions are determined for different ship types and for different wave climate descriptions so as to determine the sensitivity of the results to these variations. It is shown that the uncertainty involved in the statistical description of one specific area is larger than the differences between ocean areas.

1. INTRODUCTION

In ship design, it is becoming a routine procedure to account for the ship performance in terms of the seakeeping capabilities or of the motions in general, in choosing between competing designs. Linear strip theory has shown to give adequate predictions of the wave induced motions in the vertical plane, which are the governing ones for the assessment of the seakeeping performance.

In order to have reference values to use as design targets, it is necessary to predict the most probable maximum values of motion parameters such as accelerations, displacements or relative motions which are likely to occur in a period of the order of the ship's lifetime. These design values are obtained by determining long-term probability distributions, which depend on one side, on the transfer functions that predict the amplitude of the wave induced motions and, on the other, on the wave spectra and wave climate descriptions.

When estimating design values of any parameter there is always a degree of uncertainty associated with the prediction of events that are expected to occur sometime in the future. In general, conservative values are used for those parameters and the degree of conservatism, reflected by the safety factor adopted, will depend on the level of uncertainty in the knowledge of those parameters.

While in the past the safety factor to apply to any prediction of design value was largely a subjective matter, nowadays probability based procedures can be used to quantify the uncertainty associated with each prediction and to determine the corresponding safety factor. These probability based procedures have been developed primarily in the field of structural analysis and design but they are equally applicable to any other design variable as for example the ones associated with the ship motions.

The heave and pitch are motions that have a large effect on the living conditions on board and thus on the ship operability. The knowledge of the probability distribution of their amplitudes during the ship's lifetime will provide information about the percentage of time that operational conditions will prevail.

Rolling also influences the quality of the living conditions but, furthermore, it is also related to the safety of the ship with respect to capsizing. The safety problems involved with ship rolling and capsizing require a non-linear theory to describe them. The predictions of the linear theory are not accurate enough to predict capsizing, but one can always use the results of strip theory to make long-term predictions of the rolling motions which are indicative of the susceptibility to capsizing.

Work has already been reported on the similar problem of wave induced load effects, which has accounted for the uncertainty in the short-term situations both due to the uncertainty in the shape of the wave spectra [1], due to the uncertainty in the transfer function predictions [2], and due to the voluntary maneuvering actions in heavy weather [3]. Different uncertainty sources in long-term predictions have already been examined, in particular, the effect of using different sources of wave climate data based on visual observations and on wave measurements [4] and more recently on hindcast data also [5].

The uncertainty in wave induced load effects, which was based on calculations reported in [6], is significantly larger than one would expect in the case of heave and pitch motions. However, even so, it has been shown that the uncertainty resulting from the wave climate description is higher than that one. For these motions one can consider that the uncertainty results only from the wave climate description, and this paper deals exactly with the quantification of the uncertainty in the long-term predictions of the wave induced ship motions as a result from the uncertainty in the wave climate descriptions.

The transfer functions for these motions have a different shape than the ones for the load effects, in particular their relation with the average period of the sea state is specially relevant. Thus, the conclusions obtained in [4] and [5] for the sensitivity of the load effect predictions are not directly applicable to motion predictions, which motivated the work reported herein.

2. LONG-TERM PROBABILISTIC MODELS OF WAVE INDUCED MOTIONS

The long-term models are built from the short-term sea conditions expected to occur during the ship's lifetime which are weighted in an appropriate manner. The short-term description of the sea state and of the ship motions are based on a mathematical formulation which describes the probabilistic properties of the process under study. On the other hand the long-term variation of the parameters that govern the load effects to be experienced is empirical and thus, different types of probability distribution must be tested so as to choose the most appropriate one to the case under consideration.

In the short-term situation one models the sea surface elevation as a stationary Gaussian stochastic process. This implies that the amplitudes of the wave elevation are described by a Rayleigh distribution, in which case the probability Q_S of exceeding the amplitude x in a sea state of variance R is given by [7]:

$$Q_S(x \mid R) = \exp - \frac{x^2}{2R} \qquad (1)$$

The variance R of the sea surface elevation can be determined from the spectrum $S_H(\omega)$ of the wave elevation h. It is in fact its zeroth moment:

$$R = \int_0^{\infty} S_H(\omega)\, d\omega \qquad (2)$$

which is obtained by integrating the spectrum over the frequency ω.

In theory the Rayleigh distribution is only applicable to narrow band processes, although it has already been shown to be applicable to moderately wide band situations. However evidence has also been put forward about the lack of fit of the Rayleigh distribution in the low probability tail in wide band processes. In reexamining the situation, Longuet-Higgins [8] has shown that equation (1) can still be used provided that the variance as determined by expression (2) is afected by an appropriate correction factor.

The wave spectra are described by theoretical models that have become well established. In fully developed sea states one can use the ISSC parameterisation of the Pierson-Moskowitz spectrum [9], while for developing sea states the ISSC parameterisation of the JONSWAP spectrum [10] is the appropriate one. In combined sea states a double peaked spectrum with four parameters can be used [11].

The effect of adopting the different spectral models to calculate the wave induced load effects was studied in [1] where it was shown that, while the short term responses are sensitive to the type of spectral model used, the long term predictions could be accurately determined using only the Pierson-Moskowitz model in the calculations. Thus, only this type of spectral model will be used in the long-term calculations reported here.

The linear response to the wave spectrum is most conveniently determined in the frequency domain as the product of the square of the transfer function $H(\omega)$ by the wave spectrum, where the effects of the directional spreading $D(\theta)$ of wave energy in the sea state can also be accounted for.

The transfer function will depend on the direction of the wave system as well as on the wave frequency. Thus the variance of the response for a given relative ship heading is obtained by integrating both over the frequency and over the various relative headings:

$$R(\overline{\theta}) = \int_0^{\infty} \int_{\overline{\theta}-\pi/2}^{\overline{\theta}+\pi/2} S_H(\omega)\, H^2(\omega, \overline{\theta}+\theta)\, D(\theta)\, d\theta\, d\omega \quad (8)$$

where $\overline{\theta}$ is the main wave direction relative to the ship heading, θ is the relative direction of the wave system relative to the main direction, D is the directionality function and the response variance $R(\overline{\theta})$ concerns the main wave direction $\overline{\theta}$.

Different functions have been proposed for the spreading function but a common one is a cosine type:

$$D(\theta) = k_n \cos^n(\theta), \; -\frac{\pi}{2} \leq \theta \leq \frac{\pi}{2} \qquad (4a)$$

where k_n is a normalising constant:

$$k_n = \left(\int_{-\pi/2}^{\pi/2} \cos^n\theta\, d\theta \right)^{-1} \qquad (4b)$$

It is common to adopt a value of n=2 to represent sea states with significant spreading, which tend to occur for low values of significant wave height. A unidirectional wave system can be represented by adopting large values of n, up to 20. Most sea states can be modelled using intermediate values of the exponent n.

Since the response is linear with respect to the wave excitation, it has the same probabilistic properties as the surface elevation process i.e., it can be modelled as a stationary stochastic process. Thus, the probability of exceedance of an amplitude can also be described by equation (1) where the variance R of the motion must now be used.

The response process is in general more narrow banded than the excitation because the transfer function acts as a sort of a filter. This implies that the Rayleigh distribution is generally better applied to ship motions than to the sea surface elevation.

To extend this short-term formulation to the long-term situation one must recognise that the former results are conditional on a sea state spectrum defined by a significant wave height H_s and by an average period T_0, as is implied in the value of the variance R given by expression (3).

Considering the whole lifetime of the structure, the value of the variance of the motion amplitude at a random point in time can be described by a probability density function $f_R(r)$. Thus the probability Q_L of exceeding an amplitude x at a random point in time during the structure's lifetime can be obtained by unconditioning the short-term probability of exceedance:

$$Q_L(x) = \int_0^{\infty} w(T_o) \cdot Q_S(x|R)\, f_R(r)\, dr \quad (5)$$

where $w(T_o)$ is a weighting factor that is a function of the average period of the sea state, and which accounts for the different number of amplitudes that occur in sea states with the same duration but with different average periods.

The probability density function of the variance of the motion amplitude must account for the different load conditions, headings, speeds and even the voluntary changes of speed and heading that are performed under heavy seas. Different forms of this function have been proposed in the literature but probably the most complete one can be found in [4] as:

$$\begin{aligned} f_R(r) &= f(\theta, V, T_o, H_s, C) = \\ &= f_\Theta(\theta|H_s)\, f_D(\theta|H_s)\, f_M(\theta|H_s) \cdot \\ &\cdot f_M(v|H_s)\, f_{H_s,T_o}(h,t)\, f_C(c) \qquad (6) \end{aligned}$$

where f_Θ is the probability density function of relative headings between the ship and the waves, which is usually assumed to be uniform, f_D reflects the directionality of the wave climate, f_M models the effect of maneuvering in heavy weather [3] both on the ship heading and speed V, f_{H_s,T_o} is the joint probability density function of significant wave heights and average periods and f_C is the probability density function of ship cargo condition. This latter can be an homogeneous distribution as in the case of containerships or an heterogeneous one for bulk carriers and tankers which tend to operate in ballast and loaded condition [12].

A return period larger than the ship's lifetime is required for an acceptable level of probability, as discussed for example in [13]. The periods commonly used in offshore practice are 100 or 50 years but for ships it has been common to consider the characteristic value at the 10^{-8} probability level, which corresponds to the number of wave cycles to be expected in a period of 20 years of operation of a ship without stoping.

3. WAVE CLIMATE DESCRIPTIONS

The long term model of the wave climate is made up of two different time scales, as already indicated in the previous section. The short-term description is given by the wave height spectrum and the long term model indicates how the spectral parameters vary in a large time scale. The long term model is empirical in nature and is built from measurements or from visual observations in the different ocean areas to which it applies.

The first type of ocean wave statistics that became available was based on visual observations of the waves which were performed either in stationary Ocean Weather Stations [14] or in transiting ships that were reporting observations on a voluntary basis [15,16]. This type of observations cover large ocean areas like the North Atlantic [14] or even worldwide [15,16]. Another type of wave statistics result from measurements made with buoys of the waverider type. However they are more localised in coastal waters [17,18].

The visual observations of the wave properties are less precise and have a larger variability than the measurements. However because there are very many observations acumulated it is possible to obtain good estimates of the mean values of the observed conditions and the dispersion tends to decrease. This has been studied in [19] and [20] for the observations of the wave height and period respectively. In addition to reviewing all the previous work on the subject, those papers proposed calibration equations that are based on regression studies which correct the visually observed values to yield the spectral parameters that would be measured by a waverider buoy. It was found that the observations in Ocean Weather Stations had different characteristics from the ones of transiting ships, which led to propose different calibration expressions.

A different approach was adopted by Bales et al [21], who used hindcast models to predict the evolution of the wave spectra in the North Atlantic, based on information of the wind fields. Using the spectral parameters that were obtained at regular time intervals, a statistical compilation of the wave parameters was produced [21] as an alternative to the existing sources of information.

The long term distribution of the wave height and average period from these various wave data sources were compared in [5]. It was shown that the probability density function of significant wave height that is obtained from the data sets of Hogben et al [16] and Bales et al [21] agree reasonably well. The distribution of Walden's height data [14] has a larger percentage of waves of low significant wave height, around 3m, and a smaller frequency in waves of 6m.

A similar situation occurs for the average period data. The two most recent wave data sets [16, 21] agree relatively well but show a tendency to larger periods than the existing data sets. However the agreement of the probability distribution functions is better for the wave heights than for the mean periods. In fact, the long term average of the mean wave periods differs by about 2 seconds in the two sets of data of Hogben et al [16] and Bales et al [21].

When comparing the four different data sets one must keep in mind that the data of Hogben et al [15,16], being obtained from transiting ships, has inbuilt the effect of the bad weather avoidance, and tends to have a smaller percentage of high seas than one would expect in observations at fixed locations like in the one of Walden [14]. The statistics of the hindcast predictions of Bales et al [21] are based only on meteorologic data and thus should be directly comparable with Walden's set. When this comparison is made a not so good agreement is apparent.

The North Atlantic wave climate gives a good basis for comparisons between the different wave data sources. However, often ships have other predominant routes. To be able to determine the corresponding wave climate one must use the data of Hogben et al since it is the only one that covers the world ocean areas.

A set of typical ship routes have been considered in this work. The wave data corresponding

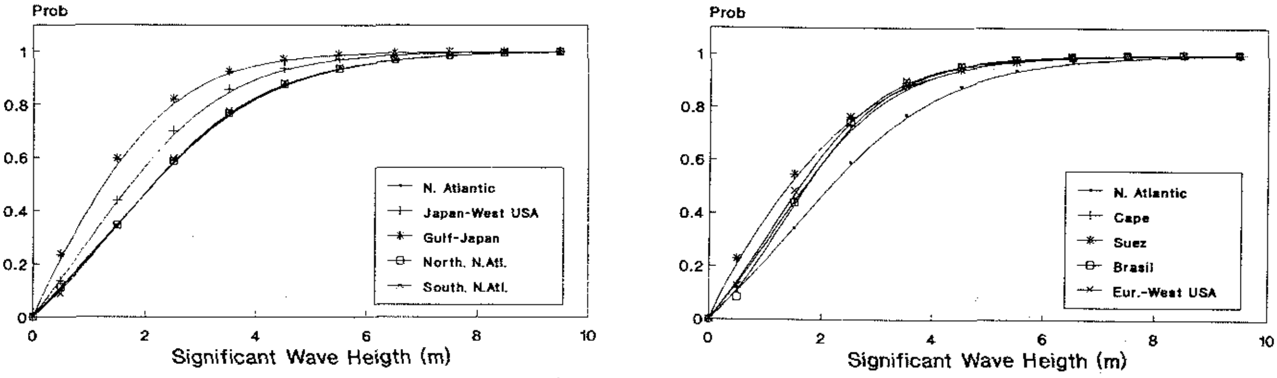

Figure 1- Probability distribution function of significant wave heights in various shipping routes.

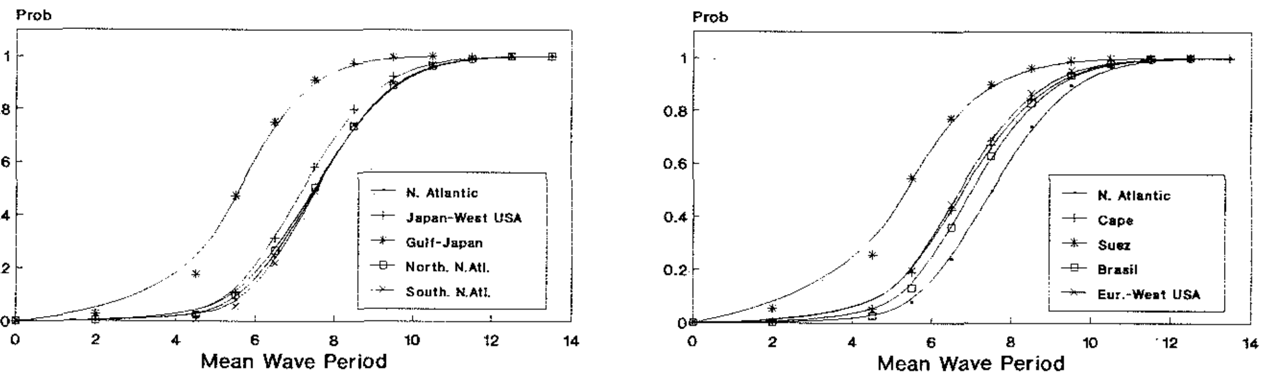

Figure 2- Probability distribution function of mean wave periods in various shipping routes.

to each route has been obtained by combining the data from the various ocean areas that are crossed by the ship, weighting each data set by a factor proportional to the duration of the ship's exposure to that weather. The combination of the wave statistics was performed on the basis of the probability distribution function so as to avoid the effect of unequal number of observations in the different ocean areas.

Consideration has been given to routes of Northern and Southern North Atlantic, from the Persian Gulf to the West Coast of the U.S.A. and, from that coast to Japan, as indicated in figure 1.

Furthermore, the North Atlantic wave climate is compared with the one of the routes of Europe to the Persian Gulf through the Cape and through the Suez Channel, from Europe to Brasil and from Europe to the West Coast of USA, through the Channel of Panama. These set of routes, which had already been used in [22], covers the main commercial trade routes. Inspection of figures 1 and 2 indicates that no major difference exists between Northern and Southern North Atlantic but the North Atlantic weather is worst than any other route.

The variability of mean wave periods is larger than the one of significant wave height but in general no major differences are apparent except in the routes of the Suez and from the Gulf to Japan which show a lower mean period.

4. ANALYSIS OF NON-LINEAR SHIP ROLLING

The approach presented in section 2 allows long-term predictions of linear ship motions to be made. However, rolling motions often extend into the range of large angles to which only non-linear theories are suitable. Different approaches have been adopted to study these non-linear motions, but they can be categorised in equivalent linearisation techniques, perturbation methods and applications of the Fokker-Planck equation.

The methods of equivalent linearisation replace the non-linearity of the equation by a suitable linear term which minimizes the differences between the non-linear motion and the predictions of the linearised equation. Vassilopoulos [23] developed a method based on linearisation techniques for the case of non-linear damping and restoring force:

$$\ddot{\phi} + 2\xi\, \omega_{\phi} \,(\dot{\phi} + \beta \dot{\phi}\, |\dot{\phi}|) + \omega_{\phi}^{2}\, (\phi + \gamma \phi^{3}) = M_{\phi} \qquad (7)$$

where ϕ is the roll angle, ξ the damping coefficient, ω_ϕ the roll undamped natural frequency, β and γ are coefficients governing the degree of non-linearity and M_ϕ is an inclining moment. He showed that the standard deviation of the non-linear roll angles $\sigma_{\phi N}$ is given by:

$$\sigma_{\phi N} = \frac{-\sqrt{2\omega_{\phi}\, \beta\, \sigma_{\phi_L}^{2}/\pi}}{1 + 3\gamma\, \sigma_{\phi_L}^{2}} + \frac{\sqrt{\left[(2\beta^{2}\, \omega_{\phi}^{2}/\pi) + 3\gamma\right] \sigma_{\phi_L}^{4} + \sigma_{\phi_L}^{2}}}{1 + 3\gamma\, \sigma_{\phi_L}^{2}} \qquad (8)$$

where $\sigma_{\phi L}$ is the standard deviation of the linear roll predictions i.e., the square root of eqn. (3). This expression can be used for long term predictions of non-linear roll amplitudes as predicted by eqn. (5), where now R is given by the square of eqn. (8).

Another method that yields a simple expression for $\sigma_{\phi N}$ which can be combined with the long-term formulation presented here, is based on perturbation methods. For a ship with non-linear restoring force or with non-linear but small damping, the variance of the non-linear roll response is given by [24, 25]:

$$\sigma^2_{\phi N} = \sigma^2_{\phi L} - 3\,\epsilon\,\omega^2_{\phi}\,\sigma^4_{\phi L} \qquad (9)$$

where ϵ is a parameter representing the non-linearity of the restoring or the damping term, depending on the situation.

The Fokker-Planck approach adopted by some authors [26] have been shown to give good results in some situations but the formulation did not result in such simple expressions as eqns. (8) and (9) which are easily adapted to this long-term formulation. However, an alternative approach to long-term predictions has been proposed for that approach [27].

This brief discussion has indicated that the formulation presented in the previous sections can also be applicable to predict non-linear roll amplitudes which are more likely to be the maximum lifetime values experienced by ships.

The numerical results presented in the next section for the sensitivity of linear roll predictions to wave climate descriptions are expected to give a good indication about the sensitivity of the non-linear predictions.

5. NUMERICAL RESULTS

The sensitivity of the long-term predictions to wave climate description is assessed by calculating the 10^{-8} characteristic values of the motion predictions based on the different sets of wave data. The transfer functions for the ship motions have been calculated using a program based on the theory of Salvesen, Tuck and Faltinsen [28]. Some additional transfer functions were extracted from published results [29 - 30].

The results of the predictions of the long-term values of heave, pitch and roll for the North Atlantic are indicated in Table 1. The predictions were normalised by the ones obtained with the hindcast data of Bales et al [21], so as to make the relative effect clearer.

The main characteristics of the ships adopted

	Ship	Average All	Walden	Walden Modif.	Hogben & Lumb	Hg. & L. Modif.	Hogben et al	Bales et al	Bales et al
Heave	Containersh	0.83	0.89	0.86	0.76	0.71	0.74	1.00	15.8
	SL7	0.84	0.90	0.86	0.79	0.73	0.77	1.00	16.9
	Mariner	0.85	0.91	0.86	0.81	0.75	0.79	1.00	17.7
	Wolv. State	0.85	0.91	0.87	0.80	0.74	0.77	1.00	16.9
	S60-CB7	0.86	0.92	0.88	0.82	0.76	0.79	1.00	17.3
	S60-CB8	0.86	0.91	0.86	0.81	0.75	0.80	1.00	17.7
	Catamaran	0.91	0.97	0.94	0.88	0.82	0.85	1.00	18.8
	Destroyer	0.89	0.94	0.89	0.86	0.79	0.84	1.00	18.6
	All	0.86	0.92	0.88	0.82	0.76	0.79	1.00	17.5
Pitch	Containersh	0.81	0.90	0.93	0.73	0.71	0.61	1.00	13.6
	SL7	0.81	0.89	0.91	0.73	0.70	0.63	1.00	13.8
	Mariner	0.81	0.87	0.85	0.74	0.70	0.71	1.00	16.1
	Wolv. State	0.82	0.88	0.86	0.74	0.70	0.71	1.00	15.6
	S60-CB7	0.82	0.88	0.87	0.75	0.70	0.72	1.00	15.3
	S60-CB8	0.82	0.90	0.92	0.74	0.71	0.64	1.00	13.1
	Catamaran	0.75	0.72	0.76	0.69	0.65	0.67	1.00	19.1
	Destroyer	0.66	0.60	0.64	0.58	0.55	0.56	1.00	11.7
	All	0.79	0.83	0.84	0.71	0.68	0.66	1.00	14.8
Roll	Containersh	0.81	0.87	0.81	0.74	0.69	0.72	1.00	13.1
	SL7	0.85	0.91	0.80	0.80	0.75	0.81	1.00	80.8
	Mariner	0.83	0.88	0.82	0.78	0.72	0.76	1.00	40.0
	Wolv. State	1.04	1.15	1.11	1.03	0.97	1.00	1.00	39.2
	S60-CB7	1.22	1.50	1.29	1.19	1.16	1.16	1.00	29.1
	S60-CB8	1.10	1.23	1.20	1.09	1.04	1.05	1.00	25.2
	Catamaran	1.25	1.51	1.38	1.25	1.19	1.19	1.00	11.6
	Destroyer	1.22	1.33	1.38	1.24	1.18	1.19	1.00	19.0
	All	1.04	1.17	1.10	1.02	0.96	0.98	1.00	32.3
All Motions		0.90	0.97	0.94	0.85	0.80	0.81	1.00	21.5

Table 1- Relative value of the 10^{-8} characteristic value of wave induced motion amplitudes in the North Atlantic, normalised by the prediction based on the hindcast data. The last column indicate the absolute value of the latter predictions, with the heave amplitudes normalised by the wave amplitude and the pitch and roll amplitudes normalised by the wave slope.

in the calculations are shown in Table 2.

Table 1 shows the relative effect of using different sources of wave data. For each ship the difference in the predictions varies with the data set that is used by as much as 50%. The global average of all data sets is as much as 20% different from the predictions obtained with the data set of Bales et al.

In some cases the predictions based on the sets of visual observations led to lower values than the predictions from hindcast data but in others they led to larger values. This effect, which had also been observed in a similar study of the wave induced bending moments [5], does not allow a consistent trend to be determined from the data. Therefore this

SHIP	L(m)	C_b	Fn
Containership	270.00	0.60	0.245
SL7	260.38	0.53	0.200
Mariner	160.93	0.60	0.200
Wolverine State	151.18	0.61	0.200
S60-CB7	121.92	0.70	0.150
S60-CB8	193.00	0.80	0.150
Catamaran	67.00	-----	0.310
Destroyer	100.00	0.48	0.210

Table 2- Main characteristics of the ships considered in the calculations.

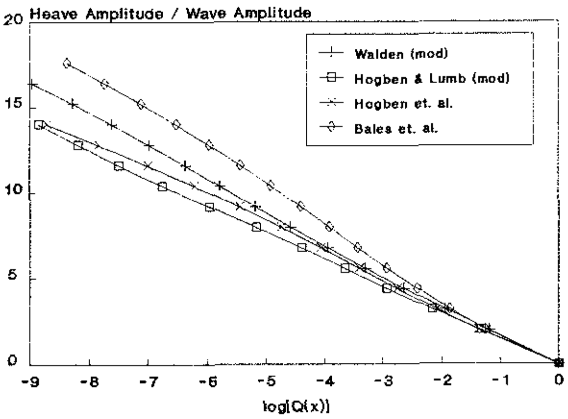

Figure 3a- Long-term probability distribution of heave amplitudes on the North Atlantic as obtained with different wave data set descriptions.

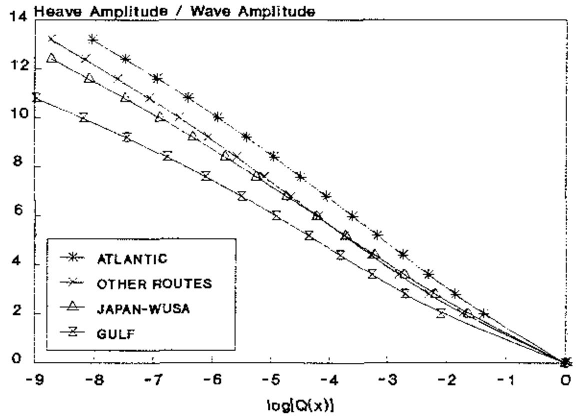

Figure 3b- Long-term probability distribution of heave amplitudes in various ship routes, based on the wave data set of Hogben et al [16].

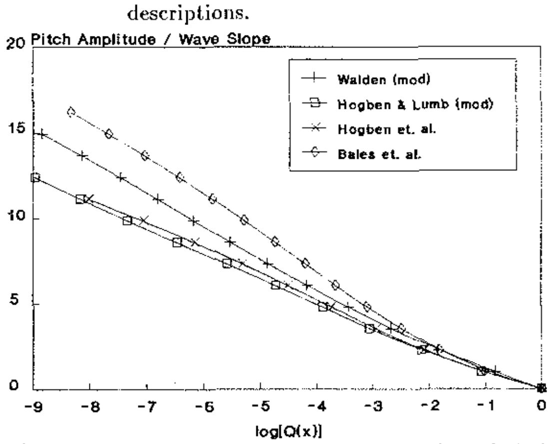

Figure 4a- Long-term probability distribution of pitch amplitudes on the North Atlantic as obtained with different wave data set descriptions.

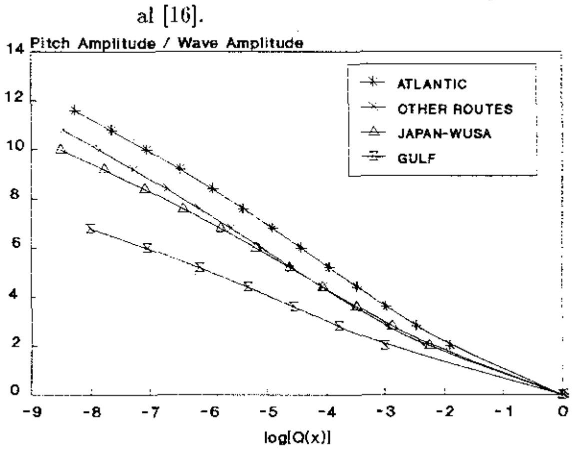

Figure 4b- Long-term probability distribution of pitch amplitudes in various ship routes, based on the wave data set of Hogben et al [16].

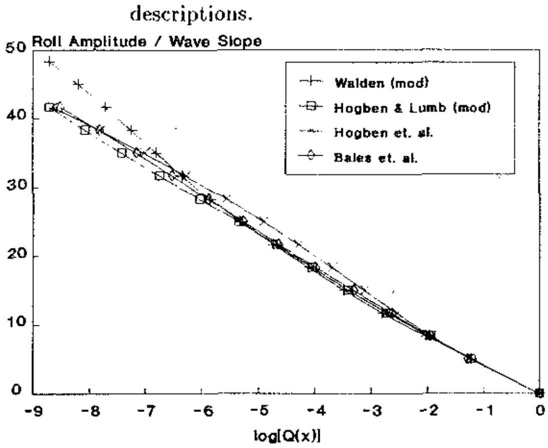

Figure 5a- Long-term probability distribution of roll amplitudes on the North Atlantic as obtained with different wave data set descriptions.

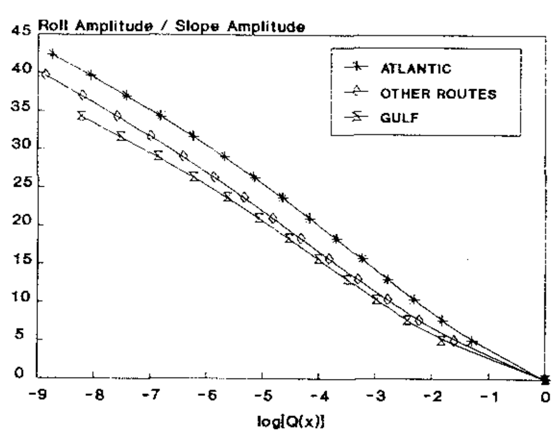

Figure 5b- Long-term probability distribution of roll amplitudes in various ship routes, based on the wave data set of Hogben et al [16].

effect has to be represented as a modelling uncertainty that quantifies the statistical uncertainty of having probabilistic descriptions of wave climate based on limited samples.

Table 1 also shows that the sensitivity to the wave climate description is different for the various types of motion under consideration, which again does not allow the same correction factor to be applied to all types of motions.

The predictions from the North Atlantic climate are compared with the ones from the other shipping routes, in Table 3, all of which were determined with the data set of Hogben, DaCunha and Olliver [16].

It is apparent that there are no significant differences within the North Atlantic. The differences for most of the other shipping routes and for all types of ships are consistently on the order of 10% lower. The only exception is the route from the Gulf to Japan and Japan to the West Coast of USA which show predictions 20% to 30% lower than for the North Atlantic.

Figures 3a to 5a show the long term probability distributions of heave, pitch and roll for the Wolverine State in the North Atlantic based on four different wave climate descriptions. It is apparent that the spreading is larger for heave and pitch than for roll. Figures 3b to 5b show the comparison of the North Atlantic predictions with the ones from the other ship routes, based on the wave data of Hogben et al [16].

The important difference between the results of Table 1 and Table 3 is that the latter are consistent while significant variations are observed in the first. Thus one can say that for most shipping routes the long-term predictions of wave induced heave, pitch and roll motions are about 10% lower than the corresponding predictions make for the North Atlantic, as can be observed in figures 3b to 5b in the case of the Wolverine State.

Except for the results from the route of Japan to the West Coast of the USA and the route from the Gulf to Japan, all the others coincide and thus they have been shown only as one probability density function in figures 3b to 5b.

The disturbing fact in the results is that the uncertainty of the long-term predictions made for the North Atlantic from different wave data sets is of the order of 20% to 40% as can be observed in Table 1, and in figures 3a to 5a. This implies that the uncertainties involved in the statistical description of the ocean wave climate are significantly larger than the differences among the results from different ocean

	Ship	Average All	Cape	Suez	Brasil	Eur. W. USA	Japan W. USA	Gulf	North. N.Atl.	South. N.Atl.	N. Atlantic	N. Atlant.
	Containersh	0.91	0.92	0.92	0.92	0.91	0.86	0.64	1.01	0.99	1.00	11.8
	SL7	0.92	0.92	0.92	0.93	0.92	0.88	0.72	1.01	0.99	1.00	13.0
	Mariner	0.93	0.92	0.93	0.93	0.92	0.88	0.75	1.01	0.99	1.00	13.9
Heave	Wolv. State	0.93	0.92	0.93	0.93	0.92	0.88	0.75	1.01	0.99	1.00	13.0
	S60-CB7	0.93	0.92	0.93	0.94	0.92	0.89	0.78	1.01	0.99	1.00	13.7
	S60-CB8	0.92	0.92	0.93	0.93	0.92	0.88	0.73	1.01	0.99	1.00	14.2
	Catamaran	0.95	0.92	0.95	0.93	0.91	0.93	0.89	1.01	0.99	1.00	15.9
	Destroyer	0.95	0.92	0.95	0.93	0.92	0.93	0.89	1.01	0.99	1.00	15.6
	All	0.93	0.92	0.93	0.93	0.92	0.89	0.77	1.01	0.99	1.00	13.9
	Containersh	0.88	0.93	0.89	0.90	0.89	0.85	0.49	1.02	0.97	1.00	8.28
	SL7	0.89	0.93	0.90	0.91	0.90	0.85	0.51	1.02	0.98	1.00	8.69
	Mariner	0.90	0.92	0.91	0.92	0.91	0.85	0.60	1.01	0.98	1.00	11.5
Pitch	Wolv. State	0.90	0.92	0.91	0.92	0.91	0.85	0.61	1.01	0.99	1.00	11.1
	S60-CB7	0.90	0.92	0.91	0.92	0.91	0.85	0.63	1.01	0.99	1.00	11.0
	S60-CB8	0.89	0.92	0.90	0.91	0.90	0.85	0.54	1.02	0.98	1.00	8.45
	Catamaran	0.94	0.92	0.94	0.94	0.92	0.93	0.85	1.00	1.00	1.00	12.9
	Destroyer	0.94	0.92	0.94	0.94	0.92	0.93	0.85	1.00	1.00	1.00	6.59
	All	0.91	0.92	0.91	0.92	0.91	0.87	0.63	1.01	0.99	1.00	9.81
	Containersh	0.89	0.92	0.91	0.91	0.90	0.85	0.57	1.01	0.98	1.00	9.44
	SL7	0.90	0.91	0.91	0.92	0.91	0.85	0.60	1.01	0.99	1.00	65.7
	Mariner	0.92	0.92	0.92	0.92	0.91	0.86	0.72	1.01	0.99	1.00	30.2
Roll	Wolv. State	0.94	0.92	0.94	0.94	0.92	0.91	0.85	1.01	0.99	1.00	39.1
	S60-CB7	0.94	0.90	0.96	0.89	0.87	0.90	0.92	1.03	0.95	1.00	33.8
	S60-CB8	0.95	0.92	0.95	0.93	0.91	0.92	0.90	1.01	0.98	1.00	26.4
	Catamaran	0.95	0.92	0.95	0.92	0.91	0.92	0.90	1.01	0.98	1.00	13.8
	Destroyer	0.95	0.92	0.94	0.93	0.92	0.93	0.88	1.01	0.99	1.00	22.6
	All	0.93	0.92	0.93	0.92	0.91	0.89	0.79	1.01	0.98	1.00	30.1
All Motions		0.92	0.92	0.92	0.92	0.91	0.88	0.73	1.01	0.99	1.00	17.7

Table 3- Relative value of the 10^{-8} characteristic value of wave induced motion amplitudes in various ship routes, normalised by the prediction based on the hindcast data. The last column indicate the absolute value of the latter predictions, with the heave amplitudes normalised by the wave amplitude and the pitch and roll amplitudes normalised by the wave slope.

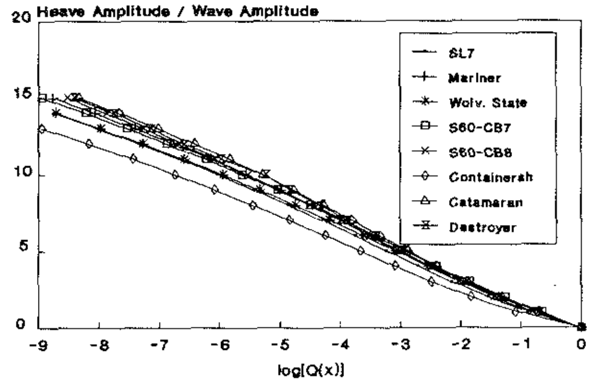

Figure 6a- Long-term probability distribution of heave amplitudes of different ships on the North Atlantic obtained from the data set of Hogben et al [16].

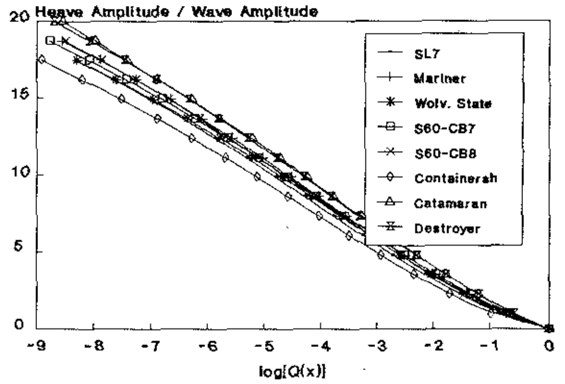

Figure 6b- Long-term probability distribution of heave amplitudes of different ships on the North Atlantic obtained from the data set of Bales et al [21].

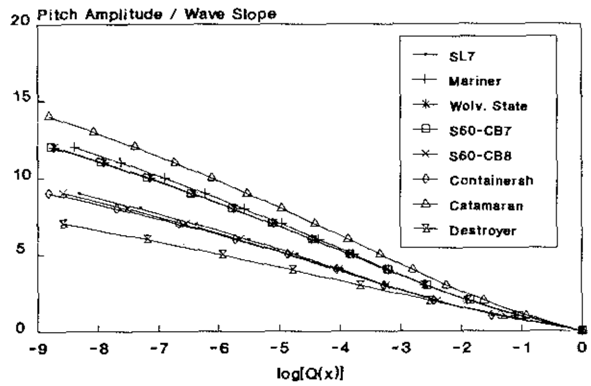

Figure 7a- Long-term probability distribution of pitch amplitudes of different ships on the North Atlantic obtained from the data set of Hogben et al [16].

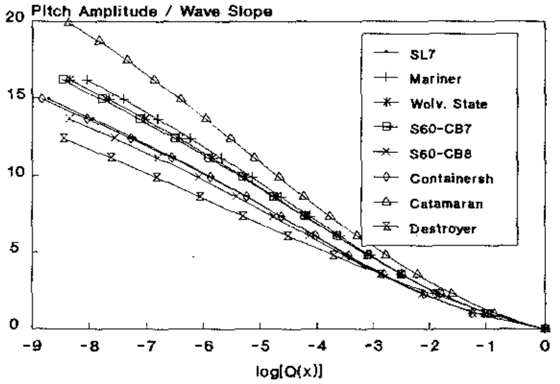

Figure 7b- Long-term probability distribution of pitch amplitudes of different ships on the North Atlantic obtained from the data set of Bales et al [21].

areas.

Figures 6 and 7 show the long term probability distributions for the different ships considered in this study subjected to the North Atlantic wave climate, as described with the data of Hogben et al [16] (figures a) or of Bales et al [21] (figures b).

It is interesting to observe in those results that the spreading of the characteristic 10^{-8} pitch amplitudes is larger than the one of heave amplitudes.

Furthermore, the predictions based on the data of Bales et al [21] are larger than the ones from Hogben et al [16].

6. CONCLUSIONS

This work has shown that for the set of ships studied, the 10^{-8} characteristic values of motion amplitudes obtained from long-term distributions based on different wave climate data sources can have differences ranging between 70% and 100% of the predictions based on hindcast data or 55% to 100% in the case of pitch, or 70% to 150% for the roll predictions.

The larger differences were obtained between the results from the hindcast data and the ones from Hogben, DaCunha and Olliver in some cases and the ones of Walden in others.

The design values based on the hindcast data are generally conservative for heave and pitch but the tendency is not so clear for roll.

Since there is no definite basis to choose one data set instead of other, the discrepancies shown in Table 1 must be interpreted as the range of uncertainty due to that lack of knowledge.

ACKNOWLEDGEMENTS

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THE ASSESSMENT OF DAMAGED STABILITY CRITERIA USING MODEL TESTS

Alan Graham[1]

SYNOPSIS

The Court of Inquiry, set up to investigate the reasons why 192 lives were lost aboard the United Kingdom-registered Ro Ro passenger ferry off Zeebrugge Harbour, made a number of recommendations concerning research into residual stability standards for such ferries. This paper outlines the research which was commissioned by the Department of Transport (Marine Directorate).

In particular, the research relates to two series of 'damaged' model tests carried out in waves of varying significant wave height, with the models beam-on to on-coming waves. The intention was to determine critical zones where capsize or non-capsize was equally probable.

At the same time, statical calculations were performed by computer at the appropriate ship condition (determined by draught, trim and KG for the damaged condition). In this manner, a link was made between the residual stability as required by the regulations and the dynamic situation indicated by the test results.

The particular set of regulations used to make this comparison are those which came into force on 29 April this year for new passenger ships. (Sometimes referred to as the SOLAS '90 residual stability standards). An important conclusion from an examination of the results of this research is that in order to provide reasonable protection against capsize, assuming side damage occurs in the most critical region of the ship, the residual stability standard should be at least that of SOLAS '90.

The corollary to this is that Ro Ro ferries built before April 1990 are unlikely to possess adequate residual stability standards, except in well-nigh still water conditions.

The IMO members have been informed of these important findings; the topic concerning the application of these admittedly higher standards to existing passenger Ro Ro ferries is to be discussed at the relevant meetings next year - the Sub-Committee on Subdivision, Loadlines and Fishing Vessels Safety in February and the Maritime Safety Committee in May.

Further research is shortly to be commissioned by the Department of Transport. This second phase of research will deal with the degree of enhancement in survivability that the fitting of various devices or design modifications to present designs might provide.

Up until now, this improvement in residual stability was measured by a series of statical calculations. Model tests are needed to verify that the various devices would provide this improvement in a seaway (or indeed, whether dynamically the improvement may be even better than that indicated by the purely statical calculations).

It is hoped that tentative results should be available in time for the IMO discussions next year.

INTRODUCTION

It is now more than three years since the "Herald of Free Enterprise" capsized outside Zeebrugge Harbour, with heavy loss of life. The prime cause of this tragic event was, as is stated in the Court of Inquiry report (1), that the vessel went to sea with both her inner and outer bow doors open. As a result, considerable quantities of flood water accumulated on the vehicle deck which caused the vessel to heel very quickly to a significant angle, resulting in a very rapid capsize. Hopefully, the statutory measures that have since been introduced should ensure that there is no recurrence of this type of incident. However, it is recognised that the provision of large, unrestricted spaces within a typical Ro Ro passenger ship is potentially dangerous, if water in considerable quantities is permitted to accumulate on the vehicle deck of such vessels. It is considered that the greatest chance of flood water gaining entry to the vehicle space is when a side collision has occurred, either with another vessel or a fixed object.

The formal investigation into the loss of the "Herald", in addition to establishing the circumstances surrounding the casualty, also considered what future measures could be taken to contribute to enhanced safety of life at sea in the future. As a result of recommendations made in the Inquiry report, the Marine Directorate of the UK Department of Transport commissioned a comprehensive research programme with the objective of enhancing the survivability of Ro Ro passenger ferries, after specified side damage has been assumed to occur.

The programme consisted of various elements, with the basic objective of determining the standard of residual stability necessary to enable Ro Ro passenger ferries to survive flooding, to a prescribed extent, and to avoid rapid capsize in realistic sea-going conditions. A Risk analysis study was made to establish the level of risk involved in operating a typical Ro Ro passenger ferry between the United Kingdom and the near-Continent of Europe. In this way, any proposed improvement measures arising from the research work intended to improve post-damage survival characteristics, could be assessed in terms of level of risk from other hazards such as fire/explosion and mal-operation. It should be noted that the study confirmed that the primary hazard that might lead to a rapid capsize was, indeed, a major side-collision with another

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vessel. Before attempting to decide what was a suitable survivability standard for Ro Ro passenger ferries, it was necessary to establish the current standards applying to the UK fleet. Accordingly, ten typical designs of Ro Ro passenger ferry were chosen; statical calculations (assuming still water) were made to determine the degree of compliance with the set of residual stability criteria, (2), being discussed at the time at the International Maritime Organisation (IMO). This same set of criteria, with only relative minor modifications, was accepted by IMO members and entered into force on 29 April 1990, and apply to all new passenger ships built after that date. These criteria may be referred to as SOLAS '90 residual stability standards. (3) This computer study showed quite clearly that conventional Ro Ro passenger ferry designs have little chance of complying with these new residual stability standards; in most cases, radical design charges are indicated. It was recognised from the outset that it was likely that the survivability standards of current Ro Ro ferries would need to be enhanced significantly, to meet the standards indicated by the research. There were strong indications of this in the series of model tests carried out in the early 1970s in the United Kingdom on a typical Cross-Channel ferry of that time (4). Accordingly, the research included the consideration of various possible design changes and enhancing devices which could be employed on existing ferries to improve their survivability characteristics. These improvements may be achieved in two main ways - by the fitting of internal or external arrangements. In broad terms, the former restricts the extent of internal flooding which might occur, whilst the latter increases the potential to right the vessel after damage has occurred. All the research studies described previously are useful and provide valuable data for the future, but the essential question - "What stability standards are required to give a reasonable guarantee that a ferry built to those standards will not capsize rapidly?" - needs to be answered. Unfortunately, a study of historical damage data, whilst useful in indicating the likely position and extent of any future damage, does not provide much assistance in answering this question. It was decided that it was necessary to conduct a series of damaged model experiments in controlled conditions of damage location and extent, condition of loading and weather conditions (measured in terms of sea state). The data provided from such tests would then provide the means to decide what minimum residual standards of stability are needed to avoid rapid capsize in sea-going conditions.

A more complete summary of the research work is contained in an Overview report (5), published by the Department of Transport and also distributed to IMO members as an information paper (6).

TEST PROCEDURES

Two organisations were chosen to carry out the model tests - British Maritime Technology (BMT) of Teddington, England and the Danish Maritime Institute (DMI) of Lyngby, Denmark. Each organisation was commissioned to construct and test a 42nd scale GRP model, representing a typical cross-Channel Ro Ro ferry. Fig 1 illustrates the principal features of one of the models.

The damaged space below the vehicle deck was located in the midships region, the length and longitudinal position being arranged such that pure sinkage occurs without significant trim down to the level of the vehicle deck. The assumed damage was fixed at the statutory limit of 0.03L + 3.0 (metres), with a Vee-shaped penetration of B/5 at the deck level. The ship condition was adjusted by the use of light, impervious inserts at both ends of the damaged space, together with appropriate means of adjusting the model KG. The vehicle space and the midships damaged space were essentially free of obstructions which might restrict the flow of floodwater during the tests.

The models were tested in irregular beam seas, generated by a wavemaker, for various significant wave heights ranging from 0.5 metres to 5.0 metres. JONSWAP wave spectra were used with modal periods appropriate to wave scatter data gathered on an all-seasons basis from the southern North Sea region. For the tests, the models were placed in the path of the on-coming waves, with the damage opening facing them, for the majority of cases. (Some tests were carried out with the damage on the side remote from the on-coming waves). Readings of the model motions were taken continuously, whilst simultaneously the behaviour of both the model and the flood water within the hull were being observed.

For these tests where capsize did not occur within a specified test period, the model KG was raised and the entire procedure repeated until capsize took place.

Where capsize occurred extremely rapidly, the tests were repeated using lower KG values to ensure that the estimated capsize zone was defined within as narrow a band as possible.

LINKING TEST RESULTS WITH STATICAL CRITERIA

Present-day statutory stability criteria are based on static principles and it is unlikely that this situation will change within the foreseeable future. These empirical standards are essentially based on the characteristics of a GZ curve, calculated assuming still-water conditions for a specific condition defined in terms of draught, trim and KG. They do not

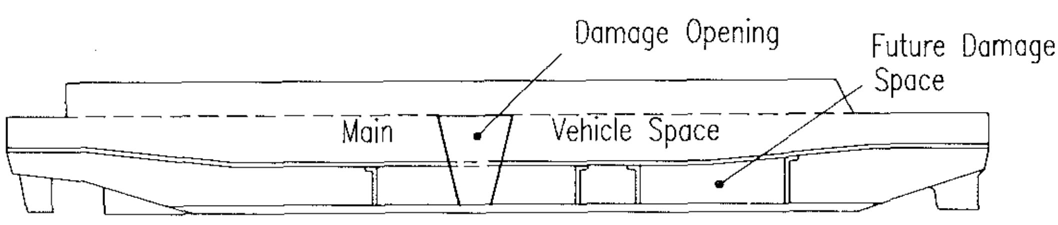

Fig 1 : General Arrangement of one of the two Models therefore take direct account of the complicated dynamic motions which actually occur in sea-going conditions.

In respect of present intact stability standards, all the available evidence suggests that they are adequate for all sea states likely to be encountered, provided that watertightness is preserved and shift of cargo does not take place.

However, where residual stability standards are concerned, empirical data is sparse and there is therefore some doubt as to whether a ship complying with these standards would survive statutory damage in realistic sea-going conditions.

It is reasonable to assume a positive correlation between the residual stability curve characteristics (consisting of GZ max, range and area) and the corresponding ability of a ship having a residual standard capable of resisting capsize in reasonable sea-going conditions.

The term 'capsize' needs to be defined as far as the conduct of these model tests is concerned. A rigid interpretation was deliberately not chosen; rather, 'capsize' was considered to have taken place when the rate of change in heel shows a distinct increase.

Each test run was carried out over a maximum timespan, equivalent to at least 60 minutes ship-scale. Non-capsize was assumed, unless capsize (as defined above) occurred within this timescale.

STATUTORY RESIDUAL STABILITY CRITERIA

Intact stability criteria are empirical in nature and based upon the principal characteristics of a GZ curve - positive area(s) under the curve, GZ max, and range. These criteria have been applied for many years now and are thoroughly tested in all sea conditions likely to be encountered. There is no firm evidence that an ocean-going ship has been lost due to inadequate intact stability, always provided of course that no water ingress or shift of cargo took place. Indeed, there may be some redundancy in the present criteria; moreover certain ship types and lengths may have criteria which are more stringent than is strictly required. However, the present level of intact stability criteria seems to meet with general agreement.

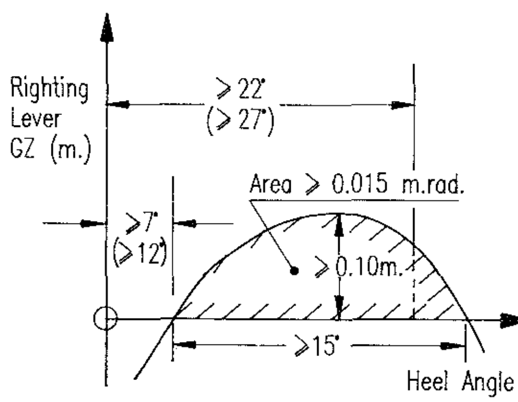

Fig 2 : Diagrammatic representation of the SOLAS '90 residual stability criteria
Note : ( ) are the criteria for two-compartment standard

In respect of damage stability, the criteria are similar in format to that which apply to intact stability; there is an additional requirement that a minimum residual freeboard is to be retained after assumed damage - in other words, a requirement that the margin line is not to be immersed. However, because the incidence of serious flooding caused by side-collision damage is relatively low, evidence regarding the desired level of residual stability needed to survive is quite sparse.

Fig 2 shows diagrammatically the recently adopted international standard of residual stability which applies to passenger ships built after April 1990. They represent a significant improvement in safety standards compared to those applied previously, but they remain standards which do not relate directly to actual sea conditions, being based on statical considerations. It is most likely that, for the forseeable future, statutory stability requirements will be based on statical calculations. Historical data derived from previous damage incidents will, hopefully, continue to be rare; therefore there is a need to allow for the dynamics of a flooding scenario and the most straightforward way of doing this seems to be by a series of controlled tests involving "damaged" models.

A direct link may then be established between the results of statical (still-water) calculations and the corresponding test conditions, to enable a judgement to be made as to the chances of survival of a ship in a given sea-state.

By such means, the main factors which govern survival after damage - damage location and extent, ship condition, internal arrangement of watertight divisions and permeability of damaged spaces - may be examined in a systematic manner.

ANALYSIS OF THE MODEL TEST RESULTS

The two ferries modelled for the tests are quite representative of those UK passenger ferries engaged on the cross-Channel routes. Present-day designs of such ferries do not vary significantly in either size or proportions. All tend to have the common features of extensive, unrestricted vehicle spaces - apart from relatively small side or centre casings. To derive maximum benefit from the two sets of tests - and, in addition, to take account of the tests carried out in the UK some 20 years ago (4) - a logical, coherent theory is needed to extend the analysis to ships of different size and proportions. Such a theoretical approach needs to be consistent with the observed behaviour of the models under test, when subjected to various sea states and with the damage opening facing towards (or away from) the on-coming waves.

Observations at the time of the test runs indicated that in order that capsize should take place, sinkage has to occur. In other words, there has to be a net increase in the amount of flood water taken on to the vehicle deck. In addition, there is a gradual increase in heel angle - in most cases, but not all, towards the side of the damage openings - culminating in a significantly increased rate of heel angle immediately prior to capsize.

Capsize is unlikely to occur where the minimum residual freeboard (in still water) is at least as great as the height of the on-coming waves, (as represented by the significant wave height).

Qualitatively, then, a ferry's ability to resist capsize is dependent on

: the condition of loading,

: the area of the vehicle deck capable of being flooded, and

: the minimum freeboard after damage.

Additionally, a more severe sea state will mean that increasing amounts of flood water will be taken aboard, and thus reduce the ability to resist capsize. Hence, for a given sea state the ability to survive may be regarded as being represented by some complex function of KG (or GM), B, f and C_B .

Fig 3 illustrates a typical statical residual GZ curve calculated for a specified KG position and residual freeboard, the latter values being taken from the test results at positions where the model had a roughly equal chance of surviving or capsizing.

Fig 4 shows a plot of such positions, given for a specified damage location, damage orientation and residual freeboard. It is represented as significant wave height against flooded GM.

It is important that the results obtained from both sets of model tests, and the previous UK tests mentioned earlier should be presented in a format capable of a more general analysis. A non-dimensionalised form of presentation was suggested by BMT and tested on the two sets of results - see Fig 5. More work needs to be done in this respect, but the initial indications are that this line of approach is promising.

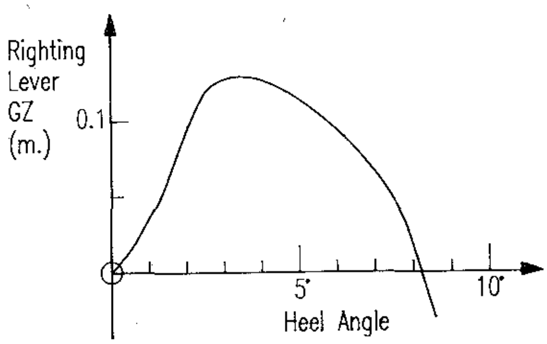

Fig 3 : Typical calculated residual GZ curve (ship-scale)
Corresponding Model Test Condn.
Midship Damage : Facing Waves:
0.58m. residual freeboard:
1.00m. significant wave height.

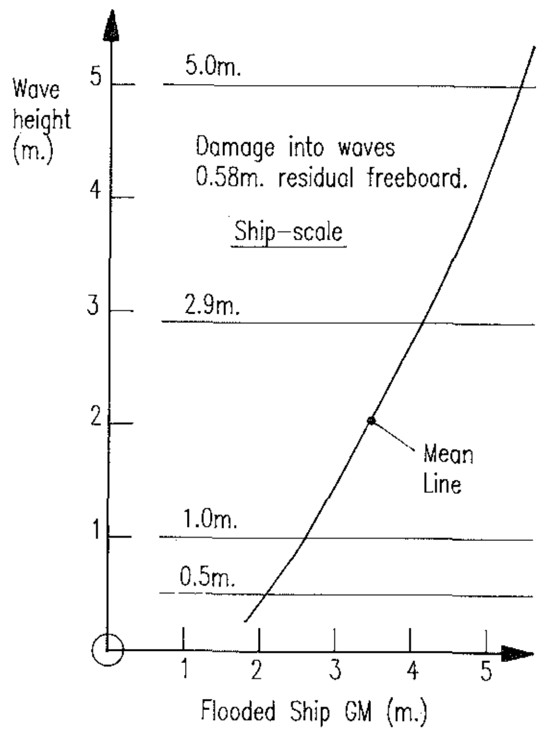

Fig 4: Plot of wave height ~ flooded GM
Note : The mean line has been derived from a zone of possible capsize either side of this mean.

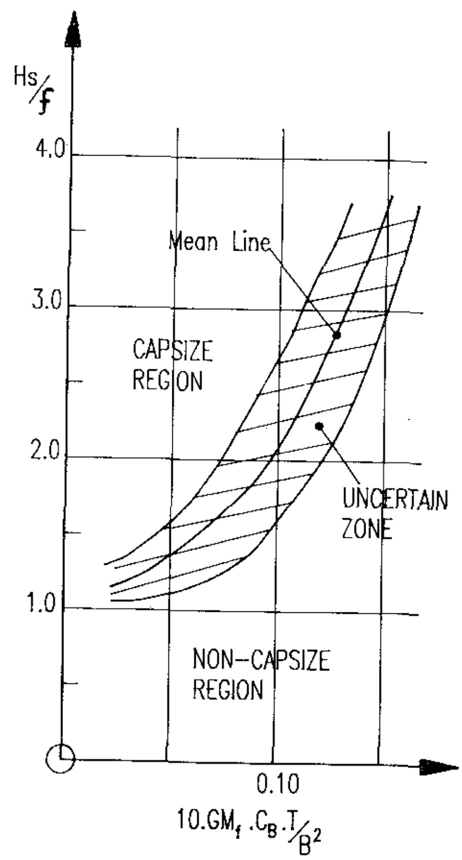

Fig.5 : A suggested non-dimensional representation of the test results.

FURTHER RESEARCH

All the tests conducted to-date relate to a typical cross-Channel ferry of current design, with few or no obstructions within the vehicle spaces.

In the research undertaken so far, the enhancement in survivability through the fitting of various devices/arrangements has been investigated by means of statical (still water) computer calculations. There is a need, therefore, to conduct further tests to confirm that the improvements in residual stability indicated by these calculations are attainable in sea-going conditions. Therefore, it is intended to carry out a further series of model tests, using the models as-built, together with a modelling of the principal devices/arrangements studied in Phase 1 of the tests.

When the two models were constructed, provision was made for assumed damage in the forward end, and it is intended to run some tests so as to indicate what standards of survivability are required for damage in this region; this will enable comparisons to be made with the results obained assuming damage amidships.

In making the damage stability calculations, it was clear that the choice of the permeability factor for the vehicle space could affect the GZ max and range significantly. It is intended to model a full load of vehicles on deck and conduct tests, so that the results might be compared to the corresponding statical calculations - the latter using an appropriate permeability factor for the vehicle space.

ACTION ON THE RESEARCH

The Steering Committee, which was formed to oversee the UK research programme, produced a report which was forwarded to the Minister for Transport, in which they recommended that SOLAS 90 residual stability standards be applied to existing UK Ro ro passenger ferries as soon as possible. They further recommended that these same standards should also apply to non-UK ferries using UK ports. The Minister accepted these recommendations and subsequently the UK has put forward its views at the 58th Session of MSC.

Recently, the near-European Administrations were approached to explain the UK position and to establish the degree of support the UK might receive in applying SOLAS 90 standards to existing passenger ferries. A formal presentation to IMO of the UK position will be made at the forthcoming SLF Sub-Committee and the MSC next year.

It is expected that some preliminary findings of the Phase 2 model tests should be available at that time.

SUMMARY AND CONCLUSIONS

The main objective of the reseach programme described was to determine the standard of residual stability needed to enable a Ro Ro passenger ferry to survive flooding and avoid rapid capsize in realistic sea-going conditions.

Relating the model results and subsequent analysis to typical present-day Ro Ro ferries, it can be said that this primary objective has been achieved. In particular, the results indicate that a ferry (with a vehicle space free of obstructions) should have a reasonable chance of survival (ie. will not capsize rapidly) if built to comply with SOLAS 90 standards of residual stability.

The research programme also provided designers with much useful data concerning various devices/design arrangements which might be fitted on new or existing ferries in order to enhance their survivability. There is a need to model these devices in a further series of tests in order to establish that the apparent enhancement indicated by still-water computer calculations is indeed achieved in a dynamic situation.

Complete benefit from the model test results will not be achieved until there is a consistent, logical theory which will enable the rsults to be applied to other ship forms and sizes. Observing the models during the tests provides an insight into the main factors which need to be taken into account in producing such a theory. The model results provide a good basis for judging the intrinsic worth of any theory proposed.

NOMENCLATURE

IMO	Intergovernmental Maritime Organisation
SLF	Sub-Committee on Sub-division, Loadlines and Fishing Vessels Safety
MSC	Maritime Safety Committee
SOLAS	Safety of Life at Sea (Convention)
SOLAS 90	A set of residual stability criteria which is to be applied to passenger ships built after 29 April 1990
HMSO	Her Majesty's Stationery Office
KG	Height of ship centre of gravity
GM	Metacentric height
GZ	Righting lever
f	Minimum flooded freeboard
B	Ship's moulded breadth
T	Ship's moulded draught
H	Significant wave height

ACKNOWLEDGEMENTS

The UK research programme on Ro Ro passenger ferry safety has provided the source material for this paper. The research was commissioned by the Marine Directorate of the Department of Transport, and permission by the Directorate's Surveyor General to publish is duly acknowledged. Essentially, the contents of the paper are factual and informative; where opinions are expressed, they do not necessarily reflect those of the Department of Transport.

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SMALL VESSEL OPTIMIZATION FOR INCREASED SEAKEEPING AND STABILITY PERFORMANCE

Radoslav NABERGOJ (*)

ABSTRACT

In this paper an extended version of the rank criterion is proposed for optimizing both the seakeeping and stability qualities of fishing vessels since concept and preliminary design stages. The mathematical model is quantified by means of a large amount of stability calculations and of vertical plane ship responses in different sea conditions for a family of homogeneous hull forms of equal displacement. In the model, the vessel hull geometry is related to a global rank in order to evaluate the influence of main characteristics in determining the performance of the ship in both seakeeping and stability.

The derived results are an important tool for practical use due to their reliability and simplicity. Furthermore, the rank criterion here presented can be included as an analytical module within a general design code for global design optimization when using multicriterial decision-making techniques.

INTRODUCTION

It is very difficult to obtain safe and confortable navigation with ships whose characteristical dimensions render them more sensitive to the action of the marine enviroment. Among these, a relevant place is reserved to fishing vessels the hull shape of which varies greatly due to local conditions, catch methods, construction material, engine weights, distance to the fishing grounds and other factors. Thus, it is problematic to design a few standard hulls which could be suitable for all enviromental conditions.

Different organizations have collected and published results of theoretical calculations, model tests and full scale trials in an attempt to indicate the trend in the factors which influence stability, resistance, powering and seakeeping of the vessels. The final goal is to reach conclusive indications on how to find the optimum hull shape when designing a new fishing boat. The statistical analysis of the data available is therefore intended for estimating the total performance of an existing design so it can be investigated if there is still room for any further improvement.

The present paper is aimed at providing the designer with easy and reliable tools for evaluating stability and seakeeping qualities of fishing vessels since the preliminary design stages. A systematic numerical analysis has been performed for a family of fifteen single-screw hull forms derived from worldwide systematic series. As a result, a quantitative design merit index has been obtained for estimating the influence of hull geometry on the global performance of a homogeneous class of vessels.

It is hoped that this work will lead to a more complete statistical evaluation of the ship's behaviour in waves in order to assist the designer in making effective improvements in habitability, operational efficiency and safety. To this end, once one recognizes particular objectives in the predictional approaches of ship design, the search for corresponding non-ambiguous optimum solutions and consistent standardized evaluations should be introduced in any optimization procedure. The philosophy is that the conditions for superior performance must be imposed already in the early stages of design synthesis of the hull forms.

FAMILY OF FISHING VESSELS

The numerical investigations were carried out on a family of fishing vessels known as the BSRA trawler series [1,2]. The parent form for the series, model XF, was chosen to represent a ship 150-ft. LBP

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× 26-ft. 4 in. moulded breadth × 13-ft. 2 in. moulded draught with a displacement of 847 tons salt water.

The hull forms of the series were derived from the parent form according to the following transformations:

- affine distortion of B/T (B/L, T/L) starting from ship XF for hull forms XG, WO, WP, 907, while keeping $L/\nabla^{1/3}$ constant;
- affine distortion of $L/\nabla^{1/3}$ (B/L, T/L) starting from ship XG for hull forms WS, WR, WQ, while keeping B/T and ∇ constant;
- variation of C_B (C_P) by conformal transformation of the sectional area curve, starting from ship XF for hull forms ZP, ZQ, but having main dimensions constant;
- same as above, starting from ship XF for hull forms 851, 852, while maintaining B/T and $L/\nabla^{1/3}$ constant;
- variation of x_{CB} from ship XG for hull forms 975, 977, 978, by modifying the sectional area curve only.

All the hulls of the family have been normalized to the same displacement through geometrical similarity and are fitted with similar superstructures independently on their length. Their main geometrical characteristics are given in Table 1. The forecastle has an extension of approximately 25% of the ship's length and the freeboard is sufficient to avoid deck immersion until a heel of 12,5°. A schematic representation of the vessels is given in Fig.1.

MATHEMATICAL MODEL

The practical need for developing valid relationships between hull forms and both its stability and seakeeping qualities requires a simple synthesis of the results obtained from computer simulations or experimental tests. In the preliminary stages of the design a merit index of the ship qualities under consideration has to be introduced for an objective comparison between different solutions. This merit index has to be related to a small number of geometrical hull form parameters and has to be quantitatively estimated through a mathematical model capable of assigning the proper weight to each hull characteristic.

Table 1 - Main characteristics of the BSRA trawler series.

	L	B	T	D	C_B	C_W
BSRA I XF	45.72	8.03	4.07	4.95	0.562	0.774
BSRA I XG	45.72	8.50	3.85	4.77	0.561	0.776
BSRA I WO	45.72	8.96	3.64	4.63	0.562	0.776
BSRA I WP	45.72	9.40	3.48	4.51	0.562	0.777
BSRA I 907	45.72	10.62	3.09	4.21	0.560	0.777
BSRA I WS	41.01	8.98	4.07	5.04	0.559	0.772
BSRA I WR	43.36	8.73	3.95	4.90	0.560	0.765
BSRA I WQ	48.09	8.30	3.75	4.46	0.561	0.775
BSRA II ZP	45.72	8.03	4.08	4.97	0.521	0.742
BSRA II ZQ	45.72	8.03	4.06	4.95	0.593	0.806
BSRA II 851	45.72	8.32	4.23	5.15	0.521	0.743
BSRA II 852	45.72	7.82	3.96	4.81	0.593	0.805
BSRA II 975	45.72	8.50	3.69	4.77	0.584	0.816
BSRA II 977	45.72	8.50	3.72	4.76	0.580	0.795
BSRA II 978	45.72	8.50	3.74	4.75	0.577	0.790

The choice of the descriptive hull parameters must be objective if one wants completeness and modularity:

- hull forms have to be described schematically but completely, not neglecting either moulded breadth or afterbodies forms;
- the statistical analysis will indicate the parameters which can be eventually disregarded.

In order to develop such a mathematical model, the optimization approach here presented follows the guidelines suggested by Bales [3] in introducing the well known "seakeeping rank". The assumptions at the root of our rank criterion are the following:

- one can define the target of the ship by means of a comparative index;
- this index can be evaluated through a linear relationship of a limited set of general geometrical parameters descriptive of the hull form.

Hence, for the considered family, it will be possible to perform:

- a detailed evaluation, by introducing several "specific ranks", each one pertaining to each single target;
- a general evaluation, by calculating a "global rank" which represents the synthesis of all the targets considered.

The single design merit indexes are determined on the basis of a suitable set of ship qualities corresponding to a large set of operative conditions. For sake of simplicity, in this preliminary study the definition of the "specific ranks" was limited to seakeeping and stability qualities of the vessel. Furthermore, each target has had the same relevance when evaluating the "global rank".

SEAKEEPING RANK

The design task of a seakeeping optimization process is to reduce motions and the induced dynamic effects. Here, to simplify the model, only the most relevant effects due to vertical motions will be taken into account: pitch, heave, relative vertical motions at bow and at stern. There are not considered ship responses such as absolute vertical motion at stern, heave acceleration, absolute vertical acceleration at bow, slamming occurrence which have been considered by other authors [3-6]. The calculations of the significant values for the responses were carried out in long-crested, head seas by means of a seakeeping program based on the strip-theory of Salvesen-Tuck-Faltinsen [7]. The longitudinal radius of gyration was assumed to have a constant ratio to the length, i.e. k_θ=0.24L, for all the vessels. The sea considered was represented by means of the Pierson-Moskowitz spectrum for modal periods T_W equal to 6, 8, 10, 12 seconds corresponding to Beaufort scale 3 through 7. The speeds considered include the whole range of operative conditions from towing to free running (V_S approximately 3, 7, 10, 14 knots). Hence the computation for 64 dynamic responses were carried out.

The evaluated responses were normalized to the significant wave height, and then combined to obtain the average responses which, to a first approximation, can be considered as a merit index for each seakeeping characteristic (Table 2). The average values were then quoted as adimensional ratios to the corresponding minimum value obtained for the family. To obtain the "seakeeping rank" for each hull, the respective adimensional response values were summed with equal weight. A change in scaling was made to facilitate the evaluation of the rank, by assigning the

Table 2 - Average significant responses in long-crested sea and seakeeping rank.

	$\frac{\theta_{1/3}}{(H_w)_{1/3}}$ (deg/m)	$\frac{z_{1/3}}{(H_w)_{1/3}}$ (m/m)	$\frac{(r_{1/3})_0}{(H_w)_{1/3}}$ (m/m)	$\frac{(r_{1/3})_{20}}{(H_w)_{1/3}}$ (m/m)	R
BSRA I XF	2.008	0.459	1.048	0.780	2,305
BSRA I XG	1.941	0.441	1.016	0.746	4,069
BSRA I WO	1.886	0.427	0.991	0.722	5,527
BSRA I WP	1.839	0.418	0.970	0.706	6,697
BSRA I 907	1.729	0.393	0.921	0.655	10,000
BSRA I WS	2.141	0.476	0.979	0.764	2,224
BSRA I WR	2.036	0.458	0.998	0.755	3,151
BSRA I WQ	1.851	0.426	1.030	0.743	4,978
BSRA II ZP	2.029	0.466	1.098	0.787	1,369
BSRA II ZQ	1.974	0.451	1.002	0.770	3,430
BSRA II 851	2.072	0.474	1.089	0.788	1,000
BSRA II 852	1.940	0.445	1.007	0.767	3,754
BSRA II 975	1.831	0.455	0.947	0.728	5,653
BSRA II 977	2.028	0.430	1.072	0.757	3,043
BSRA II 978	2.096	0.414	1.117	0.757	2,674

10.0 value to the best hull and 1.0 to the worst one with respect to the total seakeeping response. The adimensional values for the vessels with intermediate behaviour are obtained accordingly, see Fig.2. In this way, an implicit weighting is introduced for the averaged responses, thereby avoiding the risk of assigning too much importance to factors not bearing on seakeeping improvement.

In a first analysis, one can infer that the hull forms referred to as BSRA I possess, in general, the best seakeeping characteristics. As regards the parameters which appear to have a strong influence on seakeeping, one can say that good responses correspond to high B/L and to low T/L ratios. With reference to hull forms BSRA II 975, 977, 978, a backward position of x_{CB} is advantageous for reducing heave motion and thus vertical acceleration, whereas all other motions are reduced for x_{CB} forward of midship [6].

ENERGY BALANCE IN WAVES

To define the "stability rank", we will refer to the well known energy balance method recently proposed by Strathclyde University [8,9]. This method is based on the consideration of a time-dependent roll restoring moment and, in addition to the combined effects of beam wind and rolling, it takes into account also the effect of a following or quartering sea on the stability of the ship. In the proposed procedure the righting arm is computed for different positions of a wave with respect to the vessel. The wave is assumed to have the same length of the ship and the encounter period equal to her natural rolling period. Then, an ultimate half roll is supposed to occur between a windward angle and an extreme leeward angle. For each position of the wave relative to the vessel the minimum righting arm curve during the ultimate half roll is obtained and the corresponding energy-balance, taking into account damping and wind, is computed to give the net area, as shown in Fig.3. The percentage of time with net area positive during the passage of the wave along the ship is assumed as an indicator of the safety from capsizing. Further details can be found in Ref.9.

For the present application, the parameters describing the ultimate roll and the wind lever were estimated using the prescriptions of I.M.O. weather criterion. In the numerical simulations no roll damping was assumed to act on the ship when computing the net area. A rough estimate made on one of the vessels, indicated that the effect of damping could increase the percentage of net area positive by an ammount of 10 to 20%.

To apply the Strathclyde method in practical design it is necessary to compute the net area for different loading conditions. In our case only KG/D was varied, the displacement remaining constant. The results are summarised in Fig.4 where we show only the higher and the lower curve obtained for the considered vessels. The computations, including *weather and sea effects quite realistically, give an* indication of the "ability" of the ship to avoid capsize and thus they allow us to distinguish, for a certain position of the centre of gravity, between a safe and an unsafe ship. Intermediate conditions of safety for the ship are implicitly defined.

STABILITY RANK

In designing a new ship the target is to produce a vessel which allows higher locations for the centre of gravity, but, in the mean time, satisfying prescribed stability criteria. Thus, for certain constraints, a design results to be more interesting and meritorious if more cargo can be stored at higher positions.

For a whole series of different operative conditions, the Strathclyde method allows us to highlight the stability quality of the ship in a new manner. This distinction cannot be done by applying other traditional stability criteria in as much as they are, in general, pertaining mainly to the so called "threshold type", i.e. they allow the designer to decide if the ship has to be considered stable or unstable according to the criterion. On the contrary, with the aid of the Strathclyde criterion, we are able to distinguish between more or less safe ships according to the percentage of net area positive during the ultimate rolls. For example, the results of Fig.4 suggest that ships with higher curves can satisfy the above design target better than ships with lower curves simply because to the same value of KG/D will correspond a greather net area. Bigger is the net area, more meritorious could be considered the ship.

This simple concept expresses both qualitatively and quantitatively our task and therefore can be assumed as a basis for the definition of a "stability rank" for the ship. In order to quantify the rank for each ship, the excess of net area has been computed for different KG/D conditions, i.e. for NA

Table 3 - Computation of the stability rank.

	A	A-A_{min}	S
BSRA I XF	57.080	2,020	1,894
BSRA I XG	58.980	3,920	2,735
BSRA I WO	61.490	6,430	3,845
BSRA I WP	64.545	9,485	5,197
BSRA I 907	75.400	20,340	10,000
BSRA I WS	60.135	5,075	3,246
BSRA I WR	59.445	4,385	2,940
BSRA I WQ	59.385	4,325	2,914
BSRA II ZP	57.415	2,355	2,042
BSRA II ZQ	56.680	1,620	1,717
BSRA II 851	57.825	2,765	2,223
BSRA II 852	55.060	0,000	1,000
BSRA II 975	60.215	5,155	3,281
BSRA II 977	59.350	4,290	2,898
BSRA II 978	59.035	3,975	2,759

ranging from 0% to 100%. In the (NA,KG/D) plane this corresponds to the area enclosed by the single curves and the horizontal axis. The result was then compared with respect to that of the ship with the lowest curve (Table 3). The excess of area is then divided by the maximum value of the sample and the results are finally normalized on a linear scale from 1.0 to 10.0 corresponding to the "worst" and to the "best" hull, respectively.

In Fig.5 we show the "stability rank" computed according to this rule for the vessels of the family. There is a considerable spreading of the ranks between single components, highlighting a difference in their relative merit in terms of stability. A parametric analysis carried out on the sample has shown that there exists a quasi-linear dependence between S and B/T ratio, to confirm the well known practical rule in designing larger ships in order to allow higher KG/D positions.

GLOBAL RANK

The specific ranks presented in this paper can be useful and significant for an optimization process only if they are not well correlated. To see the existing correlation, we show in Fig.6 the relationship between seakeeping and stability ranks. Simple calculations indicate the existence of an intermediate value for the correlation coefficient.

This fact suggests the possibility for *introducing higher level merit indexes which should* describe the global behaviour of the ship notwithstanding the requirement of other design goals. In this paper, in order to define a "global rank", we have considered as equally important the two targets of better dynamic behaviour in waves and higher allowable positions of centre of gravity. For this reason, starting from the two ranks R and S, the "global rank" G has been calculated by summing the above values and then normalizing according to the usual procedure. The obtained results represent a list of relative merit of the ships, see Fig.7.

CONCLUDING REMARKS

In traditional CAD optimization models the major shortcomings derive from inadequate consideration of stability and seakeeping. Thus, the effectiveness of the "global rank" here defined is to be evaluated in a more general, multicriterial optimization approach. The final goal is a completely new design methodology, oriented towards overcoming of the limits of the traditional "spiral" process [10].

Consequently, this same kind of work on "stability and seakeeping ranks" must be done for fishing vessels of different sizes, operating at various loading conditions and subject to different sea loads and environment. Future research will be directed towards developing a more extended "global rank", useful at conceptual and preliminary design stages respectively. Furthermore, speed loss in waves and lateral motions should be included as well.

Nevertheless, the model here developed allows the designer to make both a global and a detailed evaluation of the effect of the main geometrical hull parameters on stability and seakeeping, leaving him room for defining an "ad hoc" mathematical model by assigning proper weights to the calculated "specific ranks". Hence, he will achieve the "best" fishing vessel according to his particular requirements. The drawback of such an approach is that the model is defined empirically and intuitively in some way.

The reliability of the proposed model for early-design synthesis will be confirmed if an optimum hull can be designed. Clearly, a comparison of seakeeping and stability performance of the optimum hull must show its superiority with respect to the hulls stored in the data base. This problem is now under investigation[11].

NOMENCLATURE

A	Area in (NA,KG/D) plane
B	Breadth moulded of vessel
C_B	Block coefficient
C_P	Prismatic coefficient
D	Depth
Fn	Froude number
G	Global rank
GZ	Righting arm
$(H_w)_{1/3}$	Significant wave height
KG	Centre of gravity above keel
k_θ	Longitudial radius of gyration
L	Length between the perpendiculars
NA	Net area
R	Seakeeping rank
$r_{1/3}$	Significant amplitude of relative motion
S	Stability rank
t	Time
T	Draught amidships
T_W	Modal period
V_S	Ship speed
x_{CB}	Longitudinal centre of buoyancy
$z_{1/3}$	Significant amplitude of heave
ϕ	Roll angle
$\theta_{1/3}$	Significant amplitude of pitch
∇	Displacement volume

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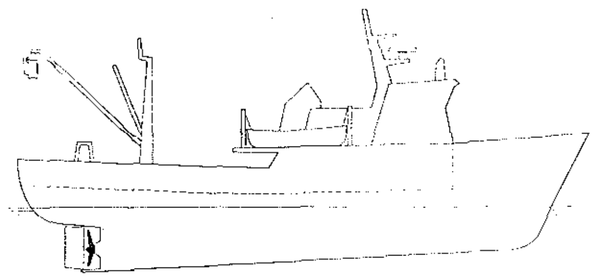

Fig.1 - Schematic rapresentation of the stern trawlers used for computations.

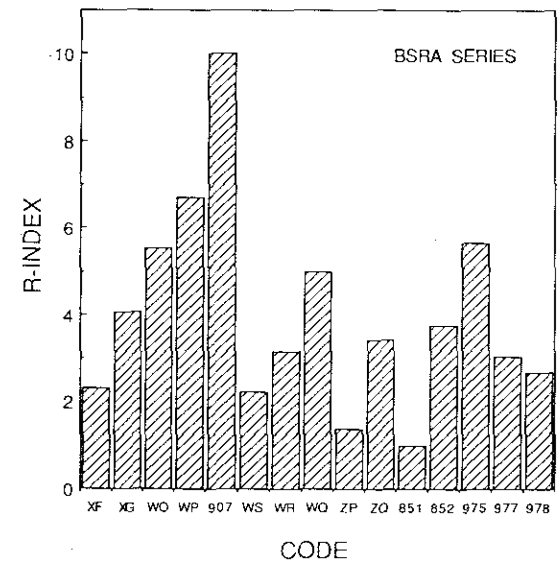

Fig.2 - Seakeeping rank for different vessels.

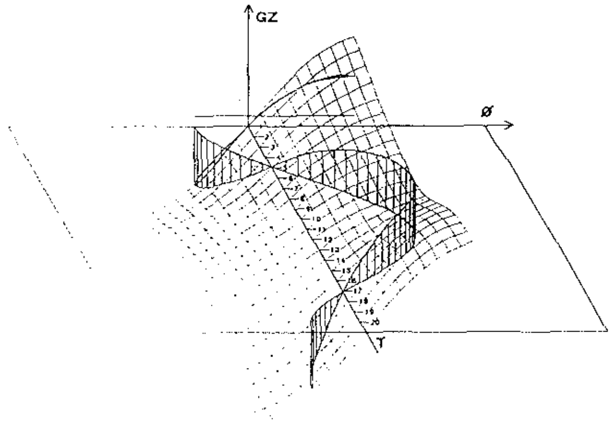

Fig.3 - The method of energetic balance of Strathclyde University.

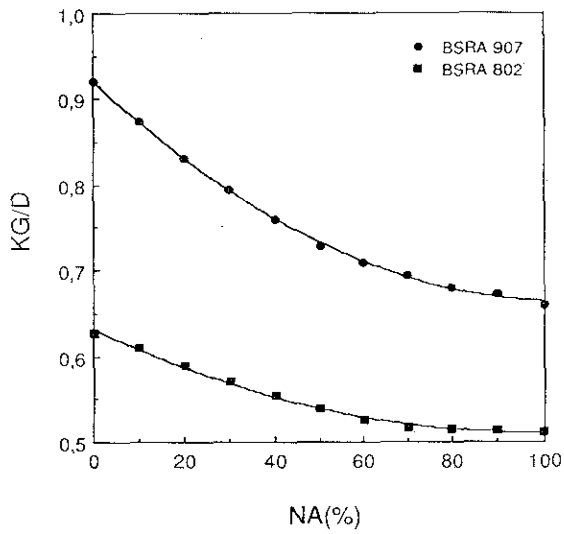

Fig.4 - KG/D versus NA for the best and the worst hull of the family.

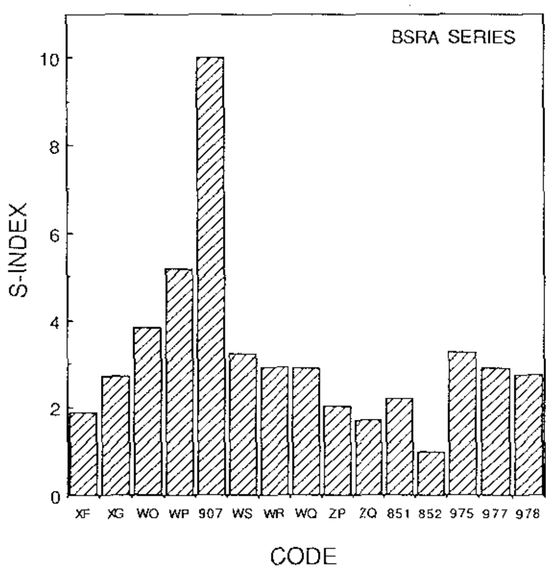

Fig.5 - Stability rank for different vessels.

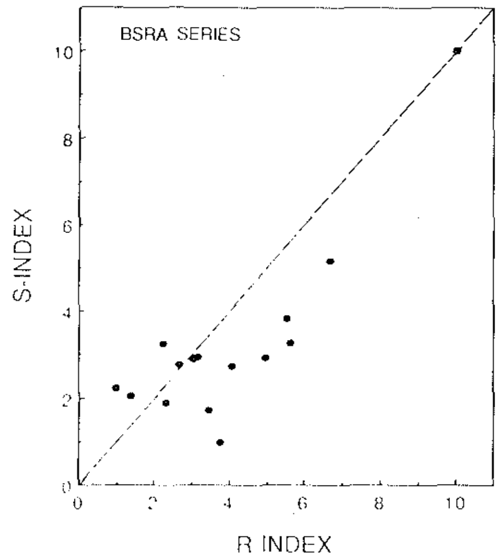

Fig.6 - Correlation between stability rank and seakeeping rank.

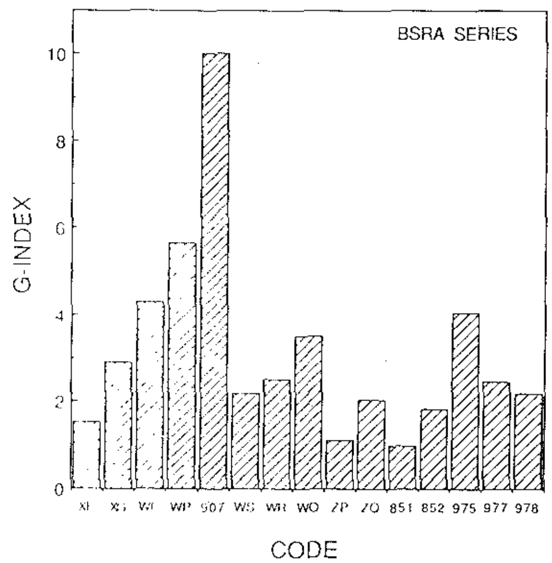

Fig.7 - Global rank for different vessels.

SOME LARGE ROLL MOTION SIMULATIONS USING MULTIPLE TIME SCALES

by Stephen B. Hodges

ABSTRACT

This work is concerned with some of the phenomena that may lead to unstable conditions for vessels that have an inherently low natural roll period, such as semisubmersibles. When such a vessel is subjected to a narrow-banded seastate, second-order rolling moments can be generated near the natural roll frequency by nonlinear interactions between neighboring frequency components of the incident wave. This can excite large amplitude rolling motions even for relatively small waves, a phenomenon which has been observed experimentally. A new method was developed for analyzng this situation where the roll motion was considered to be divided into two distinct components: one corresponding to the wave excitation frequencies with an amplitude small in the same sense as the incident waves are small, and a second component which is not restricted in amplitude, but is required to be slow in some sense associated with the bandwidth of the incident wave spectrum. A two-parameter perturbation scheme was used to derive the mathematical model. In the present work, this method is applied numerically in the time-domain to a shiplike shape subjected to wave groups consisting of two sinusoidal components with nearly the same frequency. Although the two-dimensional mathematical model cannot be compared directly with experimental results, some qualitative comparisons can be made. The character of the numerical results is quite similar to the experimental results.

INTRODUCTION

There are many factors which may contribute to instability and a potential capsizing situation. The relative importance of the parameters depends to a certain extent on the type of vessel and the nature of the seastate. For example, a ship in a following or quartering sea is susceptible to broaching, even if the waves are not extraordinarily high. For this reason, in severe conditions, ships will generally turn their bows to weather. For ships moored by a single point at the bow, this occurs naturally through "weather vaning". However, a moored vessel which is not free to turn into the weather may be subjected to waves directly on the beam. Vessels that fall in this category include offshore drill ships and semisubmersibles that use catenary mooring systems. Such vessels may remain on location for a short period of time as in the case of a drill ship, pipe lay vessel or diver support vessel, or as long as ten or twenty years for floating production systems. To avoid stability problems and to allow for safe working conditions, such vessels are usually designed to have a natural roll period that is much longer than any expected wave periods. For example, in the Gulf of Mexico, typical hurricanes and severe winter storms have most of their energy concentrated in periods below 18 seconds. Typical semisubmersibles designed to survive these conditions will have a natural roll period around 30 seconds or more so that there is no direct excitation of the natural mode. However, there are a number of nonlinear mechanisms which can excite a resonant response. There is very little damping in the roll mode so that a small excitation may excite a large roll response. These are some of the same mechanisms that excite large amplitude slow-drift sway motions of moored ships. The difference between roll and sway is that the restoring force in sway depends on the nonlinearity of the mooring system, which does not affect the hydrodynamics of the vessel itself except for a phase shift due to the large amplitude motions. The roll restoring moment

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is dependent on the shape of the wetted portion of the vessel which changes with roll angle and therefore changes the hydrodynamic characteristics.

This problem has been explored experimentally by several investigators. Takarada, *et al* [23] studied the general nonlinear motion characteristics of a semisubmersible and the effect of varying design parameters. The model was subjected to random wave and regular wave group excitations and large amplitude, slowly-varying sway and roll motions were observed. Khouri [14] conducted similar experiments on a simple generic, six column semisubmersible with the intention of examining the slow roll response. The results were similar to Takarada, *et al* [23]. Figure 1 shows a typical measured roll response from Khouri [14] when subjected to a regular wave group consisting of two regular waves with equal amplitude and slightly different frequencies. The difference in the frequencies was selected such that the envelope period of the wave groups corresponded to the natural roll period. Note that the roll response in figure 1 appears to have two distinct components, one with a relatively small amplitude at the carrier frequency and another with a much larger amplitude at the envelope frequency. Figure 2 shows a narrow-banded random wave excitation and the corresponding roll motion. Again, we see that there is a small amplitude wave frequency response superposed on a large, slowly varying motion. The mean period of the slow roll motion can be roughly related to the band width of the wave spectrum (*e.g.*, Tayfun and Lo [24]). This is very similar to the slow-drift sway motion of moored vessels, which has been extensively investigated (*e.g.*, see review article by Ogilvie [17]) and can arise theoretically from second-order terms involving the square of the first-order wave potential. This sort of behavior was observed throughout the tests, regardless of incident wave amplitude. So long as the envelope period was tuned to be close the natural roll period, low-frequency roll motions were observed with an amplitude considerably greater than the wave-frequency response.

These observations suggest that the roll motion of a vessel having a low natural roll frequency, implying low initial stability, subjected to a narrow-banded excitation can be separated into two distinct components: a small amplitude wave-frequency response superposed on a large amplitude, low-frequency response. Figure 3 shows the same random wave induced roll motion as figure 2 decomposed into low-frequency and high-frequency components. Notice how small the high-frequency motions are compared to the low-frequency motions. It can be further conjectured that if the large amplitude motions are slow enough (*e.g.* occur at a frequency much lower than the wave frequency range) then the small-amplitude wave-frequency motions can be treated by linear theory on the instantaneous wetted body. Since the large amplitude motions are relatively slow, the wetted geometry of the vessel will only change significantly over many cycles of the wave-frequency motion.

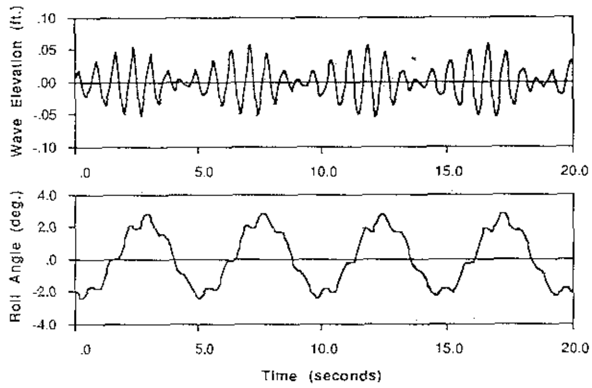

Figure 1 - Model semisubmersible roll response to two component wave groups (Khouri, [14]).

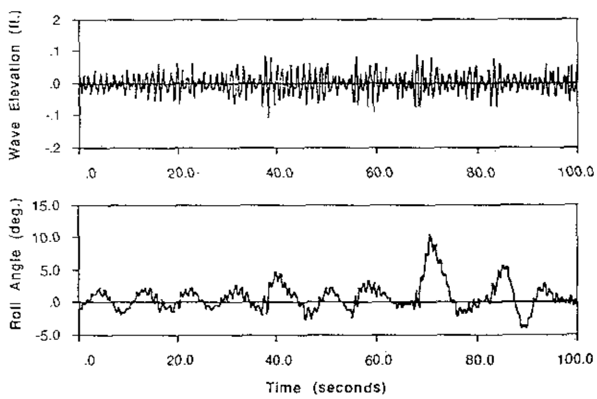

Figure 2 - Model semisubmersible roll response to narrow-banded random waves (Khouri, [14]).

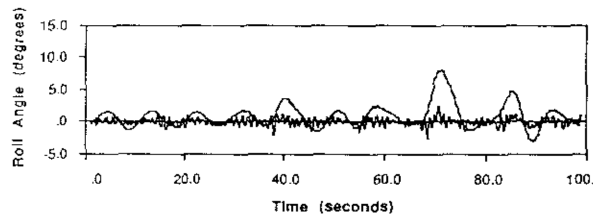

Figure 3 - Decomposition of model semisubmersible roll response to random waves into low and high frequency components.

Note that this separation does not apply when the excitation is broad-banded or the natural roll period is not significantly longer than the wave periods.

The problem of large amplitude roll motions is highly nonlinear because the changing geometry due to the large motions alters the hydrodynamic characteristics of the vessel. There has been quite a bit of work in the area of capsizing that utilizes linear theory or nonlinear extensions of linear theory (*e.g.* de Kat and Paulling [7], Oakley, *et al* [16], Pawlowski, *et al* [20]). Such methods are often very useful, primarily in examining the mechanisms of capsizing, but there is some question as to the validity of the actual motions at large amplitude. In the present theory, the nonlinear character is treated in a mathematically consistent manner, at the added expense of quite a bit more computing time. Some work has been done in the frequency domain to analyze the large motions problem (*e.g.* Dalzell [4,5], Papanikolaou [18], Papanikolaou and Nowacki [19], Roberts [21,22]). However, these methods generally consider only the original static wetted geometry with the nonlinear static restoring moments modelled by perturbation expansions about the initial geometry. These methods usually involve a Volterra series expansion to obtain second-order, quadratic transfer functions. These methods work well for sway motions but are not well suited for roll motions except in a global stability sense. The time domain is better suited to analyzing the details of large amplitude roll motions.

Small motions theory lends itself nicely to a perturbation approach where the motions are small in the same sense as the exciting wave. This is a very reasonable assumption for a well behaved, damped linear system. However, in the roll mode, most ship geometries exhibit very little damping, especially at low frequencies. For ships with a low GM, the natural frequency is very low and consequently the resonant response is sharply peaked. Although direct wave excitation is nonexistent at the natural frequency, nonlinear excitations generate a small second-order moment at low frequencies which can excite a resonant response. This suggests that perhaps a linear small motions theory is best suited for the wave frequency motions but not for the low frequency resonant response which may have an amplitude large enough that the changing geometry affects the wave frequency solution. However, if the geometry changes at a rate that is slow compared to the wave frequency motions, a number of wave cycles may pass before the geometry has changed enough to alter the solution.

This leads us to the proposition that we can solve a succession of linear problems to obtain the nonlinear, large amplitude roll motion for this special case. While this may be implemented in an intuitive manner, in order to maintain a measure of consistency with the physics of the problem, a perturbation approach is used. The mathematical development is rigorous and somewhat tedious but it leads to a mathematically consistent formulation whereby the errors in the formulation are related to the selection of the small perturbation parameter. This of course excludes errors due to the initial assumptions!

THEORY

The basis of this theory is the separation of the slow and fast components of the roll motion. This is accomplished by introducing two time scales: a "fast" time scale associated with the wave-frequency motions and a "slow" time scale associated with the low-frequency motions. This theory does not account for large, episodic waves in that the linear free-surface condition is retained. However, large amplitude slowly-varying roll motions may be excited even by relatively small waves if the frequency content is right. The same could be applied to the other degrees of freedom in a similar manner but for simplicity, we will concentrate on the roll motion. The mathematical details of the derivation are found in Hodges [10] and only a brief summary is given here.

The concept of multiple time scales is not new. Various forms of multiple time scale perturbation expansions have been used in many fields. Classic perturbation texts such as Kevorkian and Cole [13] and Nayfeh [15] include examples in several areas including orbital mechanics. A classic example involves Duffing's equation, which essentially models a simple oscillator with a spring stiffness that is cubic in displacement. Roll restoring moment acting on a ship may also be expressed in a power series to cubic order. However, the problem of ship motions is somewhat different in that, even in the linearized system, the mass and damping terms are frequency dependent. Thus, the problem must be treated in a slightly different manner.

In hydrodynamics, this technique has been examined by several researchers. Triantafyllou [25] used a multiple time scale technique in the context of control systems for dynamic positioning which are tuned to respond to low frequency lateral motions. Triantafyllou [26,27] and Triantafyllou and Bliek [28] used this same method to predict slow sway motions of moored vessels and mooring lines. Agnon and Mei [2] and Agnon, *et al* [1] used a similar procedure to analyze the swaying motion of wall and a floating cylinder. Zhou and Liu [33] also applied the concept to second-order diffraction forces on a vertical cylinder. However, none of these studies consider the roll motion where the geometry changes with time.

For simplicity, the present development will be restricted to a two-dimensional body in infinitely deep water, but there is no reason that it cannot be applied to the full three-dimensional problem in finite depth water. The basic assumptions of the theory are:

- Viscosity is ignored
- The flow is irrotational
- The wave spectrum is narrow-banded
- Initial stability is small (*e.g.* low natural roll frequency)
- Wave amplitudes and wave-frequency motions are small
- Low frequency motions are large but have small velocities

The first two assumptions lead to a potential theory formulation. The exclusion of viscosity is not necessarily accurate since the low-frequency roll damping of typical ship geometries is usually dominated by viscous damping. However, the difficulty of including viscous effects in the complete mathematical formulation is a formidable task. It has been shown that empirical formulas can be used to give a very good estimate of the viscous damping (*e.g.*, Himeno [9]). The next two assumptions are related in that the narrow-bandedness of the excitation leads to low frequency forces which excite resonant motions at the low natural roll frequency. The restrictions on the amplitudes and velocities of the motions lead to a perturbation approach. If we assume that the waves amplitudes are small, then a linear theory can be derived yielding small amplitude motions (*e.g.* John [11,12]). If we add the final assumption that the low frequency motions may have a large amplitude but are restricted to a small velocity, then the problem changes slightly.

The smallness of the wave amplitude and the fast motion amplitude can be characterized by a single parameter ε, related to the incident wave slope, as in John [11,12]. The smallness of the low frequency velocity can be characterized by another parameter δ which can be related to the ratio of the bandwidth of the wave spectrum to the mean frequency of the spectrum. It can be shown that for all practical purposes within the restrictions placed on this particular problem, we can let $\delta = \varepsilon$ without any loss of generality (Hodges [10]). We then introduce the slow time scale, $\tau = \varepsilon t$. If we let $\theta_f(t)$ be the wave-frequency "fast" roll motion and let $\theta_s(\tau)$ be the low frequency "slow" roll motion, then the total roll motion, θ, can be expressed as

$$\theta(t;\tau) = \theta_s(\tau) + \theta_f(t). \tag{1}$$

The wave-frequency motion is expanded in a power series in the small parameter ε.

$$\theta(t) = \theta_s(\tau) + \varepsilon\,\theta_{f1}(t) + \varepsilon^2\,\theta_{f2}(t) + \cdots. \tag{2}$$

Note that $\partial/\partial\tau = \varepsilon(\partial/\partial t)$. The angular velocity is then given by:

$$\theta_t(t) = \varepsilon\left[\theta_{s_\tau}(\tau) + \theta_{f1_t}(t)\right] + \varepsilon^2\,\theta_{f2_t}(t) + \cdots. \tag{3}$$

The subscript, t, indicates differentiation with respect to time. This confirms that, in the present theory, all motion velocities are small (*e.g.* $O(\varepsilon)$), regardless of amplitude. Other variables are expanded in a similar manner, including the velocity potential, Φ. The low-frequency "slow" potential can be shown to be zero for all practical cases (Hodges [10]) and so the total potential can be written as:

$$\Phi(x;t) = \varepsilon\,\Phi_{f1}(x;t) + \varepsilon^2\,\Phi_{f2}(x;t;\tau) + \cdots. \tag{4}$$

Note that the second-order potential may depend on the slow time scale, τ, as well as the fast time scale, t. In this work, the second-order potential will be ignored. The effect of the second-order potential cannot necessarily be ignored out of hand, but for the present purposes, the goal is to show that the second-order effects of the first-order potential are enough to induce large amplitude roll motions. Additional exciting moments due to the second-order potential may slightly alter the magnitude and phasing of the response, but the character should be very much the same.

By substituting these expansions into the governing equations, we arrive at a series of problems in ascending orders of ε. The zero-order problem requires that static equilibrium be satisfied in the vertical direction. The zero-order (large amplitude) roll motion is not restricted at this order as it is in small motion theory. The first-order equations look very much like the first-order small motions theory (*e.g.* Wehausen [30]) except that the wetted surface of the body, C_s, on which the potentials are evaluated depends on the slow time scale, τ. The second-order equations supply an equation of motion for the slow roll motion, θ_s driven by the second-order roll moment computed from the first-order potential, as shown in equation (5) below.

$$I_R\, \theta_{s_{\tau\tau}}(\tau) \;+\; C\, \theta_{s_\tau}(\tau) \;+\; K\, \theta_s(\tau) \;=\; M_2 \qquad (5)$$

where I_R is the rotational mass moment and C is artificial damping which may be selected from empirical data. K is the initial stiffness, $\rho g A_0 GM$, where ρ is the fluid density, g is the acceleration of gravity, A_0 is the submerged area at vertical static equilibrium and GM is the initial metacentric height. M_2 is the second-order moment based on the first-order potential solution.

SOLUTION IN THE TIME DOMAIN

The transient linear ship motions problem has been studied in a variety of contexts. Wehausen [29] derived a complete, three-dimensional small motions theory from the initial value problem in the time domain for a three-dimensional body. Yeung [31] gave a derivation of the unsteady time domain solution for a two-dimensional body. A similar derivation is used here. The problem is solved using a boundary integral formulation which requires a Green function to be evaluated on the body. This particular Green function was derived by Finkelstein [8]. Methods for computing the Green function and its derivatives can be found in Daoud [6] and Yeung [31]. The advantage of the boundary integral formulation, as opposed to a general field formulation, is that the free-surface and far-field boundary conditions are satisfied by the Green function, leaving only the wetted body boundary as the solution domain. Since this problem cannot be solved analytically, except for a few special geometric configurations, the body must be approximated by straight line segments. The potential is assumed to be constant over each segment and is evaluated at its center. This allows the integrals over the body surface to be converted to sums of integrals over the individual straight line segments. The accuracy of the method is related to the coarseness of the mesh, as discussed by Yeung [32].

Both the fast and slow solutions are advanced in time using a fourth-order Runge-Kutta method. For each slow time step, which for convenience may be an even multiple of the fast time step, vertical hydrostatic equilibrium is satisfied for the current slow roll position, which determines the wetted geometry. This may require some iteration, but it is a simple computation. The linear, small motions, fast time problem is then solved for this geometry for the number of fast time steps up to the next slow time step. The second-order moment is then computed from the first-order solution averaged over the slow time step and the second-order equation of motion for the slow roll motion is solved. This process is repeated for the four stages of the Runge-Kutta method.

Because large roll motions are allowed, the body geometry must be discretized above the mean free-surface as well as below. When the new geometry is determined, additional segments may become immersed or emerged changing the length of the segments bordering the free surface. If one segment is much smaller than the others, the problem will become poorly conditioned and large numerical errors may be introduced. To mitigate potential problems, the two panels nearest the mean free surface on either side of the body are modified to have the same length, preserving the enclosed wetted area.

RESULTS

To illustrate this theory, a simple, shiplike body was chosen. The offsets are defined by the equation $x = \pm[(D + y)/B]^{1/4}$, where B is the half beam and D is the draft. A ratio of B/D=2 was selected for this study. The body was discretized into straight line segments as shown in figure 4. The incident wave train, $\eta(x,t)$, consists of wave groups composed of two sinusoidal waves with equal amplitude, H/4, and radial frequencies ω_1 and ω_2, selected such that ω_1-ω_2 is very close to the natural roll period. The initial GM, and hence the vertical center of mass, was selected in conjunction with ω_1 and ω_2 such that $\omega_1-\omega_2$ is approximately equal to the natural frequency. The wave train is defined by:

$$\eta(x,t) = \frac{H}{2}\left[\cos(k_1x - \omega_1t) \; - \; \cos(k_2x - \omega_2t)\right]$$

$$= \frac{H}{2}\sin\left(\frac{\Delta k}{2}x - \frac{\Delta\omega}{2}t\right)\sin\left(Kx - \Omega t\right)$$

where $\Delta k = k_1 - k_2 \qquad \Delta\omega = \omega_1 - \omega_2$

$$\Omega = \frac{\omega_1 + \omega_2}{2} \qquad K = \frac{k_1 + k_2}{2}$$

Ω is the carrier frequency and $\Delta\omega$ is the envelope frequency of the wave groups, as shown in figure 1. Since infinitely deep water is assumed, the dispersion relation yields $k = \omega^2/g$.

The case examined here has a nondimensional angular mass moment of 1.0, a nondimensional wetted area of 2.95 and a nondimensional initial GM of 0.0095 giving a nondimensional initial stiffness of 0.028. The maximum crest elevation of the incident wave is 0.3 (H=0.6). For a vessel with a draft of 2.0, this is not an insignificant wave, but at the same time, it is not episodic. The second-order moment was computed from the first order-solution and the slow roll motion computed by solving equation (5). Figure 5 shows the incident wave, the second-order roll moment and the total motion. Note how the total roll motion clearly shows the slow and fast components. It should be noted here that numerical experimentation showed that the equation (5) was unstable for this large of a wave when no damping was included and the natural period was tuned closer to the envelope period. For smaller waves, this was not a problem. Since real fluid effects will cause significant roll damping, equation (5) includes

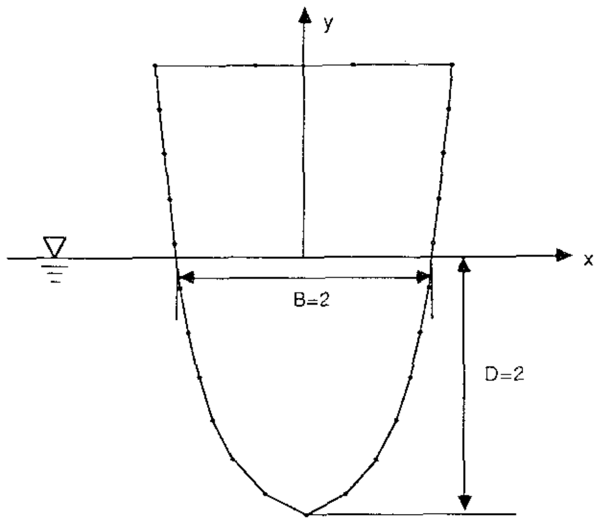

Figure 4 - Discretization of the body.

a linear damping term. In this case, a small amount of damping (10% of critical) was used. This seemed to eliminate any numerical instabilities. However, a more accurate result may be obtained with a detailed study of the viscous damping for this particular shape, but for the purposes of the present study, this is not very important. This is discussed a little further in the conlusions.

The roll motions predicted here have the same character as the experimental results of Khouri [14] shown in figure 1. This is encouraging; however, for the length of simulation shown here, the slow roll motion has clearly not reached a steady state amplitude. Unfortunately, these are very long simulations in terms of CPU time, so care must be taken as to which cases are run. Part of the expense is the fact that the two-dimensional convolution integrals used here decay very slowly with time compared with the corresponding three-dimensional functions. It would be useful to extend this method to three dimensions. Some CPU savings would be found from the truncation of the convolution integrals while additional expenses would be incurred due to more panels required to describe the body. Also, a three-dimesional version could analyze the semisubmersible models used by Takarada, *et al* [23] and Khouri [14].

CONCLUSIONS

In its present form, this theory is only of use in a very general sense. Its most salient characteristic is that it can predict large amplitude roll motions under certain circumstances without resorting to nonlinear free-surface conditions. Furthermore, it is derived in a mathematically consistent manner which allows for the quantification of errors. The drawbacks are that it is computationally intensive, although not nearly so much as other fully nonlinear theories. To be of some practical use, it must also be extended to three dimensions and an adequate viscous damping model should be incorporated.

Since the method used here is two-dimensional, it is difficult to find experimental results for comparison. The method can be easily extended to three-dimensions, although the number of discrete body panels would increase dramatically. The computation of the changing wet geometry would become more complicated but that is problematic. The first-order solution is readily available (*e.g.* Beck and Liapis [3]) and there would actually be a benefit in that the radial energy dissipation would make the convolution integrals decay faster and be better behaved. The issue of viscous damping is an important factor in roll motions. In the present theory, it is included only in a very approximate sense. This part of the problem can be partly addressed by empirical means. This method shows promise as a numerical calibration for other simpler models.

ACKNOWLEDGEMENTS

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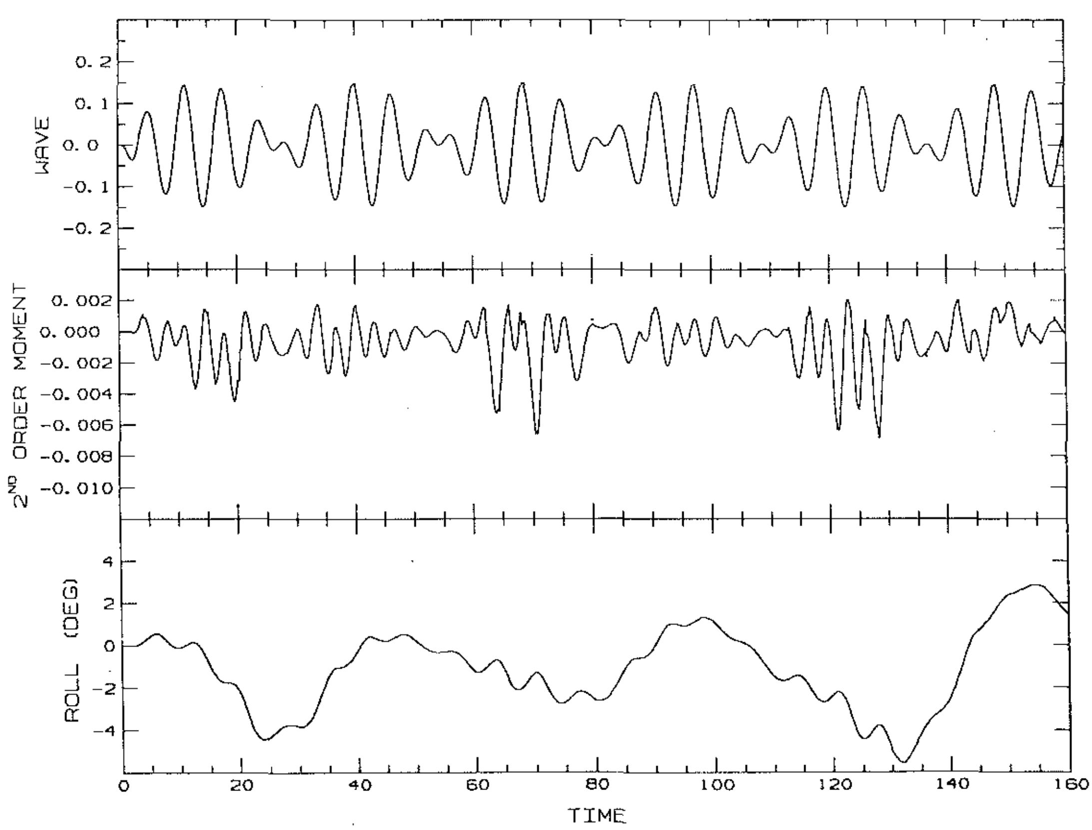

Figure 5 - Simulated results two component wave group excitation: incident wave, second-order moment, roll motion.

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THE TRANSIENT CAPSIZE DIAGRAM - A ROUTE TO SOUNDLY-BASED NEW STABILITY REGULATIONS

R.C.T.RAINEY*, J.M.T.THOMPSON**, G.W.TAM+ & P.G.NOBLE+

ABSTRACT

Recent developments in the theory of non-linear dynamics and chaos promise a breakthrough in the description of ship "capsizability". They suggest that ship behaviour in transient conditions offers a highly repeatable index of "capsizability", which is quick and simple to establish by physical model tests or (in principle) computer simulation. This index is the Transient Capsize Diagram. Using it, it should be possible, with the aid of suitable physical model tests and computer simulations, to resolve longstanding arguments about the allowance which existing regulations should make for roll damping, and for KG/draught ratio. For particularly critical vessels, where the cost of the model tests can be justified, the Diagram itself could be used as the basis of stability regulation, and as an operational aid. Looking further ahead, recent developments in computer simulation of vessel motions may allow the Diagram to be calculated with satisfactory accuracy during the design stage, which would greatly enhance its practicability as a regulatory tool.

1. THE CONCEPT OF THE TRANSIENT CAPSIZE DIAGRAM

The Transient Capsize Diagram, see Fig. 1 below, was originally proposed in [1] and is conceptually very simple. It merely records the waveheight above which a boat will capsize, as a function of wave period, and is determined by physical model tests or (in principle) computer simulation.

The novelty of the concept is entirely in the type of waves envisaged, which are groups of regular waves preceded by relatively calm conditions, so that the ship's roll motion is essentially transient. This makes the physical model tests (or computer simulations) quick and simple to perform - the ship either capsizes during the transient, or it does not - so that it is practical to cover the extensive range of wave heights and periods necessary to complete the Diagram. By contrast, the existing practice of testing in irregular waves requires very long runs to obtain statistically significant results, making it impractical to explore a comprehensive range of sea spectra (leaving aside the question of how such a comprehensive range of spectra might be defined - the problem is that standard families of sea spectra, e.g. Pierson-Moskowitz, generally have waves of insufficient steepness)

More fundamentally, the novelty of the concept is in the argument behind the use of transient conditions, and the reasons why it should give results which are both repeatable (despite inevitable variations in the initial conditions of the boat at the start of the transient) and more searching than equivalent regular or irregular wave conditions. These reasons spring from an important breakthrough in dynamic systems theory, due to the second author and his associates, notably M.S. Soliman, at University College London.

This work is described in a series of papers [2] - [5], recently summarised in [6]. The latest developments are given in the paper by M.S.Soliman in these proceedings. Essentially, it is shown that the extent of the "safe" initial conditions (i.e. the range of initial roll angles & roll velocities which do not lead to capsize in the transient conditions above) drops precipitously at a critical waveheight. Moreover, this drop makes practically all small roll angles and velocities "unsafe", so the critical waveheight can be readily found by examining transient motions starting from any such initial condition.

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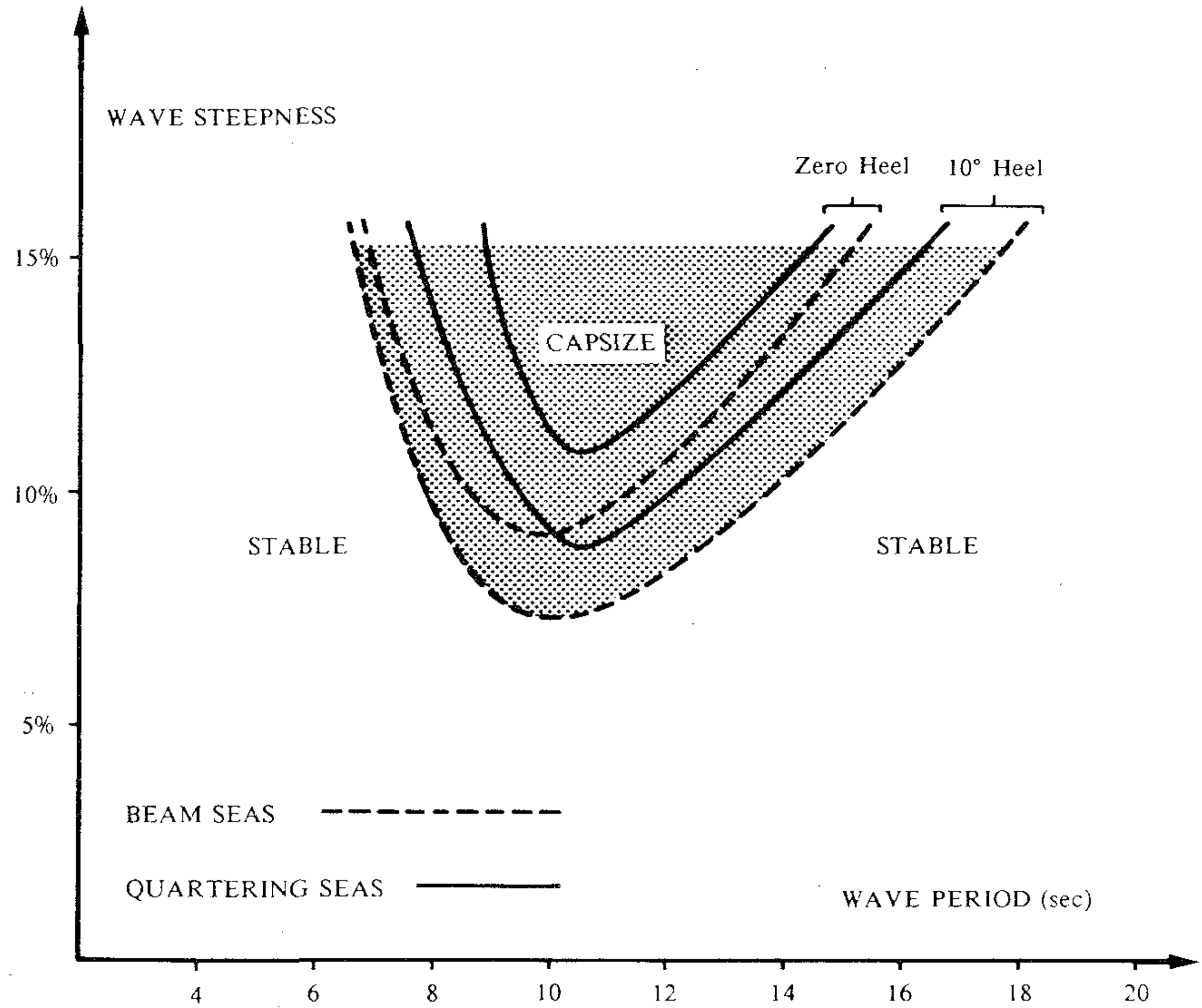

FIG. 1 TYPICAL TRANSIENT CAPSIZE DIAGRAM, FROM [1]

Fig. 2 below (taken from [4]) illustrates a simple idealised case (roll alone, parabolic GZ curve plus harmonic excitation, with 5% critical linearised damping). It shows the area of the "safe basin" (i.e. the area of non-capsizing initial conditions in the "phase space" of roll angle v. roll velocity), as a function of waveheight. The precipitous loss of basin area at a critical waveheight is evident. The small figures inset show the nature of the collapse in basin area - it is "from within", and covers the centre of the basin, corresponding to small initial roll angles and roll velocities.

Also evident in Fig. 2 is the fact that the basin area does not vanish altogether until the waveheight is increased by another 50% or so. This means that there remain a small range of initial conditions, corresponding to starting the roll motion in a near-transient-free manner, which will not lead to capsize. Thus model testing in long trains of regular waves of slowly-increasing height will tend to suppress capsizing, because it suppresses the transient. How much the waveheight can be increased beyond the critical value will be very sensitive to the extent of small disturbances during the test, so the procedure (which has been widely adopted hitherto) will evidently give results which are both non-conservative and erratic, compared with transient testing.

The final, and crucial, point is that the mechanism producing the sudden loss of basin area - chaotic transients from incursive fractals - appears quite general. It should therefore occur with mathematical roll models roll of realistic complexity, as well as the simplified case considered above. And it should occur in physical model tests.

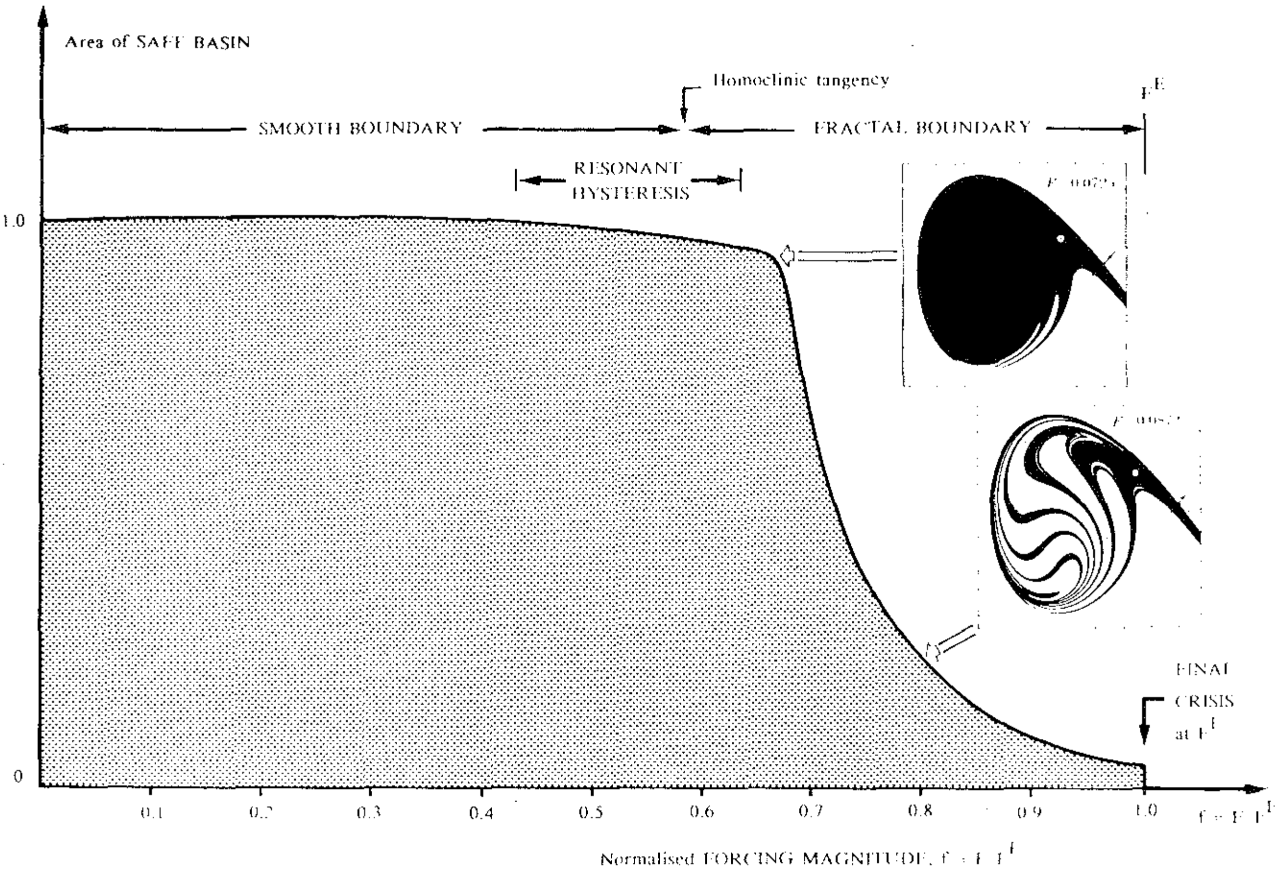

FIG. 2 AREA OF "SAFE BASIN", AS A FUNCTION OF WAVEHEIGHT, FROM [4]

2. USE OF THE TRANSIENT CAPSIZE DIAGRAM TO IMPROVE EXISTING STABILITY REGULATIONS

2.1 Provision for KG/draught Ratio

It has been conjectured for many years that that a ship with a high KG/draught ratio will be more prone to capsize than one which has the same GZ curve, but a lower KG/draught ratio. Beamy shallow-draught ships, in other words, which justify a high CG by having an even higher metacentre, are widely suspected to be less safe than narrow deep-draught ships, which achieve the same GZ curve by means of a low CG.

One of the strongest pieces of evidence for this view is the case of the Danish coastal tanker "Edith Terkol", which capsized when sailing in a shallow-draught condition, in which its GZ curve certainly met the IMO stability regulations, but in exactly the manner above (i.e. high CG but higher metacentre). This capsize is highly significant in that it proved possible to reproduce it repeatably in physical model tests [7].

Another piece of evidence is a more recent series of model tests commissioned by the US coastguard [8], which suggested that the existing IMO stability regulations (A167) are not valid for a KG/draught ratio above 1.4. More recent still is the capsize of the jack-up oil rig "Interocean 2" off Holland last year [9] - a jack-up rig under tow has an exceptionally high CG, offset by an exceptional beam and shallow draught.

Although the IMO have discussed modifications to the existing IMO regulation (A167) to allow for high KG/draught ratio [10], the issue remains controversial. The latest IMO proposal [11], for example, calls for wind heeling-moment calculations, that would implicitly penalise some (but not all) cases of high KG/draught ratio.

It is the purpose of this paper to suggest that the Transient Capsize Diagram offers a means of ending the controversy, by demonstrating the increased "capsizability" produced by high KG/draught ratio, in an unambiguous quantitative manner. Such Diagrams could clearly be constructed from physical models such as the "Edith Terkol" - if moreover the same Diagrams could be produced by computer simulation (see 4.2 below), the physical mechanism responsible could be elucidated, so that a conclusive case could be established for changing the existing regulations.

2.2 Provision for Roll Damping

Another longstanding and widely-supported conjecture is that a high level of roll damping makes ships less prone to capsize. Although the widely-adopted IMO stability regulations (A167) make no provision for roll damping, the latest IMO proposal [11] requires calculation of the "overall area of bilge keels", which is certainly a factor in roll damping.

Once again, however, no conclusive evidence is available of the extra safety margins involved, and the issue remains controversial. And once again, the purpose of this paper is to suggest that the Transient Capsize Diagram offers a means of ending the controversy, by systematically *demonstrating the quantitative importance* of roll damping, in both model tests and computer simulations. These could then form the basis of logical modifications to the existing stability regulations.

3. DIRECT USE OF THE TRANSIENT CAPSIZE DIAGRAM AS A NEW REGULATORY TOOL

Existing stability regulations, of course, are merely an attempt to characterise the "capsizability" of a ship in terms of its still-water righting lever (the GZ curve - in rare instances, e.g. [12], the righting-lever is also calculated by quasi-hydrostatic calculations in a following sea).

The Transient Capsize Diagram is manifestly a much more direct measure of "capsizability", because it measures actual dynamic capsize behaviour in waves. It could be argued, therefore, that this Diagram should not be used merely as a means of justifying modifications to the existing stability rules, but itself as the basis of stability regulation.

This type of regulation would simply state that a ship must have a Transient Capsize Diagram of certain specified minimum characteristics. This certainly has the attraction of simplicity and incontrovertibility. For example, it would probably highlight the capsizability of the "Edith Terkol" compared with the "Gaul" (which could not be capsized intact during model tests [13]) - a verdict to which it would then be hard to refuse assent.

Moreover, the minimum characteristics required of the Transient Capsize Diagram *could actually be calculated, on a* semi-scientific basis, by appealing to the desired Probability of Capsize, and the known statistics of ocean waves in the ship's operational theatre. This would bring capsize safety into line with other branches of marine safety, where quantitative risk analysis methodologies are followed.

Perhaps even more significantly, the dangerous combinations of ship speed and heading, and wave parameters, would be unambiguously highlighted; this would provide invaluable operational guidance, especially to inexperienced crews. The "Edith Terkol", for example, might well have been saved if the dangerous following sea condition that capsized her had been displayed (on her bridge, say!) in a Transient Capsize Diagram.

The only objection to such stability regulations is cost. This is because they require a series of model tests to be performed, rather than a simple hydrostatic calculation. Especially at the design stage, when many alternative configurations may be under investigation, this is a considerable penalty. It could perhaps most readily be borne in ships with a particularly high value or long production run, and where payload is at a great premium (e.g. warships or general-purpose trawlers, respectively).

For more general application, it appears necessary on economic grounds to consider the feasibility of calculating Transient Capsize Diagrams not by physical model tests, but by computer simulation. See 4.3 below.

4. THE RELEVANCE OF COMPUTER SIMULATION

4.1 A New Credibility for Computer Simulation, through Validation using the Transient Capsize Diagram

It is arguable that the major problem in computer simulations of ship capsize is no longer in the computers or their programs, but in demonstrating that the simulations are giving a faithful description of reality. For example, [14] describes one of the latest and most comprehensive simulation studies, and displays several hundred motion time-histories from simulations and experiments, from which it is not at all clear how faithful the simulation is, in general terms, in reproducing capsize. This is because the number of empirical parameters in the program is large (so that one-off agreement between theory and experiment can often be obtained by suitable choice of parameters), and the number of seastates under consideration is even larger (and not all of them are relevant to capsize).

One of the purposes of this paper is to point out that the Transient Capsize Diagram appears to offer a major step forward in this area: by summarising in a single figure the capsizing behaviour of a ship over its whole operational envelope, it appears to be an ideal means of displaying the veracity of a computer simulation, compared with a model test.

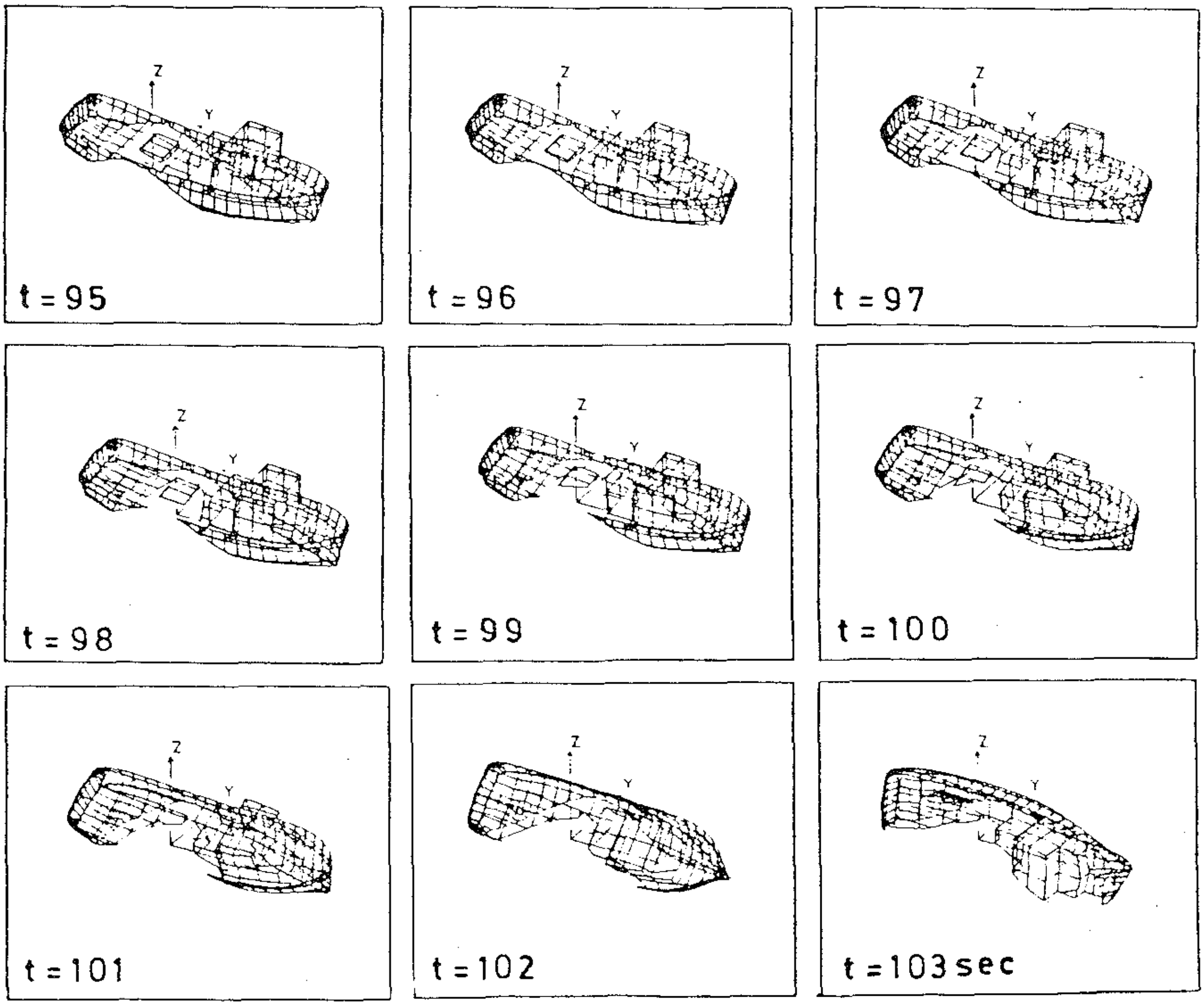

FIG. 3 "FROUDE-KRILOV-TYPE" COMPUTER SIMULATION OF FISHING BOAT CAPSIZE, FROM [16]

The problem of empirically-set parameters within the computer program, in particular, can be "brought into the open" by recomputing the complete Diagram (a task clearly well-suited to a computer graphics package linked to the simulation outputs) for a suitable range of all the relevant parameters.

It is perhaps therefore not fanciful to suggest that computer simulation of vessel capsize will in the next ten years gain considerable credibility, by this essentially pictorial means. And that there is an analogy with the theory of non-linear dynamics and chaos, which has gained considerable credibility in the last ten years, through similar computer-generated pictures (see e.g. [15])

4.2 Computer Simulation as a Means of Elucidating the Physical Mechanisms of Capsize

Having established the validity of the computer simulation, an extremely important use to which it can be put is to elucidate the physical mechanisms involved in capsize. This is possible because a computer simulation inevitably incorporates some idealisations and simplifications of the ship dynamics, which break down the forces acting into comprehensible elements. These individual forces can thus be displayed at each timestep, so that the critical ones can be identified.

For example, it should prove possible to identify in this way the physical mechanism linking the large KG/draught ratio of the "Edith Terkol" to its propensity to capsize. It has already been demonstrated [7] that the variation in quasi-static GM in various wave positions (i.e. variations calculated in the manner of [12]) are irrelevant. It is possible, however, that the heaving motion of the ship produced changes in instantaneous displacement which gave large GM variations, or that lateral hydrodynamic drag forces were to blame, as speculated in [7].

The truth or otherwise of all these explanations can be readily explored with a relatively simple "Froude-Krilov-type" computer simulation, such as that used in [16]. Fig. 3 above illustrates computer graphics from [16] - the computer program used (AQWANAUT from WS Atkins) simply integrates the water pressure in an undisturbed wave over the instantaneous wetted surface of the hull, combines it with an approximation to the hydrodynamic effects, and solves the ship's rigid-body equation of motion.

However it is accomplished, the goal of finding the physical mechanism responsible for a capsize is undoubtedly worthwhile in practical terms. It is the "smoking gun" which, if submitted in addition to the model-test evidence, alone appears capable of achieving change at the international, or even national, level.

4.3 Computer Simulations as a Regulatory Tool in their Own Right

The ultimate goal for computer simulation is to provide a direct replacement for physical model testing, thereby offering the speed and flexibility in calculation of Transient Capsize Diagrams, which alone makes them an economic general-purpose replacement for stability regulations in the present style.

This is however a much more challenging goal than merely identifying the main physical mechanisms involved in a particular capsize, as above. The reason is that the computer simulation is required to be general-purpose - for example correctly modelling all three of the canonical capsize mechanisms originally highlighted in [17], viz. "pure loss of stability in waves", "low cycle resonance" and "broaching", to which might perhaps be added "resonant rolling in beam seas" (as primarily envisaged in [1] - [6]). And the hydrodynamic mechanisms involved are very different - the first two being widely supposed to be largely quasi-hydrostatic problems, the third a matter of directional stability (which involves keel & rudder design, etc.), and the last a matter of roll damping among other things.

For a semisubmersible oil rig, the hydrostatic and hydrodynamic phenomena are greatly simplified because the effect of the free surface is relatively slight, and because the structural members can be treated as slender bodies. A rigorous slender-body approach [18] therefore looks promising for general-purpose applications, as argued in [19]. It is possible that a modification to this approach [20] will prove satisfactory for ships too: at least the rigorous methodology of [18] has the advantage that the final equations of motion are based on a consistent approximation scheme, and therefore lead to computer programs which can be debugged by comparisons with classical analytical solutions. This is in contrast to the semi-empirical approach of [14], in which a large number of separate and essentially arbitrary approximations are made, making the program very difficult to debug.

In any event, the challenge is clear - the feasibility of finding Transient Capsize Diagrams by computer simulation will only eventually be decided by trying it in practice.

5. CONCLUSIONS

By describing a ship's capsizing characteristics in waves over its whole operational envelope, the Transient Capsize Diagram appears capable of bringing a much-needed focus to stability research.

In particular, it should:

1) Clearly establish the capsize risks inherent in high KG/draught ratio and low roll damping.

2) Thereby give useful operational guidance on problem vessels, and pave the way for rational improvements in existing-style stability criteria

3) Provide a format for establishing the validity of computer simulations, enabling them to be used to discover the physical mechanisms responsible for capsize

4) Ultimately establish the validity of computer simulations sufficiently for them to be used directly as a means of "scientific" stability regulation.

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A REVIEW OF THE STABILITY CHARACTERISTICS OF SMALLER COMMERCIAL FISHING VESSELS OF THE UNITED STATES

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ABSTRACT

The commercial fishing fleet of the U.S. has been exempt from most maritime safety regulations. With the passage of the Commercial Fishing Vessel Safety Act of 1988 the U.S. Coast Guard is required to implement safety regulations, including stability regulations for commercial fishing vessels.

The vast majority of vessels in the fishing fleet are less than 24 meters in length. Creating appropriate stability standards for these vessels is difficult because there is little knowledge of the stability levels of the existing vessels.

This paper reviews the stability characteristics of eight smaller fishing vessels from the Pacific Northwestern portion of the U.S. Stability comparisons are drawn with the proposed energy criteria for a variety of loading conditions. Six of the eight vessels examined would not meet the proposed stability standards.

INTRODUCTION

The United States of America has traditionally been a nation heavily involved in maritime commerce. The history of fishing as an important profession dates from the earliest colonial period to today. Vessels engaged in commercial fishing range from small open dories operated by individual fishermen to large (over 100 m) catcher/processor vessels which carry a large crew of industrial workers in addition to the crew of fishermen.

Although the numbers can not be measured precisely, there are about 30,000 documented commercial fishing vessels. These are vessels admeasured at over 5 net tons, and they are required to register with the U.S. Coast Guard. Smaller commercial fishing vessels are registered with the individual states and their number is estimated to be about 80,000.

Casualty statistics for the commercial fishing fleet are kept by the U.S. Coast Guard along with those for all merchant vessels. A review of these statistics indicates that they are probably quite accurate in recording deaths and the total loss of larger vessels. They are less accurate for damage to smaller vessels (which also may be under the minimum reporting threshold of $25,000 in damage) and do not cover the more common injuries experienced by fishermen.

Available casualty data indicate that on the average there are over 200 vessel total losses and over 100 lives lost each year in the U.S. commercial fishing industry.

For most of our history, commercial fishing vessels have been exempt from most safety regulations applied to the merchant fleet. Because of the large numbers of losses and the errors uncovered in major casualty investigations [1], the Congress experienced increasing pressure to bring the commercial fishing fleet under government regulation.

Passage of the Commercial Fishing Vessel Safety Act of 1988 (Public Law 100-424) will eventually impose a number of safety related regulations on commercial fishing vessels. This law requires the U.S. Coast Guard to draft safety regulations and eventually to enforce these regulations. Proposed regulations were issued in April 1990 with an open comment period through August 1990. Following this comment period, final regulations will be drafted and issued after January 1991.

The drafting of proposed regulations and the call for comments on these proposed regulations has stimulated a great deal of discussion of the

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entire range of safety concerns. One of the most important areas of vessel design, construction and operation, is the area covered by the proposed vessel stability regulations. This has been a very controversial subject with a variety of opinions expressed. Unfortunately, there is also very little collected information on the current stability levels within the U.S. commercial fishing fleet, especially information concerning vessels smaller than 24 meters in length.

In this paper the proposed regulations are summarized and some examples of smaller vessels are examined to show their operating stability.

PROPOSED FISHING VESSEL STABILITY REGULATIONS

The proposed rules were published April 19, 1990 [2] and contain a number of safety aspects. Subpart E details the proposed fishing vessel stability regulations. These rules will apply to vessels built or substantially altered after the effective date of the regulations.

There are some proposed simplified tests which may apply to very small vessels. In addition, there are guidelines on conducting the inclining test, dealing with free-surface effect in the tanks and applicable icing conditions. The required sizes of freeing ports are specified as well as the type of watertight closures to be employed above the main deck.

The major proposed requirements are summarized in the following sections.

Intact Stability When Using Lifting Gear

In this calculation the difference between the righting arm curve and the heeling arm curve is used. The area of the difference curve is calculated between the angle of equilibrium and the least of the angle of maximum righting arm, the angle of downflooding, or 40 degrees. This area must be at least 0.46 meter-degrees (15 foot-degrees).

Water on Deck

The water on deck criterion requires drawing a heeling arm curve caused by the water on deck on the same figure as the righting arm curve. Area *a* is the area above the righting arm curve but below the heeling arm curve. Area *b* is the area under the righting arm curve from the angle where the two curves intersect up to 40 degrees or the angle of downflooding, whichever is less. It is required that area *b* is greater than or equal to area *a*.

Intact Righting Energy

This is essentially the IMO criteria contained in resolution A.168 with one addition. The requirements are:

(1) An initial metacentric height of at least 0.35 meters (1.15 feet).

(2) A righting arm of at least 0.20 meters (0.66 feet) at an angle of heel not less than 30 degrees.

(3) A maximum righting arm that occurs at an angle of heel not less than 25 degrees.

(4) An area under each righting arm curve of at least 0.09 meter-radians (16.9 foot-degrees) up to the lesser of 40 degrees or the angle of downflooding.

(5) An area under each righting arm curve of at least 0.055 meter-radians (10.3 foot-degrees) up to an angle of heel of 30 degrees.

(6) An area under each righting arm curve of at least 0.03 meter-radians between 30 degrees and the lesser of 40 degrees or the angle of downflooding.

(7) Positive righting arm through an angle of heel of 60 degrees.

Severe Wind and Roll

In this calculation a gust heeling arm is calculated based on the profile of the vessel. This is plotted on the righting arm curve and there are requirements on the angle of equilibrium and various areas defined by the two curves.

Unintentional Flooding

This criteria applies only to newly constructed vessels over 12.2 meters (40 feet) in length. The vessel must survive flooding of any individual compartment in any loading condition.

DESCRIPTION OF VESSELS

For this study eight smaller fishing vessels from the Pacific Northwest of the United States were examined. Complete stability tests were performed and the vessels stability characteristics in various operating conditions were calculated.

The most important fishery in this region is the salmon fishery. All of the vessels examined were originally designed principally as salmon seining vessels. Some of the vessels are intended for combinatión service and one vessel no longer fishes for salmon. As salmon seiners, these vessels are restricted in size by a fisheries conservation regulation imposed by the State of Alaska. The overall length of a salmon seiner in Alaska must be 17.68 meters (58 feet) or less.

Eight vessels are included in this report and are assigned numbers from 1 to 8 to identify them. These vessels are arranged in order of increasing length with vessel number 1 the smallest. Table 1 lists the basic characteristics of these fishing vessels.

Vessels Number 1 and 2

The hulls for these two fiberglass vessels come from the same mold. They have round bilges and are shallow draft salmon seiners operating in the Prince William Sound area of Alaska. Both vessels have circulating refrigerated seawater systems for their hold, but the arrangements are a little different. Vessel 1 has an enclosed steering station above the house while Vessel 2 has an open steering station.

Although these two vessels are similar, the lightship metacentric heights differ by 0.29 meters and Vessel 1 with the higher house has greater metacentric height. There is no clear explanation for this difference.

Vessel Number 3

Vessel number 3 is a hard chined, fiberglass salmon seiner which operates in Prince William Sound. It is of a somewhat older style with a narrower beam and a deeper hull form than newer vessels in the same size range.

Vessel Number 4

Vessel number 4 represents a modern fiberglass salmon seine vessel which operates in Prince William Sound. It is a shallow draft hull with rounded bilges and a partial tunnel enclosing a single screw. Since this vessel was built the trend has been toward even longer and beamier vessels which provide a platform for use in multiple fisheries.

Vessel Number 5

This vessel is an older traditional single chined steel vessel which has recently been reconstructed. It is a relatively narrow vessel which has been operated safely for twelve years despite having the smallest metacentric height of all the vessels tested.

Vessel Number 6

Vessel number 6 is a newer style high-speed fiberglass seiner/crabber. It can be used for the salmon or herring seine fisheries or operated as a dungeness crabber carrying crab pots. The vessel is designed for 20 knot speed and has twin screws. The high speed is necessary to be able to transit large distances and arrive in time for the short local openings during the Alaskan herring season.

The hull form has rounded bilges and a larger length to beam ratio than comparable new steel vessels. It has a forecastle deck forward with a house above.

Vessel Number 7

Vessel number 7 is a typical newer "Alaskan limit seiner" having the smallest length/beam ratio of all the eight vessels. Constrained by the length limitation, these vessels have increased in beam. While salmon seining remains their principal fishery, the economics of operating a successful fishing vessel have driven them to become a platform for a variety of other operations including crabbing and tendering. This particular vessel seines in both Alaska and Puget Sound and fishes for king crab in Alaska.

Vessel number 7 is a hard-chined steel vessel with a forecastle and a house above.

Vessel Number 8

Vessel number 8 is an early "Alaskan limit seiner" and represents a vessel designed for seining. It has less beam than newer limit seiners and does not have a forecastle. Over the years it has been used in a variety of fisheries until the present. It is no longer used as a seiner.

Although vessels 7 and 8 are the only limit seiners included in this study, they represent the extremes of older vessels and newer vessels. In numbers, the limit seiner, represents a large percentage of the total fleet.

COMPARISON WITH INTACT ENERGY CRITERIA

While a variety of stability criteria have been included in the proposed stability regulations for the U.S. fishing fleet, it is expected that the energy criteria will form the basic core of the final regulations. In order to examine the impact of the energy criteria on designers and to evaluate their applicability to smaller vessels in the U.S. fishing fleet, the stability of the eight vessels was evaluated for a variety of loading conditions.

It should be noted that the eight vessels included in this study were not deliberately selected to represent the entire fleet. Participation was purely voluntary. Consequently, there may be some bias toward more stable vessels in this sample. Owners of less capable vessels tend to avoid participation in a program of voluntary stability testing.

Salmon Seining

One of the most important developments in recent years in the salmon industry has been the treatment of the fish after they are caught. There has been a strong emphasis on high quality and a bonus paid for fish kept in refrigerated seawater as opposed to the traditional iced condition. As a result there has been an almost universal conversion to holds equipped with refrigerated seawater systems. This leads to a possible departure condition with the hold filled with seawater or empty. Even those vessels that depart with the hold empty generally do not travel long distances before fishing commences. These vessels then would also be filling their holds in a condition very near the departure condition.

Tables 2, 3, 4, 5, and 6, are a comparison of the salmon seining vessels' actual stability with the proposed intact energy criteria in a variety of different loading conditions. In these tables, except for the measurements in degrees, the actual values for the vessels are divided by the required value. In these rows an entry of 1.00 or greater implies that the requirement is met.

Crabbing

Dungeness or king crabbing are fisheries only found on the west coast of the United States. In many cases stability reports have been generated for vessels engaged in these fisheries because insurance companies insisted on them after sustaining considerable losses. Lacking any other standard, the IMO energy criteria has generally been used for all vessels engaged in crabbing.

Typically these reports are prepared by applying the appropriate loading to the vessel and then adding crab pots until adding one more pot results in the vessel not meeting one of the criteria.

For king crabbing very large pots are used while a much smaller, lighter pot is used for dungeness crabbing. There is a gear limit of 300 pots which can be used by a vessel for dungeness crabbing. For these vessels it is convenient to be able to carry 300 pots at once.

Tables 7 and 8 show the king crabbing operations for vessels 7 and 8. Tables 9 and 10 show the dungeness crabbing operation for vessels 5 and 6. The "restricted departure" condition provides for reduced fuel and water in order to maximize pot carrying capacity.

Trawling

Table 11 indicatess the stability of vessel number 8 when engaged in the trawling.

Summary

Table 12 is a summary of the previous tables indicating which existing vessels would meet or fail the proposed energy criteria.

DISCUSSION AND CONCLUSIONS

Examining the comparisons of the stability of the vessels included in this study to the proposed energy criteria reveals that six out of the eight vessels examined would not meet these criteria for specified loading conditions. The criterion that seemed to pose the most difficulty was the 0.20 meter righting arm required at an angle of heel of 30 degrees or greater.

Table 12 reveals that the departure condition with the hold filled with seawater is a particularly severe condition. Unfortunately, with the advent of circulating or refrigerated seawater in the hold this is a more likely condition of loading.

There have been many discussions among naval architects in the United States about the requirement that the maximum of the righting arm curve be at an angle of heel greater than 25 degrees. Many would like to eliminate this requirement. The problem is illustrated in Table 15 for vessel number 6 while dungeness crabbing. In the hold empty condition it does not meet the proposed stability criteria because the peak of the righting arm curve is at an angle of heel of less than 25 degrees. All the other criteria are exceeded by a wide margin. The objection arises because naval architects feel that a vessel meeting all the other criteria would have an adequate margin of stability.

It is ironic that vessel number 5 has stability characteristics which are well below the proposed energy criteria requirements. This vessel typifies a large number of older vessels which have been operating successfully for a very long time in the U.S. fishing fleet. Vessel number 5 has been operating for 12 years without any problems. Her owner describes her as an excellent sea boat which is very stable. The owner is also a very experienced operator. Perhaps it is as important to examine who is operating our fishing vessels and what training and experience they possess as it is to regulate the vessels.

A comparison of fiberglass and steel vessels is very enlightening. Clearly, the fiberglass vessels tend to be considerably lighter. As a result they can be built with less beam and yet achieve greater statical stability. The fiberglass vessels have a larger metacentric height which results from the fact that they have very low displacements and a large metacentric radius.

In seeking appropriate stability criteria for smaller U.S. fishing vessels we need considerably more study. A continuing effort to assess the current fleet with emphasis on some of the older vessels which have been operating over twenty years would be valuable.

It would also be very valuable to carefully examine the capsizing cases to find the conditions under which smaller vessels capsize and to establish the vessels' stability level at the time of loss.

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Table 1. Physical Properties of Eight Commercial Fishing Vessels.

Vessel No.	1	2	3	4	5	6	7	8
Length (deck) [m]	12.51	12.51	12.73	13.23	16.31	16.62	17.58	17.63
Beam (max @ deck) [m]	4.42	4.42	3.96	4.47	4.88	5.18	6.30	5.54
Depth [ft] main deck to bottom	1.35	1.35	1.68	1.62	2.92	1.85	2.84	2.80
Length/Beam	2.83	2.83	3.21	2.96	3.34	3.21	2.79	3.18
Year built	1982	1982	1979	1985	1978	1989	1982	1974
Hull material	F	F	F	F	S	F	S	S
Fisheries	SS	SS	SS	SS	SS DC	SS DC	SS KC	ST KC
Lightship								
Displacement [tonnes]	17.20	17.27	13.15	15.76	54.66	27.81	76.30	77.58
Metacentric Height [m]	2.112	1.836	0.826	1.76	0.631	2.387	10.735	0.622
$\frac{\text{Displaced Volume}}{\text{Length}^3}$	8.57×10^{-3}	8.61×10^{-3}	6.22×10^{-3}	6.64×10^{-3}	12.3×10^{-3}	5.90×10^{-3}	13.7×10^{-3}	13.8×10^{-3}
Hold (seawater)								
Fwd [tonnes]	14.94	15.43	13.77	16.30	30.42	19.89	41.64	29.79
Aft [tonnes]	—	—	—	—	4.24	11.17	16.58	—
Consumables								
Fuel capacity [liter]	3733	3733	2604	3294	15,000	8,504	29,774	12,823
Fresh water capacity [liter]	757	757	379	628	2,067	1,113	2,124	1,514

F = Fiberglass; S = Steel; SS = Salmon Seining; ST = Stern Trawling; DC = Dungeness Crabbing; KC = King Crabbing

Table 2. Comparison of Fishing Vessels with Intact Righting Energy Criteria, Salmon Seining Vessels Departure Condition — Holds Empty

Vessel No.	1	2	3	4	5	5	6	7	7
Fishing Region	Alaska	Alaska	Alaska	Alaska	Alaska	Puget Sound	Alaska	Alaska	Puget Sound
Initial Metacentric Height / Required Metacentric Height	4.08	3.84	1.39	4.28	1.14	—	3.99	2.11	1.73
Righting Arm @ 30° or Greater / Required Righting Arm	2.24	2.03	1.03	2.11	0.27	—	2.27	1.41	0.77
Angle of Max Righting Arm (degrees)	25.0	25.0	35.0	25.0	17.5	—	25.0	30.0	25.0
Area 0°–40° / Required Area	2.74	2.53	1.18	2.56	0.46	—	2.80	1.59	1.00
Area 0°–30° / Required Area	3.06	2.86	1.12	2.86	0.59	—	2.25	1.71	1.15
Area 30°–40° / Required Area	2.60	2.35	1.18	2.43	0.31	—	2.67	1.64	0.89
Angle of Vanishing Stability (degrees)	>60	>60	>60	>60	39	—	>60	>60	55

Table 3. Comparison of Fishing Vessels with Intact Righting Energy Criteria, Salmon Seining Vessels Departure Condition — One Hold Filled with Seawater

Vessel No.	1	2	3	4	5	5	6	7	7
Fishing Region	Alaska	Alaska	Alaska	Alaska	Alaska	Puget Sound	Alaska	Alaska	Puget Sound
Initial Metacentric Height / Required Metacentric Height	3.09	2.96	1.28	2.83	1.16	—	3.45	2.67	2.37
Righting Arm @ 30° or Greater / Required Righting Arm	0.82	0.71	0.95	1.55	0.73	—	2.03	1.42	0.53
Angle of Max Righting Arm (degrees)	17.5	17.5	30.0	25.0	25.0	—	30.0	40.0	40.0
Area 0°–40° / Required Area	1.12	1.01	1.06	1.84	0.85	—	2.36	1.37	060
Area 0°–30° / Required Area	1.32	1.20	1.13	2.04	0.93	—	2.57	1.46	0.67
Area 30°–40° / Required Area	0.95	0.83	1.11	1.77	0.84	—	2.37	1.43	0.55
Angle of Vanishing Stability (degrees)	>60	>60	>60	>60	51	—	>60	>60	>60

Table 4. Comparison of Fishing Vessels with Intact Righting Energy Criteria, Salmon Seining Vessels Fishing Condition — One Hold Filled with Seawater

Vessel No.	1	2	3	4	5	5	6	7	7
Fishing Region	Alaska	Alaska	Alaska	Alaska	Alaska	Puget Sound	Alaska	Alaska	Puget Sound
Initial Metacentric Height / Required Metacentric Height	3.13	3.01	1.20	4.57	1.55	—	4.27	—	2.43
Righting Arm @ 30° or Greater / Required Righting Arm	0.95	0.85	0.95	1.61	0.44	—	2.91	—	1.62
Angle of Max Righting Arm (degrees)	17.5	17.5	30.0	25.0	22.5	—	40.0	—	40.0
Area 0°–40° / Required Area	1.28	1.16	1.04	1.92	0.60	—	3.18	—	1.54
Area 0°–30° / Required Area	1.49	1.37	1.10	2.11	0.70	—	3.43	—	1.63
Area 30°–40° / Required Area	1.11	0.98	1.10	1.87	0.51	—	3.26	—	1.65
Angle of Vanishing Stability (degrees)	>60	>60	>60	>60	>60	—	>60	—	>60

Table 5. Comparison of Fishing Vessels with Intact Righting Energy Criteria, Salmon Seining Vessels Burned Out Condition, No Holds Filled

Vessel No.	1	2	3	4	5	5	6	7	7
Fishing Region	Alaska	Alaska	Alaska	Alaska	Alaska	Puget Sound	Alaska	Alaska	Puget Sound
Initial Metacentric Height / Required Metacentric Height	4.64	4.36	1.54	4.70	1.04	1.19	4.90	1.93	1.57
Righting Arm @ 30° or Greater / Required Righting Arm	2.52	2.27	0.70	1.79	0.53	0.74	2.23	1.26	0.91
Angle of Max Righting Arm (degrees)	27.5	25.0	30.0	25.0	22.5	25.0	27.5	30.0	25.0
Area 0°–40° / Required Area	3.01	2.78	0.86	2.26	0.71	0.89	2.76	1.39	1.08
Area 0°–30° / Required Area	3.33	3.10	0.97	2.56	0.83	0.99	3.11	1.49	1.20
Area 30°–40° / Required Area	2.91	2.64	0.81	2.07	0.61	0.85	2.59	1.46	1.05
Angle of Vanishing Stability (degrees)	>60	>60	>60	>60	42	48	>60	>60	57

Table 6. Comparison of Fishing Vessels with Intact Righting Energy Criteria, Salmon Seining Vessels, Burned Out Condition, One Hold Filled with Seawater

Vessel No.	1	2	3	4	5	5	6	7	7
Fishing Region	Alaska	Alaska	Alaska	Alaska	Alaska	Puget Sound	Alaska	Alaska	Puget Sound
Initial Metacentric Height / Required Metacentric Height	3.26	3.13	1.23	4.58	1.35	1.41	3.48	2.62	2.27
Righting Arm @ 30° or Greater / Required Righting Arm	1.39	1.27	1.12	1.88	0.03	0.36	1.95	2.12	1.36
Angle of Max Righting Arm (degrees)	25.0	22.5	30.0	27.5	10.0	20.0	30.0	40.0	40.0
Area 0°–40° / Required Area	1.76	1.68	1.17	2.18	0.17	0.54	2.30	1.92	1.39
Area 0°–30° / Required Area	2.00	1.94	1.21	2.37	0.26	0.64	2.52	1.98	1.48
Area 30°–40° / Required Area	1.62	1.50	1.30	2.18	0.40	0.43	2.29	2.12	1.44
Angle of Vanishing Stability (degrees)	>60	>60	>60	>60	31	>60	>60	>60	>60

Table 7. Comparison of a Fishing Vessel with Intact Righting Energy Criteria, Vessel No. 7 King Crabbing (pots are 205 kilograms each)

Load Condition	Departure Full	Departure Full	Departure Restricted	Departure Restricted	Burned Out	Burned Out
Fuel % of Capacity	100	100	50	50	10	10
Water % of Capacity	100	100	100	100	10	10
No. Crab Pots (on deck)	40	38	47	60	35	60
Holds Filled with Seawater	0	1	0	1	0	1
Initial Metacentric Height / Required Metacentric Height	1.81	2.47	1.58	2.14	1.63	1.95
Righing Arm @ 30° or Greater / Required Righting Arm	1.02	1.03	1.03	1.17	1.02	1.18
Angle of Maximum Righting Arm (degrees)	30.0	40.0	35.0	40.0	30.0	40.0
Area 0°–40° / Required Area	1.20	1.03	1.14	1.16	1.15	1.20
Area 0°–30° / Required Area	1.32	1.12	1.23	1.23	1.25	1.28
Area 30°–40° / Required Area	1.17	1.05	1.18	1.23	1.18	1.26
Angle of Vanishing Stability	>60	>60	>60	>60	>60	>60

Table 8. Comparison of a Fishing Vessel with Intact Righting Energy Criteria, Vessel No. 8
King Crabbing (pots are 205 kilograms each)

Load Condition	Departure Full	Departure Full	Departure Restricted	Departure Restricted	Burned Out	Burned Out
Fuel % of Capacity	100	100	65	65	10	10
Water % of Capacity	100	100	50	50	10	10
No. Crab Pots (on deck)	18	0	24	18	23	14
Holds Filled with Seawater	0	1	0	1	0	1
Initial Metacentric Height / Required Metacentric Height	1.68	2.29	1.61	2.15	1.46	2.10
Righing Arm @ 30° or Greater / Required Righting Arm	1.00	1.23	1.02	1.00	1.00	1.08
Angle of Maximum Righting Arm (degrees)	27.5	60.0	27.5	60.0	27.5	60.0
Area 0°–40° / Required Area	1.17	1.18	1.17	1.07	1.13	1.21
Area 0°–30° / Required Area	1.27	1.30	1.27	1.19	1.22	1.33
Area 30°–40° / Required Area	1.16	1.15	1.18	1.01	1.16	1.19
Angle of Vanishing Stability	>60	>60	>60	>60	>60	>60

Table 9. Comparison of a Fishing Vessel with
Intact Righting, Energy Criteria , Vessel No. 5
Dungeness Crabbing , (pots are 45.4 kilograms each)

Load Condition	Departure Restricted	Departure Restricted
Fuel % of Capacity	34	34
Water % of Capacity	100	100
No. Crab Pots (in holds) / No. Crab Pots (on deck)	141 / 67	0 / 0
Holds Filled with Seawater	0	1
Initial Metacentric Height / Required Metacentric Height	1.44	1.75
Righing Arm @ 30° or Greater / Required Righting Arm	1.01	0.95
Angle of Maximum Righting Arm (degrees)	27.5	60.0
Area 0°–40° / Required Area	1.13	0.97
Area 0°–30° / Required Area	1.22	1.07
Area 30°–40° / Required Area	1.22	0.95
Angle of Vanishing Stability	>60	>60

Table 10. Comparison of a Fishing Vessel with Intact Righting Energy Criteria, Vessel No. 6
Dungeness Crabbing (pots are 45.4 kilograms each)

Load Condition	Departure Full	Departure Full	Departure Full	Departure Full	Departure Full	Departure Full
Fuel % of Capacity	100	100	100	100	100	100
Water % of Capacity	100	100	100	100	100	100
No. Crab Pots (in holds) / No. Crab Pots (on deck)	61 / 239	61 / 239	61 / 239	0 / 239	0 / 239	0 / 239
Holds Filled with Seawater	0	1	2	0	1	2
Initial Metacentric Height / Required Metacentric Height	2.98	2.79	2.76	3.06	2.81	2.76
Righing Arm @ 30° or Greater / Required Righting Arm	1.68	1.23	0.44	1.73	1.35	0.59
Angle of Maximum Righting Arm (degrees)	22.5	25.0	12.5	22.5	25.0	15.0
Area 0°–40° / Required Area	2.07	1.56	0.68	2.13	1.68	0.88
Area 0°–30° / Required Area	2.32	1.76	0.83	2.39	1.89	1.05
Area 30°–40° / Required Area	1.95	1.44	0.52	1.99	1.57	0.70
Angle of Vanishing Stability	>60	>60	55	>60	>60	57

Table 11. Comparison of a Fishing Vessel with Intact Righting Energy Criteria
Vessel No. 8, Trawling

Load Condition	Departure Full	Departure Full	Departure Restricted	Departure Restricted	Burned Out	Burned Out
Fuel % of Capacity	100	100	65	65	10	10
Water % of Capacity	100	100	50	50	10	10
Holds Filled with Seawater	0	1	0	1	0	1
Initial Metacentric Height / Required Metacentric Height	1.66	2.10	1.65	2.12	1.48	2.02
Righing Arm @ 30° or Greater / Required Righting Arm	0.95	0.76	1.06	0.89	1.03	0.89
Angle of Maximum Righting Arm (degrees)	27.5	60.0	27.5	60.0	27.5	30.0
Area 0°–40° / Required Area	1.13	0.84	1.21	1.01	1.16	1.10
Area 0°–30° / Required Area	1.25	0.97	1.31	1.14	1.25	1.24
Area 30°–40° / Required Area	1.11	0.75	1.23	0.93	1.19	1.05
Angle of Vanishing Stability	>60	>60	>60	>60	>60	>60

Table 12. Summary of All Vessels When Compared
to Righting Energy Criteria for
Salmon Seining Operations
and Vessel 8 Trawling

	Passes or Fails Intact Righting Energy Criteria
Vessel 1	Fails Departure Tanked Condition Fishing Condition
Vessel 2	Fails Departure Tanked Condition Fishing Condition
Vessel 3	Fails Departure Tanked Condition Fishing Condition Burned Out Untanked Condition
Vessel 4	Passes All Conditions
Vessel 5	Fails All Conditions
Vessel 6	Passes all Conditions
Vessel 7	Fails Departure Untanked Condition Tanked Condition Burned Out Untanked Condition
Vessel 8	Fails as a Trawler

SUBDIVISION AND DAMAGE STABILITY OF DRY CARGO SHIPS; AN APPROVAL AUTHORITY VIEW

C. M. MAGILL AND D. J. HOLLAND

IMO has developed requirements for the subdivision and damage stability of dry cargo ships based on probabilistic concepts. These requirements are scheduled for entry into force on 1st February 1992 and will be applicable to ships of 100m length and over the keels of which are laid on or after that date. Ships presently in the design stage and existing designs in series building could therefore be immediately affected by this new legislation.

Comment on the development of the regulations and their trial application by some IMO Member States to a sample range of ship types is made together with Lloyd's Register's findings on ship designs examined for compliance. Guidance is given concerning ship arrangements leading to compliance and interpretations are given for practical aspects such as closing appliances for use in watertight divisions.

INTRODUCTION

At present there are no damage stability requirements applicable to dry cargo ships, except for those of the 1966 Load Line Convention for vessels sailing with reduced Type B freeboards. Such ships are required to survive with certain residual stability characteristics after sustaining damages up to a maximum defined size, but these damage assumptions need not necessarily be applied to all locations in a ship. However, survival of the vessel must be achieved in each case of flooding after damage. Except for the larger multi-hold bulk carriers, most of the world's dry cargo ship fleet are unable to meet this survival standard. Hence there are relatively few dry cargo ships with reduced Type B freeboards.

Although the subject of a damage stability standard for cargo ships has appeared on the IMO agenda from time to time over the last 27 years it has only been since 1983 that any real progress and agreement has been obtained. Finally IMO has now published requirements for the subdivision and damage stability of dry cargo ships, including Ro-Ro ships, based on probabilistic concepts [1]. This use of probabilistic concepts is a major change from the current deterministic methods of assessment of damage stability as specified by existing International Conventions and Codes.

Probabilistic concepts differ from deterministic methods both with respect to the nature of damage and the requirement to survive every assumed damage.

The requirements are scheduled for entry into force on 1st February 1992 and will be applicable to ships of 100m length and over, the keels of which are laid or are at a similar stage of construction on or after that date. Requirements for ships less than 100m length are due for consideration at future meetings of IMO.

PRINCIPLES OF COMPUTATION

Probabilistic concepts address the probability of damage occurring at any particular location throughout a ship and adopt a more rational approach to longitudinal subdivision by considering the likelihood of a damage resulting in the flooding of only one compartment, or any number of other adjacent compartments either longitudinally, transversely or vertically. The probability of a ship having sufficient residual buoyancy and stability to survive in each such case of damage is assessed and the summation of all positive probabilities of survival provides an "Attained Subdivision Index" for comparison against a required norm, the "Required Subdivision Index", for that size of ship.

In probabilistic concepts a ship need not necessarily survive in every possible case of assumed damage provided that there are sufficient survival cases in aggregate which contribute to the minimum "Required Subdivision Index" as determined by IMO.

The principles of this concept were explained in the paper given by Sigurdson and Russas [2] at the Third International Conference of this group in Gdansk, 1986, and these principles remain valid for the regulations now adopted by IMO. However, changes to the original IMO draft proposals on which the paper was based, have been made during the intervening years and a precis is included for reference.

How is the Attained Subdivision Index "A" determined? "A" for any ship is built up by the summation of a series of calculations which consider the effect of flooding all compartments singly and then in groups. "A" is taken as the sum of the products of probability "p", "v" and "s" for each compartment, or group of compartments, where "p" accounts for the damage position along the ships length, "v" the assumed vertical extent of damage and "s" the probability of not capsizing or sinking after such flooding. These calculations take into account that longitudinal and vertical damage is restricted to the compartment or group of adjacent compartments under consideration and the probability of survival after flooding these compartments.

Having assumed that any compartment under consideration is flooded, the probability of survival of the ship can be determined. This is dependent on certain ship characteristics such as initial draught and GM, the permeability of the damaged compartment and the residual stability characteristics after damage. Recognising that the actual loading condition at the time of any damage may vary, the calculations are to be carried out for the ship loaded to the deepest subdivision load line and also at a partial load line taken at the light ship draught plus 60% of the difference between the light ship and deepest load line draughts. The total "A" is taken as the sum of half the "A" value obtained for each of these draughts. If the total "A" is less than the Required Subdivision Index "R" then additional subdivision will be required if the initial ship characteristics are not to be altered. In some cases of non-compliance, for example, an increase in initial GM could result in "A" being greater than "R" but it is thought unlikely that the Owner of a new ship would be happy to accept the permanent carriage of ballast to achieve this result.

COMPLIANCE STANDARD

The degree of subdivision provided in ships to meet the new regulations will be affected by the Required Subdivision Index, "R". "R" is calculated in accordance with a formula where the only variable is the length of the ship, the constants having been chosen by IMO following analysis of trial application of the draft regulations by the Member States to a sample of the world's existing dry cargo fleet. Originally, in 1986, the proposed formula for "R" was the same as that used in the probabilistic method for passenger ships developed in 1973 as an alternative to the deterministic method required by 1960 SOLAS, but ignoring the number of passengers factor "N". This is the formula given in reference [2]. As the various national administrations tested the regulations against ships in their fleet so protracted discussions developed at IMO about this required level of compliance. The calculations performed on existing ship designs by various countries were collated by the United States Delegation in 1987 and, as a result of their findings, they also submitted a proposed formula for "R" using specific coefficients. Generally this set a higher standard for "R" than the adaptation of the passenger ship formula.

This new formula for "R", together with some small modifications to the draft regulations, was approved by IMO in 1988 and published as MSC Circ. 484. A trial period of application was agreed in order to eliminate the concerns of some countries at the level of the required index being set either too high or too low, it previously having been agreed at the Maritime Safety Committee that the level of subdivision would be set at the general level of existing ships.

Once again the United States Delegation collated the information gained by all countries for submission to the 57th session of the MSC in April 1989. The results were discussed in detail and, in particular, European proposals for a lower value of "R" which culminated in a compromise agreement to the slightly lower requirement:

$$R = (0.002 + 0.0009\ Ls)^{\frac{1}{3}}$$

where Ls is the subdivision length of the ship.

It will be noted from the history of the development of the regulations that the principles and method of probabilistic studies has not been in question whilst the compliance standard has been the subject of much debate. The MSC/57 multi-national collation of results prepared by the United States [3] indicates some interesting trends and deviations and provides an insight to the factors most causing debate. Part of this collation is reproduced in Fig.1 and Table 1 for reference. It should be noted that the required index for each ship in this collation is slightly higher than that finally agreed in the above formula, the "R" value in use at that time being:

$$R = (0.001\ Ls)^{\frac{1}{3}}$$

BULK CARRIERS

The results obtained for bulk carriers indicate that for current designs of all sizes there would appear to be little difficulty in complying with the impending legislation except, that is, for self unloading types using the continuous conveyor system (e.g. R = 0.5870, A = 0.3302). It will be noted that for three of the ships examined the attained index is significantly higher than the required index. These three ships are U.S. Flag and were built to standards which require one-compartment damage survivability similar to those of the 1966 Load Line

Ship Number	Service Type	Ls	Length LBP	Breadth	Depth	100% Draught	60% Draught	Required Index	Average Attained	OK/X	Country	Country Index
1	Ro-Ro	121.40	110.00	19.20	13.70	7.02	5.50	0.4952	0.2860	X	USSR	1
2	General Cargo	74.30				3.38	2.53	0.4204	0.6180	OK	FRG	1
3	General Cargo	143.80				8.20	6.09	0.5239	0.3865	X	FRG	2
4	Container	166.80				10.00	7.52	0.5505	0.4475	X	FRG	3
5	Container	210.00				11.00	8.31	0.5944	0.2930	X	FRG	4
6	Ro-Ro Car Container	190.00	180.00	32.25	12.65	8.22	6.29	0.5749	0.7551	OK	JAPAN	1
7	Container	129.13	125.00	21.00	8.10	6.08	4.56	0.5055	0.5533	OK	JAPAN	2
8	General Cargo	104.17	100.00	17.60	8.70	6.88	4.94	0.4705	0.4220	X	JAPAN	3
9	Container	106.44	101.80	17.20	7.35	5.64	4.65	0.4739	0.5065	OK	JAPAN	4
10	Container	114.90	109.00	18.00	8.25	6.13	4.56	0.4862	0.5056	OK	JAPAN	5
11	Ro-Ro Car Carrier	108.73	103.50	19.00	8.10	5.02	4.34	0.4773	0.5914	OK	JAPAN	6
12	Bulk Carrier	127.66	122.00	20.00	11.00	8.40	5.78	0.5035	0.5047	OK	JAPAN	7
13	Bulk Carrier	156.70	150.00	24.60	13.60	9.83	6.74	0.5391	0.6589	OK	JAPAN	8
14	Ro-Ro Car Ferry	150.52	143.00	22.00	13.00	5.80	5.18	0.5319	0.6719	OK	JAPAN	9
15	Ro-Ro Car Ferry	136.27	128.00	22.40	8.00	5.50	4.91	0.5146	0.7636	OK	JAPAN	10
16	Ro-Ro Car Carrier	186.00	176.00	29.20	26.93	8.60	7.08	0.5708	0.5238	X	JAPAN	11
16	Ro-Ro Car Carrier	186.00	176.00	29.20	26.93	8.60	7.08	0.5708	0.6002	OK	JAPAN	11-1
17	General Cargo	163.70	155.00	22.86	13.85	10.20	7.40	0.5470	0.4345	X	JAPAN	12
17	General Cargo	163.70	155.00	22.86	13.85	10.20	7.40	0.5470	0.4749	X	JAPAN	12-1
18	Ro-Ro Car Carrier	176.63	170.00	28.00	12.30	7.50	6.16	0.5611	0.6123	OK	JAPAN	13
19	Heavy Lift	150.12	145.50	26.80	13.80	9.50	6.86	0.5315	0.4055	X	JAPAN	14
20	Container	214.94	204.00	32.20	18.70	11.50	9.03	0.5990	0.7753	OK	JAPAN	15
21	Ro-Ro Car Carrier	190.73	180.00	31.70	13.05	8.50	7.22	0.5756	0.1988	X	JAPAN	16
21	Ro-Ro Car Carrier	190.73	180.00	31.70	13.05	8.50	7.22	0.5756	0.7115	OK	JAPAN	16-1
22	Ro-Ro	174.00	168.00	27.50	16.75	9.00	7.00	0.5583	0.4657	X	JAPAN	17
23	Container	159.60	152.00	23.10	14.10	9.90		0.5424	0.6006	OK	JAPAN	18
24	Bulk Carrier	222.50	216.57	24.04	14.17	9.54	6.52	0.6060	0.3302	X	CANADA	1
24	Bulk Carrier	222.50	216.57	24.04	14.17	9.54	6.52	0.6060	0.7980	OK	CANADA	1-1
25	Bulk Carrier	222.54	216.10	23.08	14.63	9.77	6.74	0.6060	0.8710	OK	CANADA	2
26	Bulk Carrier (OBO)	218.50	206.05	22.86		11.07	8.26	0.6023	0.8484	OK	CANADA	3
27	Ro-Ro	190.70	178.00	27.00	17.60	9.12	7.38	0.5756	0.3488	X	CANADA	4
27	Ro-Ro	190.70	178.00	27.00	17.60	9.12	7.38	0.5756	0.5756	OK	CANADA	4-1
27	Ro-Ro	190.70	178.00	27.00	17.60	9.12	7.38	0.5756	0.5905	OK	CANADA	4-2
28	Ro-Ro	105.10						0.4719	0.2190	X	FRANCE	1
29	Ro-Ro	145.20						0.5256	0.1430	X	FRANCE	2
30	Ro-Ro	201.80						0.5866	0.5330	X	FRANCE	3
31	Bulk Carrier	100.48	92.30	13.80	7.70	6.00	4.72	0.4649	0.5944	OK	CHINA	3
32	Ro-Ro	174.80	160.00	26.50	18.50	8.52	7.26	0.5591	0.3747	X	CHINA	4
33	Ro-Ro Car Carrier	166.57	161.99	29.83	12.81	8.45	6.91	0.5502	0.3264	X	USA	A
33	Ro-Ro Car Carrier	166.57	161.99	29.83	12.81	8.45	6.91	0.5502	0.8853	OK	USA	A-1
34	Ro-Ro	206.58	193.23	29.56	20.42	9.59	7.44	0.5912	0.7851	OK	USA	B
35	Ro-Ro	167.30	156.14	21.60	13.90	6.45	5.00	0.5510	0.1542	X	USA	C
36	Ro-Ro Container	286.19	174.40	32.20	26.82	10.57	9.45	0.6590	0.9580	OK	USA	D
37	Ro-Ro	185.00	177.99	27.00	17.60	9.12	7.42	0.5698	0.6294	OK	USA	E
38	Ro-Ro Container	249.28	233.47	32.15	20.73	10.21	8.39	0.6294	0.7909	OK	USA	G
39	General Cargo	138.86	135.30	18.90	11.58	7.70	5.87	0.5178	0.9590	OK	USA	I
40	General Cargo	113.00	108.00	16.40	9.45	5.18	4.81	0.4835	0.7588	OK	USA	J
41	General Cargo	132.60	131.37	22.25	13.56	7.77	6.00	0.5099	0.8980	OK	USA	K
42	Bulk Carrier	168.90	162.00	25.00	13.80	9.92	7.94	0.5528	0.6471	OK	USA	L
43	Bulk Carrier	195.50	192.00	29.76	14.65	10.76	7.66	0.5804	0.6091	OK	USA	M
44	Open Hatch	149.24	142.14	20.20	11.20	8.41	6.19	0.5304	0.6286	OK	USA	N
45	Ro-Ro	187.00	180.80	28.00	18.85	9.20		0.5719	0.5200	X	USA	O
46	Bulk Carrier	241.90	235.10	32.20	20.00	13.92		0.6231	0.6405	OK	USA	P
47	Ro-Ro	181.90	173.20	32.26	26.80	12.00	8.40	0.5666	0.7630	OK	USA	Q
48	Ro-Ro	290.50		32.26				0.6623	0.7627	OK	USA	R
49	Ro-Ro	146.20	142.40	22.03	7.95	5.77		0.5268	0.5795	OK	USA	S
50	Container	249.30	245.00	32.20	18.80	11.88		0.6294	0.8352	OK	USA	T
51	Container	201.01	185.19	24.54	14.60	7.89		0.5858	0.9152	OK	USA	U
52	Container	265.60	260.00	39.40	23.60	9.47		0.6428	0.7456	OK	USA	V
53	Ro-Ro	224.10		32.26				0.6074	0.6326	OK	USA	W
54	Ro-Ro	213.50		32.26				0.5977	0.6335	OK	USA	X
55	Ro-Ro	187.70		32.26				0.5726	0.5604	X	USA	Y
56	Ro-Ro Railship	183.29	174.40	21.60	18.95	6.50		0.5680	0.7945	OK	USA	Z

Table 1: Summary of ships' data

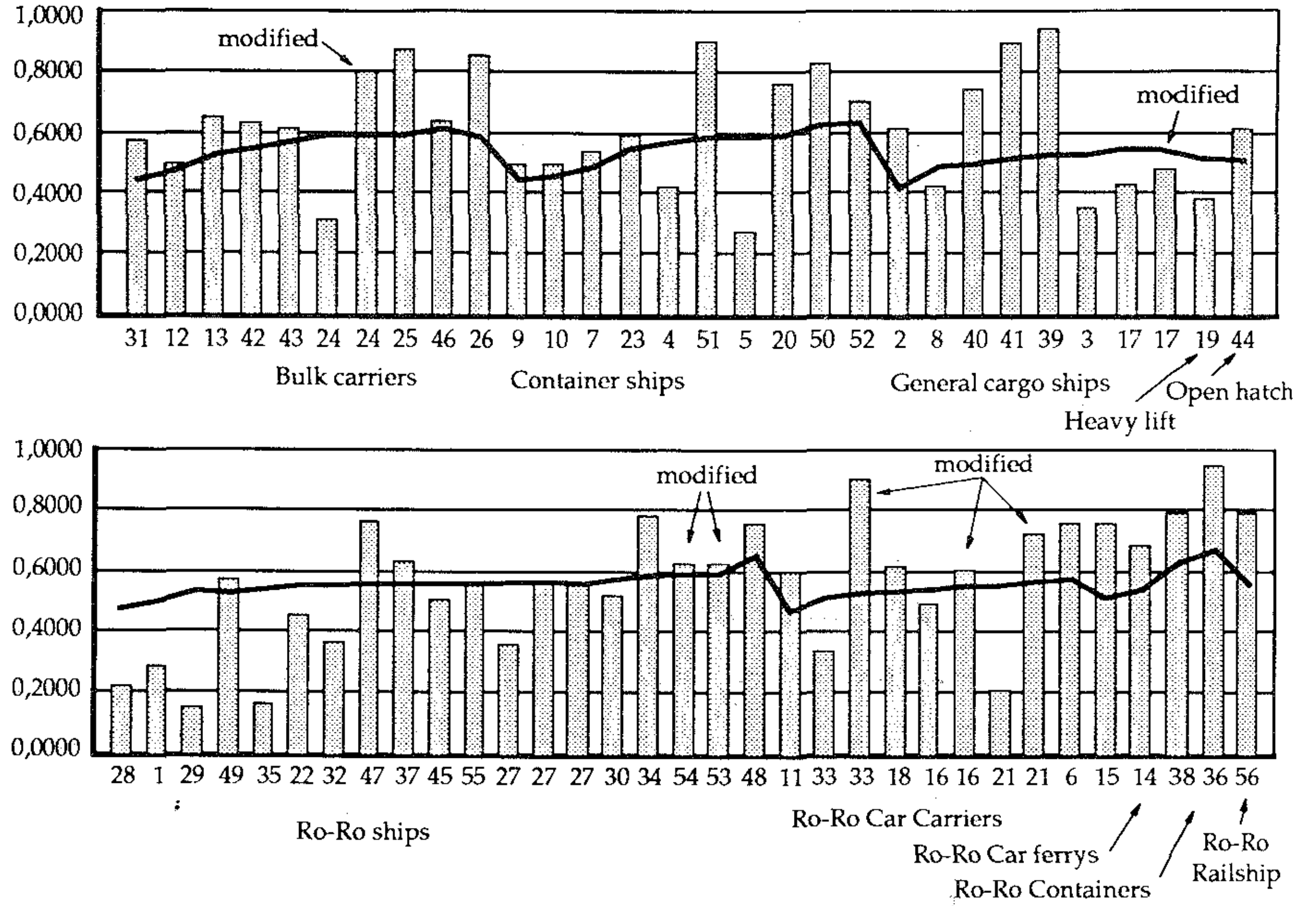

Figure 1: MSC 57 Multinational Collation of Application of MSC/Circ. 484

Convention, Regulation 27, for vessels with reduced Type B freeboards.

For these ships, survival in all cases of one-compartment damage gives rise to significant additional partial "A" values above those to be expected for other ships where flooding of the foremost and aftermost spaces is likely to lead to excessive trim, progressive flooding and non-survival. The conclusions to be drawn from these findings are addressed later in the paper.

The results obtained by Canada for self-unloading bulk carriers indicating significant deficiency of "A" from "R" are for ships where the hold bulkheads are penetrated by continuous conveyor systems. Watertight subdivision as required by both the Load Line and SOLAS Conventions is understood to be not achievable by current conveyor sealing systems and all holds therefore must be considered common for the purpose of flooded stability calculations.

Alternative self unloading systems which do not penetrate hold bulkheads would appear to offer one solution to the problem at this time. However, modification to the basic design of vessels evolved from the Great Lakes type may be less radical than a change of unloading system. Great Lakes types have upper saddle tanks and lower hopper tanks interconnected by a "double skin" effect of about 6.6.% of the beam.

Void spaces, which are open and common with the cargo holds, exist on each side of the conveyor tunnel. Modification of this arrangement whereby the centreline void space is made watertight between hold bulkheads, see Figure 2, the upper saddle tanks made independently watertight and the "double skin" effect increased to about 8.5% to 9% of the beam depending on details of the overall compartmentation, would appear to provide a basis design complying with the impending legislation. To further reduce the width of the double skin, subdivision of the tanks within the double skin could be utilised to provide more flooding survival cases and thus additional contributions to "A".

CONTAINER AND GENERAL CARGO SHIPS

These types showed significant variation in the Attained Index with as many ships attaining or exceeding the required index as there were not, thus indicating that a number of existing designs will require modification with respect to the position of main bulkheads if series building, for example, is to continue after February, 1992.

Similar to the findings for bulk carriers, the

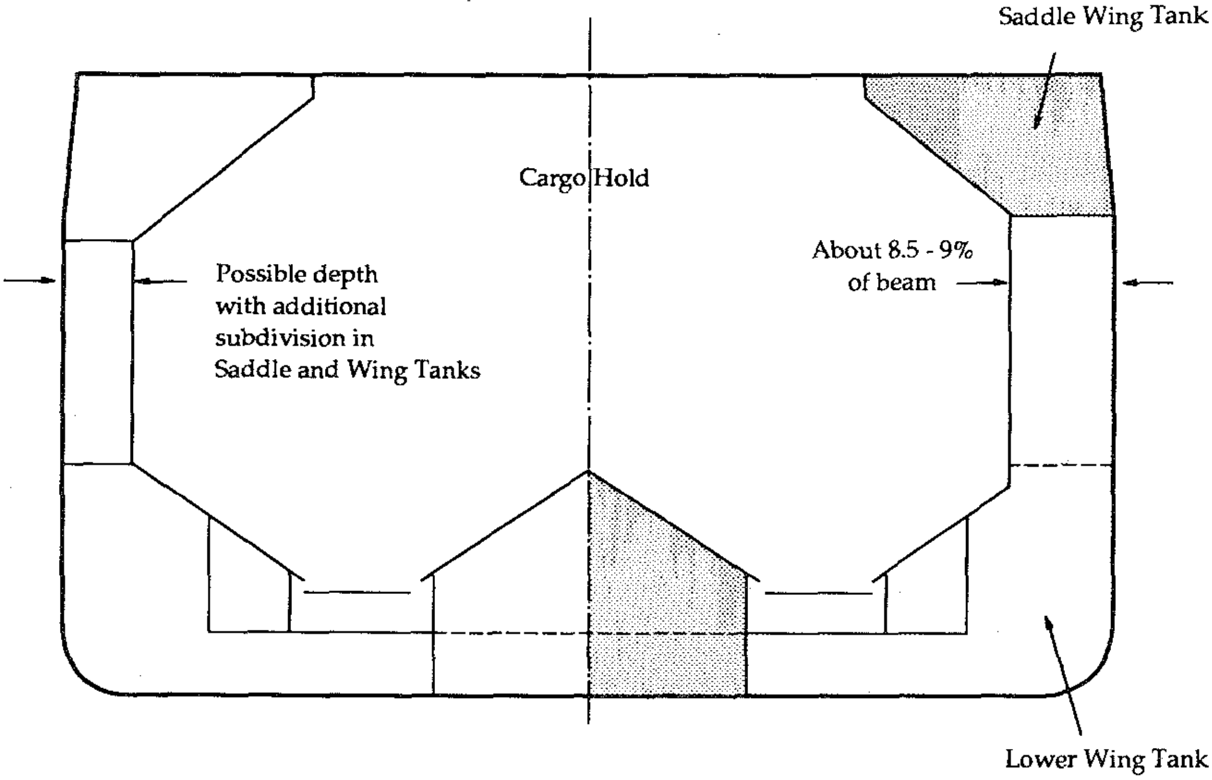

Figure 2: Self Unloading Bulk Carrier (Great Lakes Type)

exceptionally high "A" values obtained by USA for some general cargo and container ships is indicative of construction to a one-compartment damage survivability standard giving rise to additional partial "A" values for the greater number of flooding survival cases.

Another significantly high "A" value is that obtained for a hatchcoverless design. This trend has been confirmed by Lloyd's Register on hatchcoverless container ship designs ranging in size from 500 TEU up to 4000 TEU Panamax. Hatchcoverless vessels have increased freeboards to reduce the ingress of water whilst in a seaway and the double skin in way of this increased freeboard usually has, at least, an underdeck passageway from one end of the vessel to the other. This underdeck passageway provides a watertight reserve of buoyancy above the maximum possible height of damage, Hmax, at least in the partially loaded condition. This undamageable reserve of buoyancy contributes to more cases of survival and corresponding partial "A" values than can be expected for conventional container ships, see Figure 3. The relationship of Hmax with dl or dp as appropriate is shown in Figure 4 for reference.

In a conventional design this underdeck passageway is in the damageable region and flooding throughout its length is often the reason for non-survivability in many cases of flooding, this non-survival being a major factor is not achieving compliance with the Required Index "R". Subdivision of the passageway becomes necessary in such cases and watertight doors of the hinged type may be accepted in such locations since they will be above the "freeboard" deck and normally closed at sea.

Such doors should be of scantlings suitable for the pressure head of water involved; be of single lever, multiple clip operation and provided with mechanisms to ensure ease of closure and the safety of transiting personnel whilst the vessel is in a seaway. Open/closed indicators at the bridge position are also required in accordance with the regulations.

RO-RO CARGO SHIPS

Ro-Ro ships as a type showed similar results to container and general cargo ships but only for lengths above about 140 to 150 metres. For ships with subdivision lengths below this it would appear that many existing designs must have additional subdivision if further "similar ship" new buildings are to be contemplated after 1992. However, it is more likely that new buildings after 1992 will utilise notable arrangements from the Ro-Ro ships which were examined and found to comply with the legislation.

In general, compliance was found for arrangements where the main Ro-Ro deck is located above the deepest load waterline and cellar deck holds are protected by a "double skin" effect provided by water ballast or other side tanks. In such arrangements the cellar deck hold has been aft of .25 to .3 L from

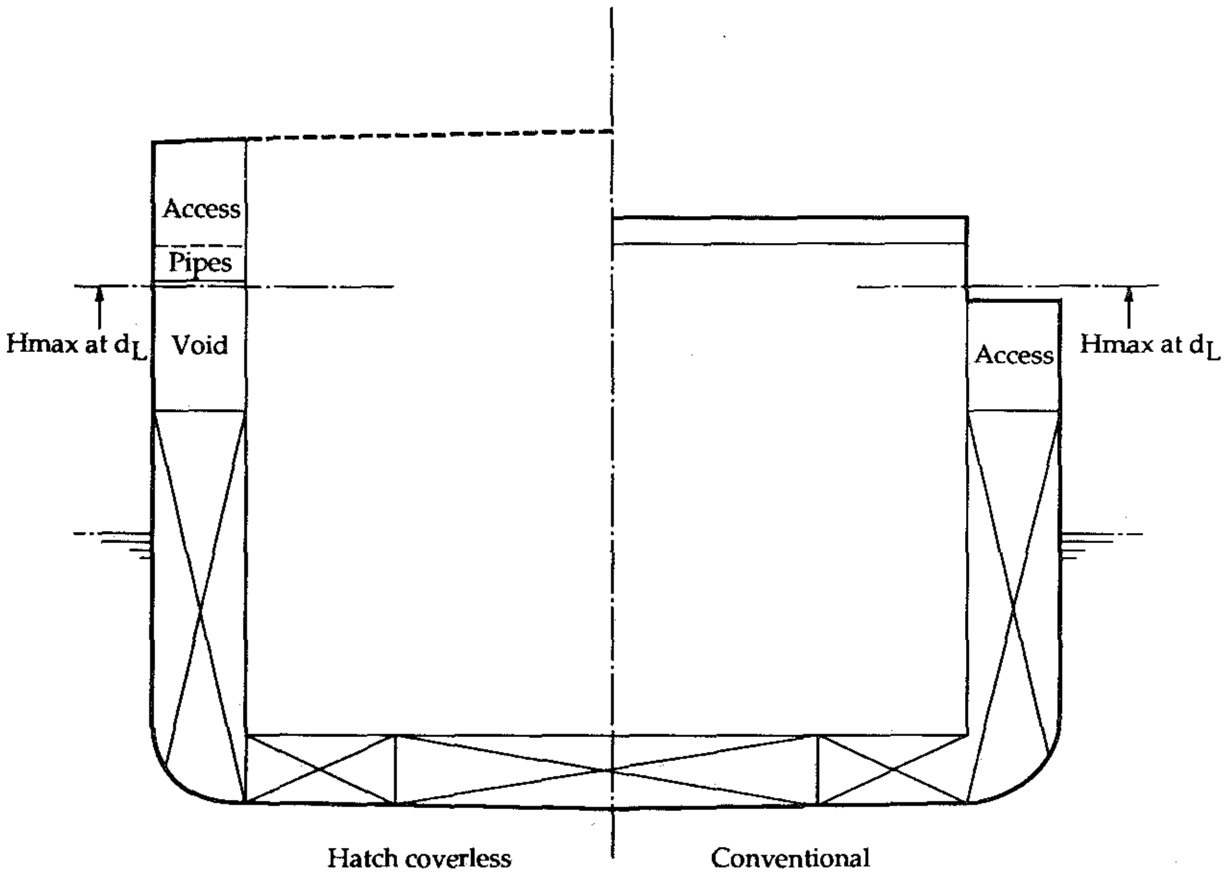

Figure 3: Effect of Maximum Height of Damage

forward and significant contribution to the attained index has been achieved from the good watertight subdivision in the forward part of the ship.

Alternatively the "double skin" concept could be developed throughout the length of the vessel in order to provide sufficient damage survival cases of shell penetration only to compensate for the lack of transverse subdivision in the Ro-Ro areas.

Watertight doors and ramps in the Ro-Ro areas will also give a significant improvement in attained index. The provision of doors in Ro-Ro areas is not likely to find much favour with Owners whose principal intention when building such a ship is speed of turn-round in port. However, watertight ramps are worthy of more consideration since they may give rise to "undamageable" buoyancy spaces above Hmax which would significantly improve the attained index. Pure car carriers with their very high sides and multiple decks are a particular type of Ro-Ro which would benefit in this way. Also, utilisation of the double skin effect previously mentioned for conventional Ro-Ro's need not lose as much Ro-Ro capacity as may be first thought since, if a multiple deck ship, it should be possible to gain additional contributions to "A" from the undamageable decks above Hmax in the same way as that proven for hatchcoverless designs and therefore keep the double skin to at least less than 10% of the beam.

It must be recognised that Ro-Ro ships will be the most affected ship type following entry into force of the legislation, however it should also be remembered that this ship type is the most vulnerable to rapid capsize following a collision. Casualty statistics indicate that about 37% of Ro-Ro's in collisions were total losses through rapid capsize, a rate which is more than double that found for conventional cargo ships. The inclusion of the "v" factor in the regulations was made principally so

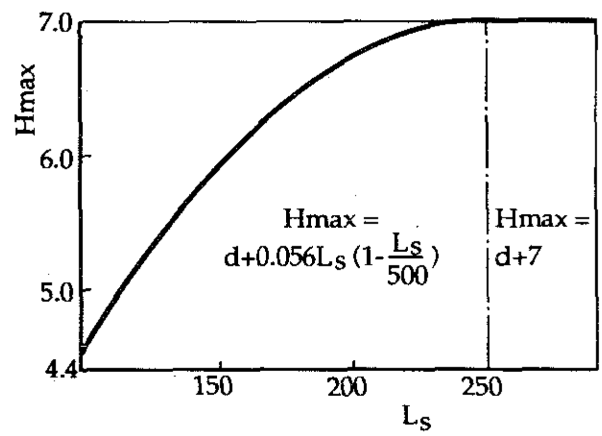

Figure 4

that Ro-Ro ships would not be unfairly penalised and to give encouragement to the horizontal subdivision which could be provided in lieu of normal transverse subdivision as in other cargo ships.

REGULATION INTERPRETATION

To be effective stability regulations must be clear in their intent and be as simple to apply as is consistent with a level of accuracy which will ensure that the required safety standards are not compromised.

Examination of the text of the regulations will indicate that the desired simplicity is not inherent in the probabilistic method although this may be in part due to their "newness" and user unfamiliarity with the standard. However, varying results for the same test ship from those Administrations taking part in the trial application have demonstrated a need for detailed guidance notes to be published by IMO in order that worldwide uniformity of application is achieved. These guidance notes are nearing completion and will be submitted to the next session of the SLF Sub-Committee in February 1991. Their content is understood to be the subject of another paper at this Conference.

However, two important aspects have already been discussed at the last SLF Sub-Committee meeting. Text has been agreed for inclusion as a footnote to Regulation 25-1 to clarify that an examination for compliance with the new regulations need not be carried out for ships which are subject to mandatory damage stability investigations in accordance with other IMO Conventions. These include the 1966 Load Line Convention, Regulation 27, damage stability requirements for bulk carriers with reduced Type B freeboards and MARPOL 73/78, Regulation 25 damage stability requirements for OBO's. The rational for this can be seen from the very high attained index values computed for the one-compartment damage survivability vessels as shown in Figure 1.

The other matter relates to Regulation 25-4 concerning the theoretical damage a vessel may sustain. For the purpose of considering the floodability of watertight compartments inboard of wing compartments Regulation 25-4.5 specifies "a rectangular penetration which extends to the ships centreline, excluding damage to any centreline bulkhead". The damage statistics on which these and all other IMO regulations are based indicate that damage penetration beyond B/5 is not probable. Therefore it is considered that in order to have a damage criteria for cargo ships which is not more severe than that for passenger ships, chemical tankers, etc., then damage to pipes and ducts connecting watertight compartments need not be taken into consideration provided they are inboard of B/5 or inboard of any longitudinal subdivision which is itself inboard of B/5. That is, Regulation 25-4.5 damage extents are for the purpose of considering a space to be broached and open to the sea without its internal piping and equipment necessarily also having been damaged. All other aspects of progressive flooding normally taken into account in damage stability investigations will apply to dry cargo ship damage cases.

FUTURE DEVELOPMENTS

The regulations have been developed such that possible future improvements can be made without the need for any fundamental change in structure. As more statistical damage information becomes available to IMO which indicates need for change, this can readily be incorporated by amendment to the relevant factors of the computation. Additionally, the assessment of attained index is totally separate from the compliance standard, "R", thus creating scope for setting a lesser standard for vessels below 100m in length without change to other parts of the regulations. The compliance standard for ships of less than 100m Ls is on the agenda for discussion at the next SLF Sub-Committee Meeting in February 1991.

There are proposals from some National Administrations to accept only probabilistic investigations for passenger and cargo ships. Whilst the time is not yet ripe for such a change it is possible to see that this is a likely development for the future. This being the case, it is also possible that the probabilistic method could be used in all other Conventions for damage stability investigations and that the assignment of load lines will eventually be based on compliance with probabilistic damage stability investigations.

CONCLUSIONS

The probabilistic method of assessment of damage stability offers a means whereby the comparative safety standard of a ship may be assessed against a predetermined norm for its length. This is a fundamental change of thinking from the criteria established in existing conventions where survival is required for specific extents of damage. In the probabilistic method survivability after flooding is not required for every conceivable extent of damage but only for a sufficient number of cases in aggregate to achieve the required norm.

It is recognised that some existing cargo ship designs, in the absence of a mandatory standard, have poor flooding survivability characteristics. IMO has set a compliance standard aimed at eliminating such designs, however, it may be some time before world shipping casualty statistics can verify if this has been achieved.

Examination of the regulations and particularly the computations required to obtain the "A" value will indicate that this is a complex reiterative process involving a great many damage scenario calculations even on a simple ship. It is clear that such a task can only be efficiently carried out using a suitable computer and dedicated software. It will also be evident that these calculations must be carried out as early as possible in the preliminary design stage since the findings could have a bearing on the final location of the main longitudinal and transverse subdivision bulkheads.

Therefore it can not be stressed too strongly that early examination for compliance is a must if financial penalties are to be avoided during building since the positions of bulkheads can easily be moved on paper without cost but this action would be very expensive after construction has started!

Recognising the importance of this legislation Lloyd's Register has decided to introduce a Descriptive Notation [4], SDS (Solas Damage Stability) for LR Classed vessels which have been examined and found to comply with the forthcoming regulations.

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SAFETY FOR DAMAGED VESSELS AS PROBABILITY OF NON-CAPSIZING IN FOLLOWING SEAS

Giorgio Trincas (*)

ABSTRACT

Analysis of damage stability during intermediate stages of flooding is a very important item to foresee dangerous situations which may induce ship capsizing. The damage scenario is simulated through a simplified deterministic model which reflects the dynamics of the ship. A time-domain simulation for computing ship motions coupled with sloshing effects in damaged compartments is developed to monitor the history of transient flooding. Variations of static and hydrodynamic characteristics of the ship and floodwater as well are taken into account at each small step in time.

To determine the non-capsizing probability during the flooding elapsed time of a ro-ro passenger ferry, initially at rest in a following sea, the Bayes' formula for the entire probability is applied by computing the probability distribution of a wave group excitation and the conditional probability of the dangerous event.

MODELLING OF FLOODING IN A SEAWAY

Accident records refer many causes leading to possible capsizing and among them the dangerous condition when a ship sailing in a longitudinal seaway is subject to the action of damage floodwater. Nowadays there are neither international statutory regulations nor reliable guidelines to facilitate the selection of data for modelling flooding scenarios and to evaluate the level of risk resulting from damage.

So far, flooding mechanism simulated in available damage stability computer programs is essentially based on a static approach and searches for static equilibrium both at intermediate and at final stages of flooding. Such a physical modelling is only a rough approximation of the real phenomenon since it is sufficient to consider the intrinsic phase lag between restoring forces and oscillations to remind that a damaged ship, also in calm water condition, cannot be considered in instantaneous hydrostatic equilibrium. It has been verified that the static approaches usually applied to assess damage stability [16,18], including the IMO probabilistic one, cannot explain the most part of capsizing accidents.

Damaged ship survivability is a very difficult target to reach also because of the unsufficient knowledge of the dynamics and hydrodynamics of flooding phenomenon. The effect of floodwater on the transient response of the ship has not been investigated extensively up to now. Flooding transients can play a dramatic role in damaged stability since it is sufficient one excessive excursion to have the system failure. Moreover, survival capability in the event of damage should not be assessed through pure deterministic approach, but from the statistics of the ship responses obtained from a time simulation of fully coupled equations of motion. The necessity of introducing statistics derives from the randomness and uncertainty of meaningful ship parameters depending also on operative conditions (centre of gravity, freeboard), of extent and position of damage, of three-dimensional permeability distribution, of sea state, of floodwater dynamics, and of initial kinematic conditions.

Because of the complex and stochastic nature of a damage scenario, in the beginning we prefer to model the phenomenon by means of a simulation that describes flooding merely through fluid mechanisms based on foronomia methods according to Bernoulli's theory of flow [12] and is inclusive of the effects of sloshing in the damaged compartments. The flooding evolution is described through a series of small time steps. The hydraulics of the mechanism operates in

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the same way to that used in static calculations for intermediate stages of flooding, but without searching for equilibrium positions. At each point of time hydrostatic and hydrodynamic characteristics of the instantaneous wetted surface of the ship are computed on the ground of the values of volume, heel, and trim at previous time step. Such a mechanism easily permits to continuously monitor the incremental variation in ship motions during the course of flooding.

The study presented here is part of a research programme on damage stability designed to be developed in progressive stages, the final goal being both the improvement of existing rules for capsizing prevention and the development of guidelines that can help the designer to limit the risk of exceeding some roll angle considered dangerous for survivability. The time-domain flooding simulation could be a useful tool for different design purposes. It should be used to determine the ship behaviour at each damage case in combination with a probabilistic assessment of the various eventual cases of accident, to predict the probable time until the vessel sinks or until a heel angle is reached at which the lifesaving gear is useless. It could be of use not only to assess the safety implications of watertight door closure, but also to determine the optimum subdivision scheme for a given level of safety. Finally, also the operative and safety procedures could be previewed by time-based calculations for realistic conditions.

MATHEMATICAL MODEL

In order to investigate how the motions of a damaged ship, particularly roll, build up and which ship parameters are the most important, a general formulation of the ship motion problem in the time-domain is considered to be representative of the physical event. It has to include theories of seakeeping, computational fluid dynamics to take into account floodwater effects, and methods of random phenomena in a seaway. Damage flooding is a typical transient phenomenon so that it can be considered as an initial-value problem. At initial time the ship in upright position is assumed at rest and subject to the disturbance of a regular wave whose profile along the free surface simulates a stochastic longitudinal sea. The improvement of such a mathematical model is deemed as preliminary to necessary experimental studies on models because numerical simulations allow large statistics and preliminary conclusions about capsizing phenomenon that should be tuned in a future experimental study.

Contrarily to other time domain approaches [18, 19], our dynamic model does not envisage time evolution as a sequence of states of hydrostatic equilibrium. In a first study by Francescutto and Trincas [7], a linear mechanical model was assumed to describe the motions in heave, roll and pitch. The mass of the ship was considered slowly varying in time under the added weight of floodwater. The equations of motion were considered only implicitly coupled through hydrostatic terms. The results were not promising enough because nonlinear terms of mechanical origin representing coupling were not introduced. Moreover, hydrostatic forces and moments were computed on the ground of a simplifying approach considering superposition of independent effects.

In the light of these considerations, a time-based simulation of damage flooding has been developed where all the time-varying restoring/excitation forces and moments, the ship hydrodynamic coefficients and floodwater forces are computed with reference to the instantaneous hull, regardless of ship equilibrium being. The ship is assumed to be a six-degree-of-freedom rigid body system in unbounded nonviscid fluid. As pure loss of stability in longitudinal seas is one of the most frequent modes of capsizing, priority has been given to the mathematical model describing the dynamics of a side damaged ship at rest in following waves.

Two sets of right handed coordinate systems are used to describe the motions of the ship (Fig. 1). The first is an inertial frame $O_0x_0y_0z_0$ with the origin fixed on the undisturbed surface, where the x_0-y_0 plane corresponds to the calm water level. The positive directions of the x_0, y_0, z_0 axes are forward, to starboard and downward respectively. The next is a body-fixed reference system Oxyz with its reference point at the centre of gravity of the ship. As large motions are forecastable and every rotation is wanted to be independent of the other angles, the so-called Eulerian angles - ψ, θ, ϕ in that sequence - are introduced to define the rotation of one system relative to the other.

Although the simulation results do not depend on the choice of the coordinate system, in a time-domain simulation it is suitable to perform hydrostatic and hydrodynamic calculations in the ship-fixed system of coordinates as the forces and moments depend on the velocity and position the ship assumes at each point of time. Kinematics gives the relationship between a displacement vector $\mathbf{x}$ defined in the ship-fixed system and the displacement vector $\mathbf{x_0}$ described in the inertial system

$$\mathbf{x} = [T]\,\mathbf{x_0}$$

where [T] is the 3x3 transformation matrix

$$[T] = \begin{bmatrix} \cos\psi\cdot\cos\theta & \cos\psi\cdot\sin\theta\cdot\sin\phi - \sin\psi\cdot\cos\phi & \cos\psi\cdot\sin\theta\cdot\cos\phi + \sin\psi\cdot\sin\phi \\ \sin\psi\cdot\cos\theta & \sin\psi\cdot\sin\theta\cdot\sin\phi + \cos\psi\cdot\cos\phi & \sin\psi\cdot\sin\theta\cdot\cos\phi - \cos\psi\cdot\sin\phi \\ -\sin\theta & \cos\theta\cdot\sin\phi & \cos\theta\cdot\cos\phi \end{bmatrix}$$

Notice that the translation vector beween the two coordinate systems is not considered because the ship is assumed at zero speed. Then, if the vectors $\mathbf{V}(u,v,w)$ and $\boldsymbol{\omega}(p,q,r)$ represent the linear and angular velocities in the instantaneous ship-fixed system, the transformations between the inertial and ship axes are given by

$$\mathbf{V}(u,v,w) = [T]\,\dot{\mathbf{x}}_0$$

$$\boldsymbol{\omega}(p,q,r) = [S]\,(\dot{\psi},\dot{\theta},\dot{\phi})$$

$$[S] = \begin{bmatrix} -\sin\theta & 0 & 1 \\ \cos\theta\,\sin\phi & \cos\phi & 0 \\ \cos\theta\,\cos\phi & -\sin\phi & 0 \end{bmatrix}$$

The forced motions η_i ($i = 1, 2, \ldots, 6$) described in the instantaneous ship-fixed reference system are given respectively by the following set of linearized second-order differential equations based on Newton's law of dynamics:

$$\begin{cases} m\,(\dot{u} + qw - rv) = X \\ m\,(\dot{v} + ru - pw) = Y \\ m\,(\dot{w} + pv - qu) = Z \\ I_{xx}\,\dot{p} + (I_{zz} - I_{yy})\,qr = K \\ I_{yy}\,\dot{q} + (I_{xx} - I_{zz})\,rp = M \\ I_{zz}\,\dot{r} + (I_{yy} - I_{xx})\,pq = N \end{cases}$$

where m is the ship mass, and I_{xx}, I_{yy}, I_{zz} are the principal moments of inertia independent of the motions. The right hand sides of the equations represent the components of the excitation force and moment in x, y, z directions and around them, respectively. They depend on the time history of the ship motion.

The set of differential equations can be written as a system of 12 coupled first-order differential equations. They are solved by means of a Runge-Kutta-Merson integration scheme where time intervals of different integration step may be selected. Thus, small steps are used in transients of relatively high frequency response and large steps beyond these transients.

The advantage of such a mathematical model comes from the fact that it allows all the external forces to be computed separately in terms of the combination of the ship condition and the sea state. The total external force in whichever mode of motion is the linear superposition of various contributors. In this study the following set of forces (and moments) has been taken into account:

- weight and hydrostatic forces of the intact ship
- hydrodynamic forces proportional to the velocity and acceleration of the ship
- excitation forces caused by the incident wave and diffraction effects
- static forces of floodwater
- sloshing forces due to the interaction between floodwater and ship motions.

As the hydrodynamic forces are expensive to compute at each frequency of oscillation, the frequency domain coefficients can be calculated off-line and stored for future interpolations, so giving the momentary forces at each time step. Two subsequent 2D interpolations by splines are used, the first in modulus- and phase-frequency spaces, and the second for each transverse section in draft space.

The ship exciting fluid actions are driven by the natural roll frequency ω_n. As hydrodynamic coefficients are frequency dependent owing to memory effect and since ω_n cannot be the frequency in upright ship position, it is assumed to be the one depending on the time-varying arm GZ [11]

$$\omega_n(\phi, t) = \frac{1}{\phi\cdot k_{xx}}\sqrt{2g\int_0^{\phi} GZ(\phi, t)\,d\phi}$$

where ϕ is the roll amplitude at previous step and k_{xx} is the transverse gyradius.

The wave exciting forces and moments, and the hydrodynamic coefficients as well are computed by means of a seakeeping program based on the two-dimensional linear potential theory inclusive of the Frank close-fit-method [17]. The nonlinear damping effects in roll are computed according to Ikeda's method [10]. Appropriate kinematic relationships are used to transfer the fluid actions from the water plane to the centre of rotation of the ship [3]. The strip-theory equations of motion are decoupled into one set for the longitudinal motions and a second set describing the lateral motions. Since the shape of the hull transverse sections can be highly asymmetric in the presence of large motions, this decoupling could be incorrect. Nevertheless, starting from the fact that the low-frequency following sea motions are dominated by the hydrodynamic coupling given in Timman-Newman relationship [20], some fundamental effect like the complete coupling of single modes of motions and the exact evaluation of the hydrodynamic coefficients can be disregarded. Moreover, when the instantaneous position of the ship notably differs from its mean position, motions will be largely determined by the quasi-static forces and moments due to the waves. Thus, it results of paramount importance to calculate them exactly by considering the time-dependent underwater geometry relative to the wave profile. The wave disturbance has to be referred to the ship-fixed coordinate system taking into account the relative motions between the at-rest water surface and the adjacent ship surface

$$\xi(x,t) = \xi_0 \cdot \cos\left[k(x + \eta_1 - \eta_2\eta_6 + \eta_3\eta_5) - \omega t \right]$$

Before developing more complicate and exact models, only the effect of static-static coupling has been introduced by considering the second order derivatives of heave force, roll moment, and pitch moment for heave-roll, roll-pitch, and pitch-heave coupling [4]. In reality, a pseudo-coupling intrinsically exists because of the mutual influence between vertical and lateral motions due to the variation of hull geometry.

The exciting forces include also the forces exerted on the ship by the floodwater which are nonlinearly dependent on the ship motions. This is a further reason to model damaged ship motions by a time-domain simulation. In order to derive the forces and moments acting on the damaged compartment it is necessary to compute the displacement, velocity and acceleration components at its reference point through a transformation procedure considering the vectorial distance from the ship centre of rotation. The fluid actions due to floodwater motion are then transferred to the ship centre of gravity by means of an inverse transformation and used to solve the motion equations at the next time step in the simulation procedure.

COUPLING BETWEEN SLOSHING AND SHIP MOTIONS

Damage flooding is a case of slack loading where ship motion is affected by the floodwater motion. The problem of determining the dynamic effects due to the floodwater movement under ship oscillations remains one of the actual problems of seakeeping studies. The magnitude of sloshing forces and moments is considerably affected by the compartment geometry and baffles, the quantity of moving floodwater, and the amplitude and frequency of its motion. All of them mainly depend on the fill level.

Theory and experiments show that progressive flooding of a ship's compartment partially filled between 10 and 90 percent of its depth can cause a resonant motion of floodwater which may match resonant ship motions. The relation for the lowest resonant liquid period in roll versus tank filling level for rectangular tanks, that correlates well enough with experimental results [2], is given as

$$T_R = \frac{2\pi}{\sqrt{(\pi g/b)\cdot\tanh(\pi h/b)}}$$

where h is the liquid filling height and b is the breadth of the compartment.

The excitation periods of ship motions have to be compared with the resonant sloshing periods of the damaged compartments. When periods overlap or the forcing period is about the natural period of floodwater, maximum amplitudes of these motions will result in maximum sloshing forces as a result of amplificated coupling between heave and roll motions [14]. When the forcing period is away from either the natural period of roll or of pitch, there is very little liquid motion so that sloshing loads do not need nonlinear mathematical models.

When the compartments are deeply filled, for excitation periods sufficiently larger than their natural period, particularly in roll motion, that is, when damaged compartments are undergoing non-resonant low amplitude motions, the linear equations given by Abramson [1] or the equation of motion of the liquid free surface derived from Lagrange's equation [15] can be of use to derive the sloshing forces on the flooded compartment. If the motions are large it is necessary to introduce nonlinear free surface conditions. When the fill level is small and it does exist strong syncronization, the Glimm's method appears to be the best tool to solve the sloshing problem [5,15].

The evaluate the interaction between sloshing in a damaged compartment and ship motions when synchronisation is expected, a simplified model of the coupled problem has been set up whereby the sway, heave, and roll motions of the ship drive the sloshing program which in turn determines the dynamic components of the floodwater action to be then combined with the other exciting forces in the next time step. For the time being, our study makes use of the 2D simulation developed by Eguchi and Niho [6]. They formulate the accelerations of the fluid elements in the damaged compartment as

$$\dot{v}_f = g_1(\phi, \dot{\phi}, \ddot{\phi}, w)$$
$$\dot{w}_f = g_2(\phi, \dot{\phi}, \ddot{\phi}, v)$$

where v_f and w_f are the velocity components relative to the compartment-fixed coordinate system. A finite-difference integration scheme is used to derive the accelerations in the momentum equations of the fluid as the body forces per unit mass.

Other linear and nonlinear techniques have been developed to compute the sloshing effects in different situations. A complete parametric analysis is deemed necessary in order to evaluate the ship responses when modelling the sloshing mechanism in alternative ways and to analyse sloshing forces when variations in subdivision arrangement are studied.

PROBABILITY OF NON-CAPSIZING

Like every deterministic model, also ours built to simulate motions of a damaged ship has poor ability to deal with uncertainties. Since variables such as loading, seaway, position and extent of damage, all of them associated to uncertainty, heavily affect ship motions, it is important to consider their probabilistic features. Among the others, a widely used procedure for dealing with uncertainty while using deterministic models is to define different damage scenarios and to rerun the deterministic model for each one of them.

Instead of computing the probability of capsizing within some duration of time under certain operative conditions and in given sea regions, safety could be better managed by the designer in terms of probability of non-occurrence to exceed some extreme roll angle either derived from some rule and constraint or believed dangerous. In order to provide adequate safety with respect to design situations, the product of the probability of occurrence of certain environmental conditions times the probability of exceeding the dangerous angle must be near to zero. The measure of survival capability is still a probability, namely, the one's complement of dangerous event

$$P(S) = 1 - P(D)$$

As extreme rolling can be the result of cumulative build up of roll due to a sequence of waves, according to Bayes' formula the risk of dangerous event can be given as

$$P(D) = \sum_j P(A_j)\, P(D|A_j) = \sum_j P(A_j \cdot D)$$

also using the axiom of total probabilities.

The first term $P(A_j)$ takes into account the environmental condition of the numerical experiment. If the presence of a wave group is assumed with a number j of successive waves exceeding a given maximum amplitude H_{max}, the hypothesis of event A_j can be given as

$$P(A_j) = P(j) = p^{j-1}(1-p)$$

where p is the probability of the simulated wave and the parameter j is given by the formula [9]

$$j = \left[1 - e^{-2(H_{max}/H_s)^2}\right]^{-1}$$

Here H_s represents the significant wave height of the wave group.

In the case of damage accident the designer can be interested in assessing risk due to rolling in very short-term situations leading to dangerous oscillation up to capsizing. That may occur if the sea spectrum becomes so narrow that it can be associated to waves whose profile resembles a slowly varying wave group. Thus, the sequences of waves will be modelled by means of a wave data basis with given probability distributions of heights and periods using the narrow-band wave spectrum concept. This one is described by a dominant frequency ω_0, a mean zero-upcrossing period T_Z, and an average wave length L_W, all of them defined in the domain of the number of waves in terms of the spectral moments.

In the time-domain analysis the amplitude expected value of the highest wave and the other characteristics of a given sea state have to be obtained for the required simulation time T_s, that is, for the number of cycles $N = T_s/T_Z$. The probability of an individual peak amplitude ξ_i of N independent waves to exceed a threshold level ξ_{max} for a certain sea spectrum of variance m_0 can be modelled as a stationary Gaussian stochastic process, described by the theoretical Rayleigh distribution

$$p = P(\xi_i > \xi_{max}) = 1 - \left[1 - e^{-\xi_{max}/2m_0} \right]^N$$

where for a narrow frequency band ocean wave the most probable wave amplitude in N cycles is related to the rms wave amplitude by the formula [13]

$$\xi_{max} = \sqrt{2m_0}\,[(\ln N)^{1/2} + 0.2886\ (\ln N)^{-1/2}]$$

The second term $P(D|A_j)$ of the formula for the total probabilities represents the conditional probability of the occurrence of exceeding the dangerous roll angle ϕ_D under the hypothesis of event A_j. It is derived as the one's complement of the cumulative probability distribution of the standard deviation of roll in the (H_{max}, T_0) plane up to the dangerous angle.

The wave profile is simulated as a regular wave whose wave amplitude is the most probable maximum corresponding to the selected energy spectrum peak. It is written as

$$\xi_j(x,t) = \xi_{max} \cdot \cos\,[k_0 x - \omega_0 t + \delta_j]$$

where x is the location of each transverse section with respect to the wave taking into account the ship relative motion whilst δ_j is a random phase angle chosen to follow an uniform distribution of probability within the range $(0,2\pi)$ by a random number generator. When the peak frequency ω_0 is considered, the wave number k_0 corresponds to the wave length assumed to have a maximum steepness given by the relation [9]

$$L_w = H_{max}/(0.151 - 0.0072\,T_0)$$

A NUMERICAL SIMULATION

The probability of damage survivability has been computed assuming to run the experiment in a sea described by the JONSWAP spectrum when using North Sea wave data [8]. Instead of representing the short-term sea by its single wave-energy spectrum and then deriving random wave amplitudes whose finite sum reproduces the sea, a family of wave spectra has been selected for the most probable peak periods and covering a meaningful range of significant wave heights. Thus, each environmental event A_{ij} has been represented as a short-term wave condition whose kinematics are derived from the value of the selected peak frequency and rms height. A number of 50 random phase angles uniformly distributed has been assumed for each wave of simulation. The conditional probabilities of dangerous events have been weighted with the probabilities of occurrence for each peak period. Therefore, the risk of dangerous event can be described as

$$P(D) = \left[\sum_{i=1}^{m}\sum_{j} w_i \cdot P(A_{ij}) \cdot P(D|A_{ij})\right] / \sum_{i=1}^{m} w_i$$

where m is the number of sea spectra of the family and w_i is the probability of occurrence of a (H_{max}, T_0) pair, that is, the fraction of the total number of observations for each T_0 to the total sum of observations for all the recordered periods.

The architecture of the computer code for the motion simulation of a damaged ship requires a previous storing in a suitably arranged data base of the results produced off-line by the modules relative to the variables that are dependent on a few parameters only (hydrostatics, hydrodynamic coefficients, wave forces). During the time-domain simulation the system of differential equations for forecasting ship motions are integrated at each time step to give the effective hull geometry with respect to the instantaneous wave profile. The driving frequency is then derived and the actual values of hydrostatics and hydrodynamics are interpolated after sorting.

The simulation runs on the VAX 8820 computer at the University of Trieste. The mean time required to compute the ship motion in a seaway with floodwater in damage compartments is about half a second of CPU time for every second of real-time motion.

Here the investigation is discussed only for one experiment. A cross-channel ro-ro passenger ferry has been used to simulate the flooding phenomenon in a stochastic sea. Principal particulars of the ship together with a small scale body plan and layout of compartments are shown in Fig. 2. The ship results not to comply with the IMO probabilistic damage stability regulations [21], achieving a survival probability of 58.2%, while the required subdivision index amounts to 70.2%. The operative and damage conditions, considered as a numerical example, are described in Fig. 3 together with the responses in heave, roll, and pitch motions. The heel angle $\phi_D = 12°$, recommended by IMO [23] as the limiting angle not to be exceeded during intermediate asymmetrical stages for flooding two compartments, was considered as the dangerous roll angle. The simulation time was assumed to be equal to 10 minutes, with the ship at rest ($\phi = \dot{\phi} = 0$) at time t =0. The flooded compartments were considered to be partially filled (~15%) when the flooding started. The summary of the results of the numerical simulation is given in the following table.

Spectrum	1	2	3	4	5	6	7
T_0	6.0	7.5	9.0	10.5	12.0	13.5	15.0
m_0	0.118	0.207	0.385	0.551	0.639	0.601	0.512
L_W	28.5	44.7	64.4	87.6	114.4	144.8	178.8
N	140	112	93	80	70	62	56
w_i	0.090	0.240	0.260	0.170	0.100	0.060	0.030
$P(A_i)$	0.016	0.035	0.107	0.105	0.103	0.243	0.244
$P(D\|A_i)$	0.012	0.031	0.139	0.178	0.222	0.200	0.165

It results a survival capability P(S) = 0.986 for the considered operative condition, sea state, position and extent of damage. The ship oscillates very irregularly in heave and roll as a result of the presence of nonlinearities at large amplitudes and of heave-roll coupling. The maxima correspond to the states when natural frequency of floodwater in roll is at near the excitation driving frequency (Fig. 4). Differently from the preliminary conclusions by Petey who studied the same phenomenon in beam seas [16], here the floodwater is not effective as a roll damper. On the contrary, it appears that flooding mechanism was exciting heave and roll motions when there was a large flow of flodwater on and off the compartment. Another important peculiarity is that the reduction of righting levers resulted to be more relevant than for the intact ship under equivalent conditions, particularly in the presence of resonance between ship driving excitation and floodwater motion.

CONCLUSIONS

The results of the numerical time-domain simulation of the damaged ship motions are to be analysed only from a qualitative point of view. Apart from the future improvement of existing modules and the introduction of lacking ones, they heavily depend on the initial conditions of ship motion and real seaway parameters. So, as it is evident to every naval architect who has not a blind faith in mathematics, also this model has to be experimentally validated with respect to its physical consistence. The present state of the art does not allow to derive simple and handsome guidelines useful during the design process as far as the implications of damage survivability on subdivision and hull form are concerned. Before reaching such a target, a deep and intensive screening of different approaches must be performed by testing their validity and reliability for routine use. For the time being, our goal is to gradually build up an appropriate model which can be useful to the designer as a qualitative tool to perform parametric analyses. As far as accuracy of predictions is concerned, nobody can dream to reach an exact solution in the near future also because time simulations of nonlinear dynamic systems cannot up to now result in reliable quantitative conclusions.

In the opinion of the author, a direct probabilistic approach is the better method to predict rare events like capsizing also in the case of flooding. In any case, improvement in understanding dynamic stability and updating of existing rules for damage stability are to be explained in terms of the traditional naval architecture. To this purpose, and also in order to investigate the parametric influence of hull form, size, and subdivision arrangement on damage stability in a seaway, the time-domain modelling approach seems to be promising and advisable. It could be included in a future stochastic optimization design. Last but not least, the time-domain approach is to be preferred to obtain a sample distribution for capsizing margin to combine with IMO marginal densities for extent and location of damage. That could be the way to define in a dynamic sense the "s-factor", that is, the probability of non-sinking or non-capsizing when defining the IMO "Attained Subdivision Index".

Acknowlegment

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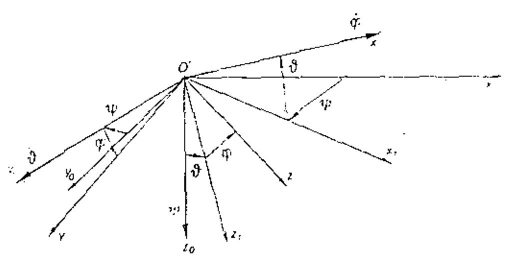


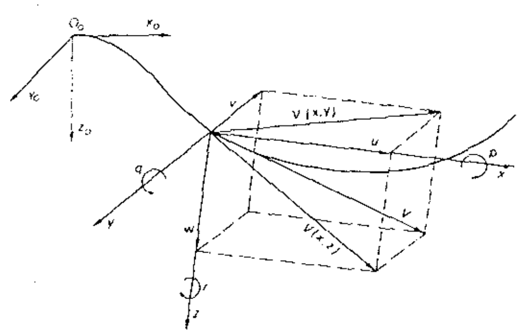
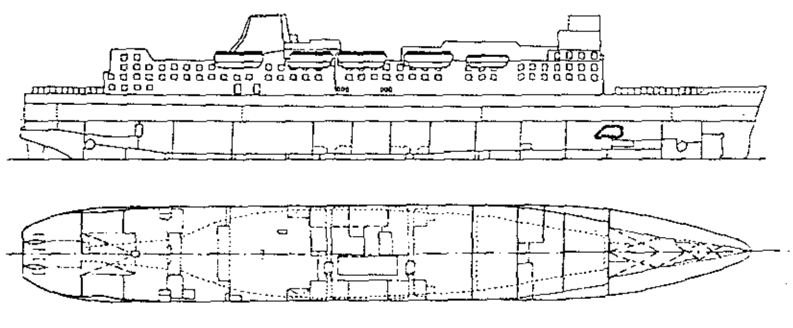


Fig. 1 : Coordinate systems



MAIN DIMENSIONS

L_{OA} = 146.00 m

L_S = 140.00 m

L_{PP} = 138.60 m

B = 18.40 m

D = 12.85 m

Fig. 2 : Ship main particulars and layout of compartments

EXPERIMENT CONDITIONS

Ship Data		Side Damage		Wave Data	
Displacement	= 9550 t	Length	= 6.00 m	Wave amplitude	= 2.485 m
Draught	= 5.85 m	Height	= 3.00 m	Wave frequency	= 0.734 s^{-1}
Trim	= 0.00 m	Z-bottom	= 4.00 m	Heading angle	= 0.00 deg
KG	= 7.81 m	Z-top	= 7.5 m	Phase lag	= 7.25 deg
Speed	= 0.0 knots	X-centre	= 122.25 m	Number of cycles	= 70

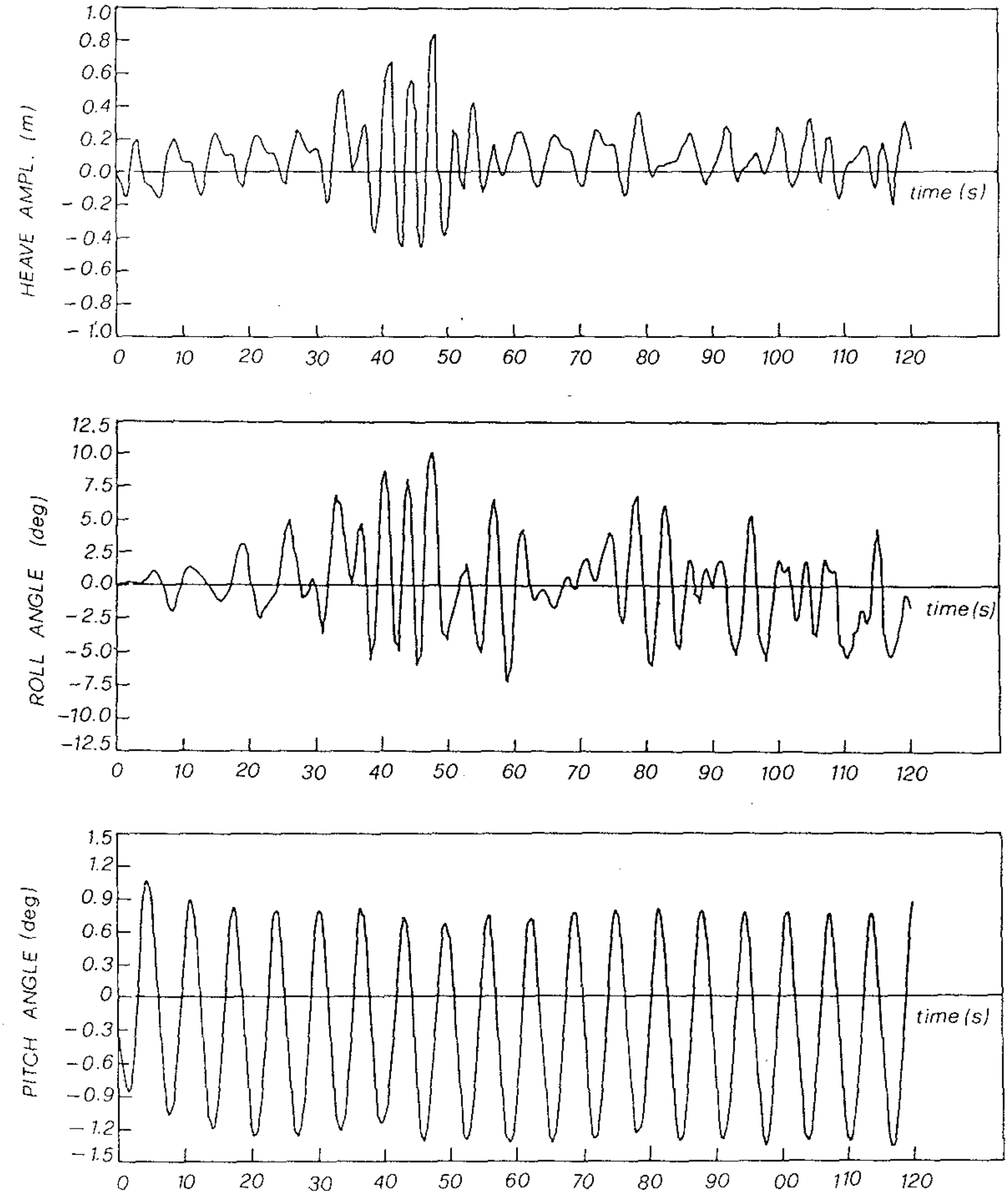


Fig. 3 : Responses in heave, roll, and pitch

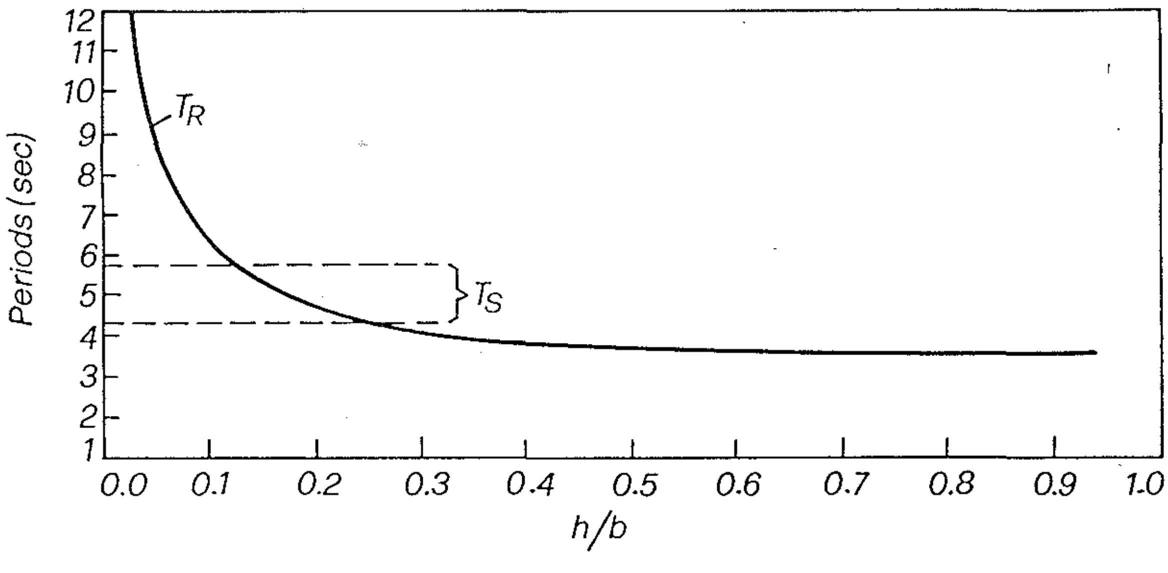


Fig. 4 : Syncronization between driving frequency and resonant liquid frequency in terms of compartment filling level.

Bifurcation Analysis of a Vessel Slowly Turning in Waves

Falzarano J.* Steindl A.† Troesch A.‡ Troger H.†

Abstract
The behavior of a vessel slowly turning in waves is studied as a nonlinear bifurcation problem. For two different shipmodels the oscillations of the ship are studied. Both external and parametric excitation effects resulting from the wave motion are considered.

1 Introduction

Vessel operators are frequently forced to change vessel heading in order to reduce undesirable rolling motions. This rolling motion in its two extreme cases results from two different types of excitation. One — and this is easy to understand — follows from the external excitation resulting from waves travelling in the direction vertical to the ship motion ([1,2]), that is, from beam seas. The resulting frequency of the ship roll motion is equal to the exciting frequency or is a subharmonic oscillation depending on the nonlinear components present in the system ([1]). The other type of excitation is parametric and follows from waves travelling in the same direction as the ship ([3,4]), that is from head or following seas. This type of excitation is best explained in [4]. The physical explanation for this parametric excitation is that the hydrostatic spring (see equation (3) below) is time dependent if waves of about the length of the ship travel along the ship. In this paper we intend to give an analysis of the roll behavior of a vessel slowly turning in waves. By slowly turning we understand that the angle ϑ of the direction of the ship motion, which will be the distinguished or bifurcation parameter, is varied quasistatically. Such a parameter study should reveal the spectrum of roll behavior between external and parametric excitation.

Another important point in our analysis will be to compare the results of two different mechanical models to describe the ship motion. The first is the frequently used one degree of freedom model describing only the roll motion. In this model the parameter values are independent of the exciting frequency and moreover the ship velocity is assumed to be zero. The second model has three degrees of freedom. We restrict to this case because it

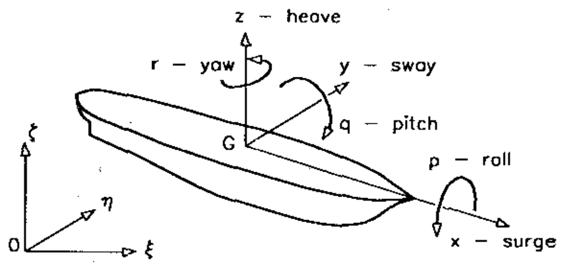


Figure 1: Vessel with 6 degrees of freedom

seems to be consistent with other assumptions especially with those of linear hydrodynamics which are used to calculate all hydrodynamic coefficients in their frequency dependency. In addition nonlinear damping and restoring moments obtained from experiments are introduced into this model.

2 Mechanical model and equations of motion

If one restricts the analysis to a *single* degree of freedom model the investigation reduces to a Duffing equation for the problem with external excitation ([1,2,3]) following from beam seas or an equation with parametric excitation following from head or following seas. In the case of quatering seas these excitations are acting simultaneously. The equation of motion can be written in the form

$$\begin{aligned}(I+A)\ddot{\varphi} + D(\dot{\varphi}) + \\ + GZ(\varphi, t, \vartheta) = K_e(t, \vartheta)\ ,\end{aligned} \tag{1}$$

where ϑ is the heading angle and A is the added mass. In this equation different authors have included different types of nonlinearities. First, besides a linear damping moment also a nonlinear one that varies quadratically with the roll velocity ([3])

$$D(\dot{\varphi}) = B\dot{\varphi} + D\dot{\varphi}|\dot{\varphi}| \tag{2}$$

is introduced. Second the nonlinear restoring moment can be given by the product of a time varying term and a fifth order polynomial curve (see [6])

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$$\Delta GZ(\varphi) = C_1\varphi + C_3\varphi^3 + C_5\varphi^5 \,. \qquad (3)$$

Usually the coefficients C_1 and C_3 are positive and C_5 is negative ([3]). Then for large roll amplitudes a softening spring characteristic is obtained which allows to describe capsizing.

In [5] a more accurate model describing the coupling of pitch and roll motion is presented. For this two degrees of freedom model nonlinear differential equations are derived. The system is studied in that parameter range where the pitch frequency is twice the frequency of the small free roll oscillations. Hence, the internal or autoparametric resonance case is studied.

The second and more realistic model we use has three degrees of freedom. They are the roll angle φ, the yaw angle α and the sway velocity v (Fig. 1). The choice of these degrees of freedom follows from a linearization of the full nonlinear equations of motion because then roll, yaw and sway decouple from heave, pitch and surge. Further much care is taken to include the frequency dependence of the coefficients. This dependence on the excitation frequency is calculated from linear hydrodynamics. Hence the added mass, damping and displacement terms in the equations of motion are frequency dependent. To these linear equations of motion a nonlinear roll damping term and a nonlinear roll restoring term are added. We write the equations in the following form

$$\begin{aligned}(\mathbf{M} + \mathbf{A}(\omega))\ddot{\boldsymbol{x}} + \mathbf{B}(\omega)\dot{\boldsymbol{x}} + \\ + \mathbf{B}_q(\omega, \dot{\boldsymbol{x}}) + \mathbf{GZ}(\boldsymbol{x}) + \\ + \varepsilon\mathbf{GM}(\omega)\cos\omega t\boldsymbol{x} = \mathbf{F}(\omega)\cos(\omega t + f_p) \,,\end{aligned} \qquad (4)$$

with $\ddot{\boldsymbol{x}} = (\dot{v}, \ddot{\varphi}, \ddot{\alpha})^T$. The 3×3 matrices and 3-vectors are: $\mathbf{M}$ is the mass and inertia matrix, $\mathbf{A}$ is the hydrodynamic added mass matrix, $\mathbf{B}$ is the linear damping matrix, $\mathbf{B}_q$ is the quadratic viscous damping vector, $\mathbf{GZ}$ is the hydrostatic restoring force and $\varepsilon\mathbf{GM}\cos\omega t$ is the time varying hydrostatic force, $\mathbf{F}$ is the linear hydrodynamic exciting force. The matrices or vectors $\mathbf{M}, \mathbf{A}, \mathbf{B}, \mathbf{B}_q$ and $\mathbf{F}$ are calculated by the program SHIPMO ([6]), $\mathbf{GZ}$ is calculated with the help of STAAF ([7]) and $\varepsilon\mathbf{GM}$ with a program developed in ([8]). The application of these various programs to calculate the coefficients is very important because these coefficients depend significantly on wave length, ship velocity and heading angle. As these quantities are not constant in a turning manoeuvre they have to be adapted to the actual parameter values at each step of the turning manoeuver.

The second order system (4) can be transformed into a four dimensional system of first order by introducing a new variable vector $\boldsymbol{y}$ defined by $y_1 = v$, $y_2 = \dot{\varphi}$, $y_3 = \dot{\alpha}$, $y_4 = \varphi$ and adding the trivial relation $\dot{y}_4 = y_2$. This set of equations is

$$\dot{\boldsymbol{y}} = \mathbf{L}\boldsymbol{y} + \boldsymbol{b}y_2|y_2| + \boldsymbol{c}(y_4, t) + \boldsymbol{f}(t) \qquad (5)$$

where all coefficients are still depending on the wave frequency ω_e. As the vessel turns with constant forward speed under variation of ϑ, the encounter frequency ω_e changes due to the following relation

$$\omega_e = \omega_0 - Vk\cos\vartheta \qquad (6)$$

where ω_0 is the wave frequency, V is the vessel speed, $k = 2\pi/\lambda$ is the wave number and λ the wave length. The heading angle $\vartheta = 0$ if the ship is travelling in following seas and $\vartheta = \pi$ if the ship is travelling into head seas.

3 Method of Analysis and Numerical Results

The branching behavior of solutions of ordinary differential equations under variation of one or several parameters has been extensively studied over the past twenty years ([9,10,11]). These efforts have been supplemented by work in numerical analysis which resulted in program packages like BIFPACK ([12]) or AUTO ([13]). Such programs have automated the calculation of the bifurcation behavior of the solutions of a set of nonlinear ordinary differential equations under variation of a parameter. For this work BIFPACK is used with the data for the fishing trawler Patti B which capsized twice.

3.1 One degree of freedom model

For this model the frequency dependence of the coefficients is not taken into account.

In order to take care of the time variation of GZ we assume that $GZ(\varphi, t, \vartheta)$ can be given by

$$\begin{aligned}&GZ(\varphi, t, \vartheta) \\ &\quad = (1 + \varepsilon GM\cos\vartheta\cos\omega t)\Delta GZ(\varphi) \,.\end{aligned} \qquad (7)$$

Further we must also include the dependence on the angle ϑ in the external excitation term which results in

$$K_e = F\sin\vartheta\cos(\omega t + \gamma) \,. \qquad (8)$$

Depending on the value of the heading angle ϑ we obtain pure parametric excitations for $\vartheta = 0$ and π and pure external excitation for $\vartheta = \pi/2$.

For the results obtained with this model we assume the speed of the ship to be very small, that is we neglect the effect given by (6).

The wave length is considered to be equal to the length of the ship (≈ 75 ft). For the height h of the waves measured from crest to trough we perform the calculations for two cases: $h_1 \approx 6$ cm ($\eta = 0.1$ ft) and $h_2 \approx 1.5$ m ($\eta = 2.5$ ft) where η is the amplitude of the exciting waves.

The results of the calculations, namely the amplitude A of the roll motion, are shown in Figures 2–5 for $h = 6$ cm and in Figures 6–8 for $h = 1.5$ m. The eigenfrequency of the ship's roll oscillation is denoted by ω^* in the figures.

3.1.1 Low amplitude exciting waves

The amplitude η of the exciting waves is 0.1 ft. Fig. 2 shows the case of pure parametric excitation ($\vartheta = 0$). There exists only a narrow region on the ω-axis about $2\omega^*$ where a roll oscillation is excited. Outside this region no roll motion exists. The frequency of the oscillation is one half of the frequency of the exciting waves ($2T$-periodic oscillation). Though there is a stable branch at high amplitude oscillations, this cannot be reached by a pure parametric excitation with small wave amplitude. The maximum amplitude which can be obtained is about 10 degrees. Fig. 3 gives the purely externally excited roll motion ($\vartheta = \pi/2$). We notice that now in the neighborhood of ω^* much larger amplitudes can occur compared to the parametrically excited ship oscillation. In fact at resonance oscillation amplitudes are possible up to a magnitude of 40 degrees. Now the oscillation has the same frequency as the exciting water waves (T-periodic oscillation).

The most interesting case is the combined external and parametric excitation given in Fig. 4 for $\vartheta = \pi/4$. In some sense it is a superposition of the oscillations shown in Figs. 2 and 3. However we note that both oscillations are reduced in their magnitude compared to their pure occurrence. From Figs. 2–4 the behavior of the ship

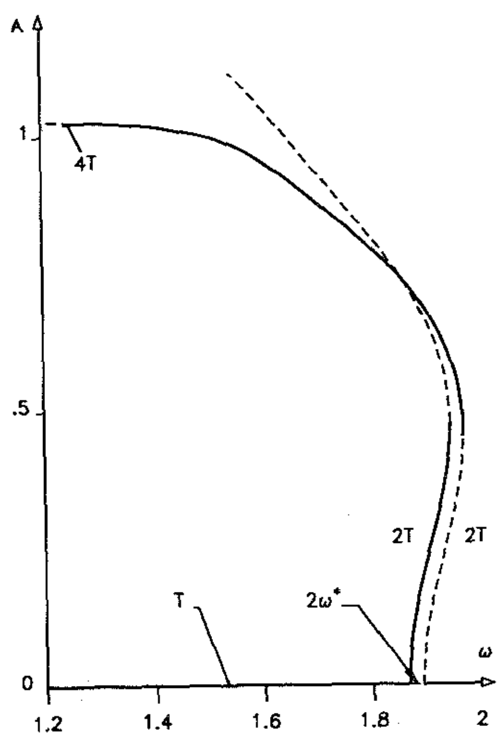

Figure 2: Amplitude A of the roll motion versus excitation frequency ω for wave amplitude $\eta = 0.1$ ft and heading angle $\vartheta = 0$.

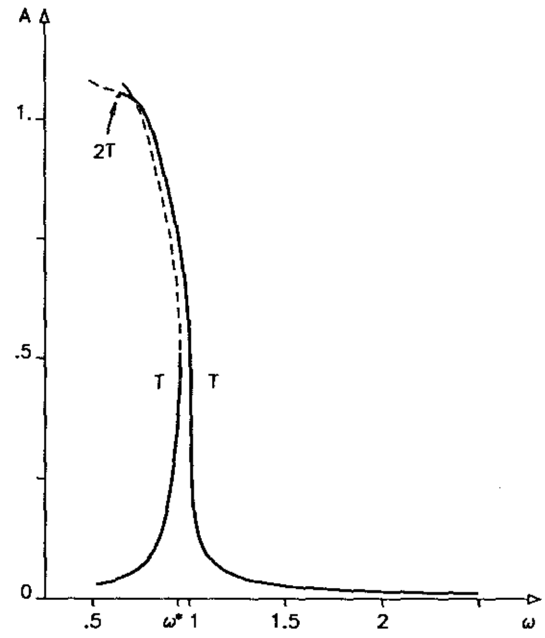

Figure 3: As in Fig. 2, but for $\vartheta = \pi/2$.

turning in small amplitude waves can be summarized as follows. A parametric excitation is very unlikely because of the narrow frequency range for which it can happen. An external excitation is always present if $\vartheta \neq 0$ or π. This external excitation can lead to large amplitude oscillations if $\omega \approx \omega^*$. If $\omega \approx 2\omega^*$ then in addition a $2T$-periodic oscillation is superimposed to the T-periodic oscillation. However both are of small amplitude in this case.

In Fig. 5 the dependence of the T-periodic oscillation amplitude A, that is the externally excited oscillation, in its dependence on the turning angle ϑ is shown for certain values of the excitation frequency ω. We see that keeping ω fixed and varying ϑ quasistatically, jump effects can occur leading to discontinuous changes in the amplitude of oscillation.

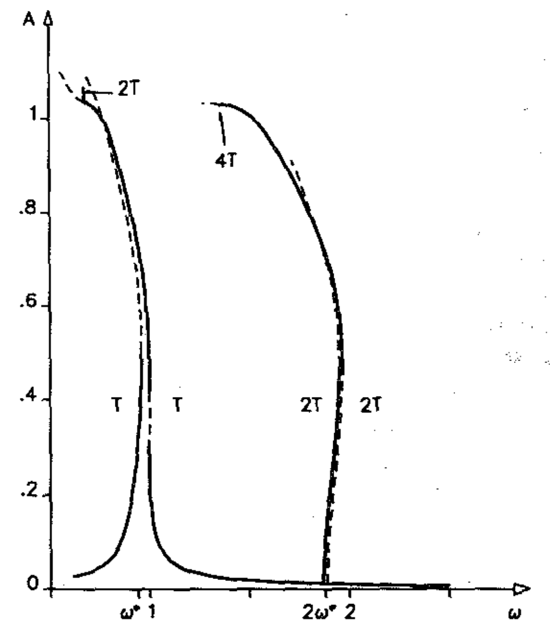

Figure 4: As in Fig. 2, but for $\vartheta = \pi/4$.

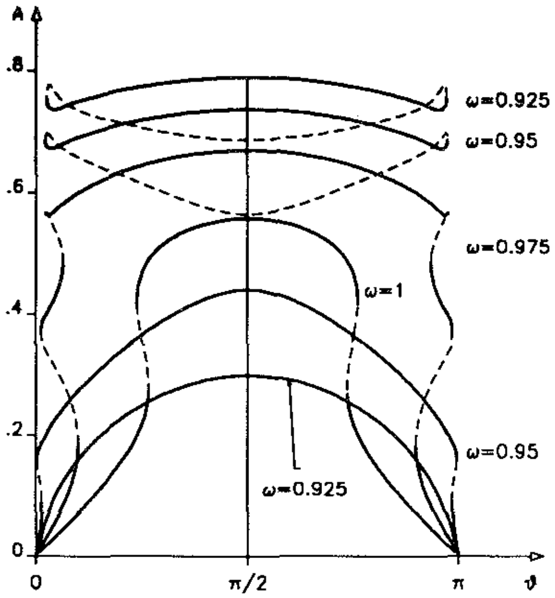

Figure 5: Amplitude A of the roll motion versus the heading angle ϑ for various values of the excitation frequency ω.

3.1.2 High amplitude exciting waves

In Figs. 6–8 the height of the exciting waves is $h = 1.5\,\mathrm{m}$ (that corresponds to an amplitude $\eta = 2.5\,\mathrm{ft}$). From Fig. 6 ($\vartheta = 0$) it is obvious that now compared to the results of Fig. 2 for the purely parametrically excited problem the domain of in-

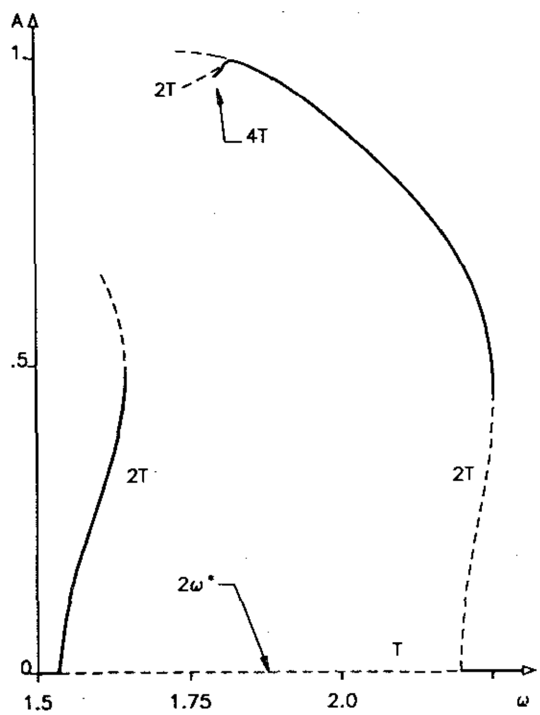

Figure 6: As in Fig. 2, but for $\eta = 2.5\,\mathrm{ft}$ and $\vartheta = 0$.

stability is of practical importance and moreover the amplitudes of the $2T$-periodic oscillation are of considerable amount ($\approx$ 50 degrees). In addition a period doubling bifurcation sequence exists which can lead to the consequence that in a narrow frequency domain even a chaotic ship roll motion may exist. This frequency range is most dangerous because in the chaotic domain oscillations can occur which include theoretically the turning over of the ship. Therefore in this frequency domain capsizing is possible.

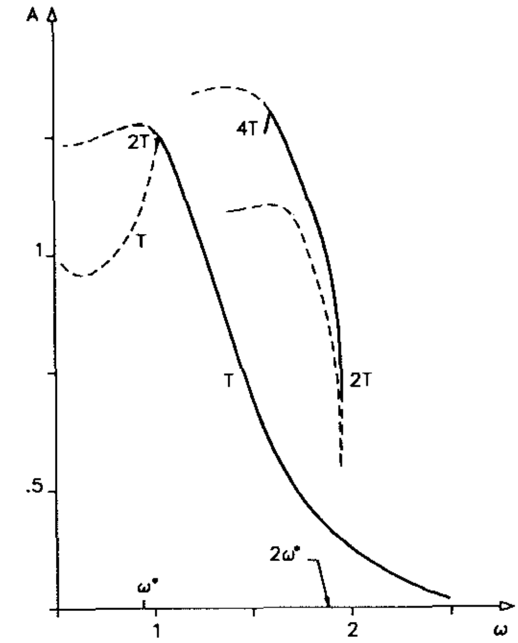

Figure 7: As in Fig. 2, but for $\eta = 2.5\,\mathrm{ft}$ and $\vartheta = \pi/2$.

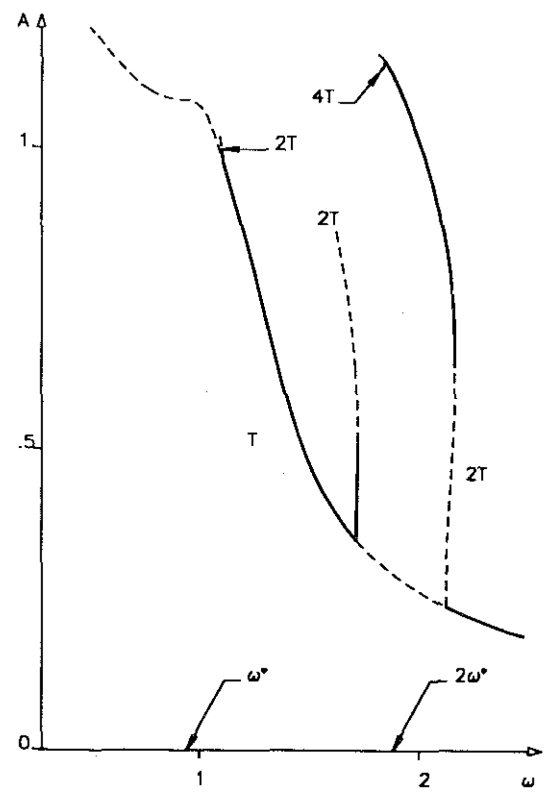

Figure 8: As in Fig. 2, but for $\eta = 2.5\,\mathrm{ft}$ and $\vartheta = \pi/4$.

Fig. 7 shows the purely external excitation ($\vartheta = \pi/2$). We see that in the interesting frequency domain only the right T-periodic branches exist and in reducing the excitation frequency a period doubling sequence starts. Again one can expect to obtain chaotic motions which can lead to capsizing of the ship.

Fig. 8 ($\vartheta = \pi/4$) shows for the combined excitation the existence of oscillations which are a superposition of T- and 2^nT-periodic oscillation.

3.2 Three degrees of freedom model

For this model we focus our numerical investigation on the frequency range where only the external excitation is relevant. That is where the encounter frequency is close to the eigenfrequency of the roll motion of the ship. In this case no parametric excitation effects occur and we can expect to obtain results which in principle are similar to

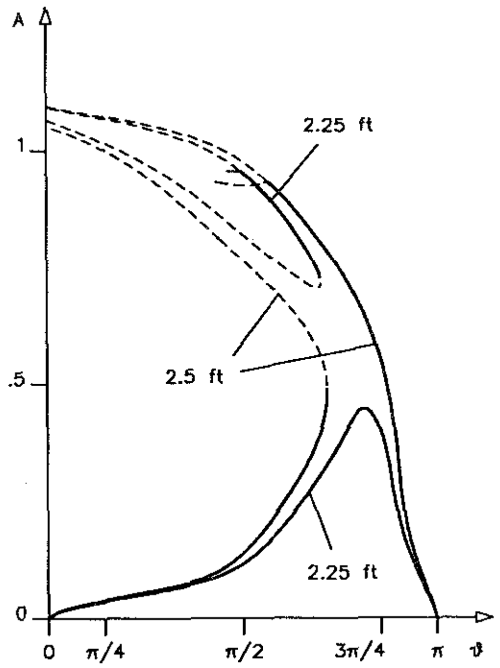

Figure 9: Amplitude A of the roll motion versus heading angle ϑ for different wave amplitudes $\eta_1 = 2.25$ ft and $\eta_2 = 2.5$ ft.

those shown in Fig. 5. In Fig. 9 the amplitude of the roll motion in its dependence on the heading angle ϑ is presented for the two wave amplitudes $\eta_1 = 2.25$ ft and $\eta_2 = 2.5$ ft. For $\eta = 2.25$ the physically relevant solution curve is connected and smooth and returns to zero after a turn of the ship of 180 degrees. The amplitude of the oscillations of the ship reaches a maximum value which however generally is not at $\vartheta = \pi/2$, but at that value of ϑ, where the encounter frequency ω_e is close to the eigenfrequency of the roll motion obtained for the linear ship model. Thus a different roll behavior is obtained in the course of the variation of ϑ depending on whether the ship turns off from following or head seas.

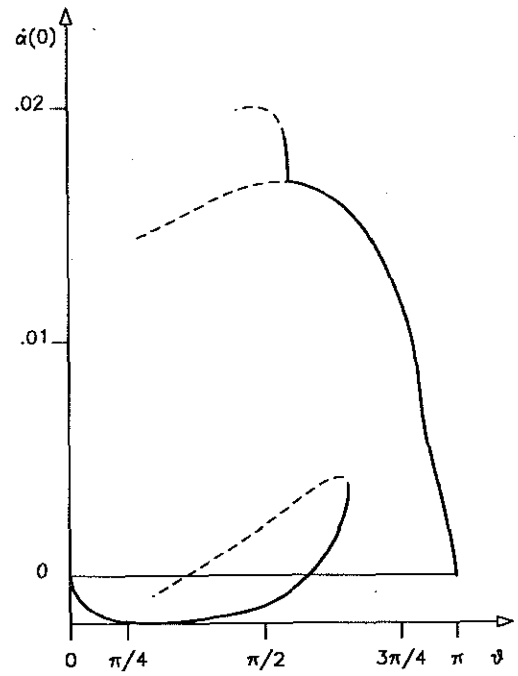

Figure 10: Solution diagram in $(\vartheta, \dot{\alpha}(0))$space.

Even more pronounced is this difference in the behavior if the wave amplitude $\eta = 2.5$ ft is considered. In this case we see that starting in following seas at $\vartheta = 0$ the roll motion has a jump at about $\vartheta = 5\pi/8$ and then the roll motion reduces to zero at $\vartheta = \pi$. However if the ship starts to turn from head seas ($\vartheta = \pi$) we see that the amplitude increases steadily and at quite large values secondary bifurcations start which again will lead to a chaotic motion possibly leading to capsizing.

In Fig. 10 we show that jumps of the yaw velocity and secondary bifurcations occur at the same frequency values as in Fig. 9. The same behavior is observed for the sway velocity shown in Fig. 11.

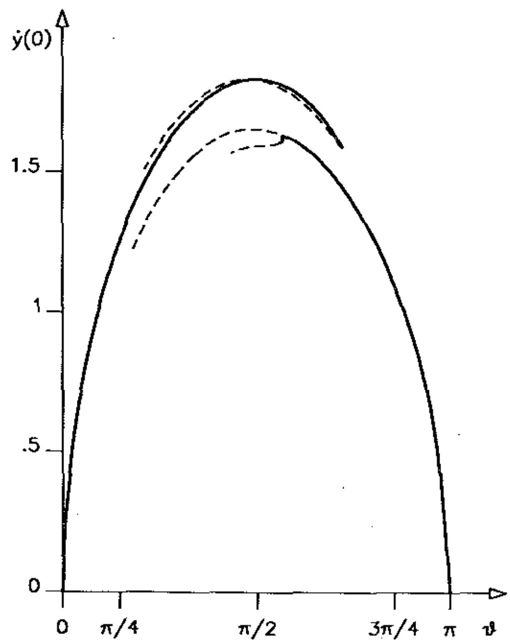

Figure 11: Solution diagram in $(\vartheta, \dot{y}(0))$space.

4 Summary

For a range of parameter values the periodic response of the ship to regular waves has been calculated. If these periodic solutions are stable they attract neighboring trajectories. Moreover these stable solutions persist under small variations (perturbations) of the system. Further multiple attractors and unstable solution branches are detected. The latter provide additional information on the domain of attraction of the stable solutions.

The comparison of the results obtained for the two different models shows that the difference in the behavior of the ship — if it turns into following or head seas — could not be calculated from the simple one degree of freedom model, where the forward speed of the ship was not taken into account. Further the interesting result can be deduced from the three degrees of freedom model that jump effects or secondary bifurcations occur in all degrees of freedom simultaneously.

Acknowledgement

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A PREVENTIVE FRAMEWORK FOR ACHIEVING EFFECTIVE SAFETY

C Kuo

The paper highlights the importance of stability for ship and ocean vehicles and reviews the progress of the past fifteen years and its practical applications. The needs of six relevant groups are then assessed and existing and possible methods of meeting them are examined. The terms "safety" and "the desired goal" are defined before the PREVENT-IT strategy is proposed as an effective way forward. This is considered as part of an overall treatment of safety for floating offshore systems, and its practical application is discussed with the aid of an illustrative example. Recommendations are made and tasks deserving special attention are outlined.

1. INTRODUCTION

The importance of stability for ships and ocean vehicles is clear to everyone, but it is of particular significance for those with either direct or indirect responsibility for its implementation. It is therefore highly appropriate that, from time to time, interested parties, representing operators, owners, governments, designers, researchers, or users, should gather together to examine the key issues of this subject.

Looking back, it could be said that the First Conference on the Stability of Ships and Ocean Vehicles, in Glasgow (1975), was motivated by a thirst for knowledge, the Second, in Tokyo (1982), by the urge to consolidate the knowledge acquired, and the Third, in Gdansk (1986), by the need to take stock. This present conference in Naples (1990) should be seen as an opportunity for the practical application of knowledge and experience, and for identifying clearly the direction for the 1990's and beyond.

Since 1975 we have witnessed a number of highly publicised and tragic marine disasters. We think of the "Alexander Kielland" (1979), the "Ocean Ranger" (1982), the "Marques" (1984), the "Herald of Free Enterprise" (1987) and Piper Alpha (1988), together with less well publicised but equally tragic losses in Bangladesh, China, Indonesia and the Philippines.

It is right and proper for those who use ships and ocean vehicles to require that their journeys on the seaways should be safe. It is equally valid for the owners/operators to seek profit from operating highly competitive international businesses, for governments to safeguard standards and for designers to provide the technical developments to these ends. But the requirements of all the interested groups have to be positively integrated and interfaced, if the demands of the users are to be met. We have made considerable progress in the past fifteen years,in terms of international scientific output but we do not have a well-defined way forward. Without this clear direction, it will not be possible to devise sound strategies to fulfil these requirements. If we are to achieve credibility here, it is essential for all of us: operators, designers, legislators and researchers, to work together in order to achieve the desired safety goal.

The aim of this paper is to put foward some new ideas for treating the stability of ships and ocean vehicles in the 1990's. It will begin with a critical review of the progress of the past fifteen years, followed by a consideration of the needs of each group involved with stability. Existing and possible solutions are considered

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before the term "safety" and the desired goal are defined. A total safety framework based on the PREVENT-IT strategy is then outlined and its practical implications are examined. The summary of an illustrative ship example demonstrates the practical application of the PREVENT-IT strategy.

2. BRIEF CRITICAL REVIEW OF PROGRESS

What contributions have been made here since 1975? In seeking to answer this question it is necessary to consider the efforts of two broad groups: those carrying out individual studies and those involved with major research programmes.

Many individual investigations have been done during this period. Reports on some of these can be found in the Proceedings of the last three international conferences, and more will be given during this present symposium. Others have appeared as special contributions to the journals of various professional institutions and other publications. At the same time there have been a number of research programmes in stability supported by various governments and industrial concerns, see Table 1 and Refs. **(1)** to **(6)**.

The key findings of these investigations can be considered under four main headings:

Regulations

For ships, a number of new regulations have been introduced, the latest being SOLAS 90 for passenger ferries. As a rule these apply to new designs introduced after a specified date. In general these new regulations represent a greater stringency in requirements, based on the experience gained from accidents or disasters involving certain types of ship.

For Mobile Offshore Drilling Units (MODUs) such as semisubmersibles, the stability criteria demand positive righting arms and metacentric height and a specified minimum ratio between righting and heeling net areas. In spite of the increasing amount of evidence from theoretical work and model experiments, regulations for MODUs have not altered since their introduction in 1968, i.e., the area ratio has remained at 1.3. Research studies have, however, indicated that this ratio value can be reduced without reducing the level of safety, and that the use of a ratio is not the most consistent and reliable method of assessing a MODU in damaged condition.

Theoretical Studies

Theoretical studies in this field can be groups under three broad headings, as follows:

a) Conventional Approaches with Greater Sophistication: The most popular research efforts have been towards extending the existing methods of treating stability by incorporating additional parameters, or introducing other effects, into the assessment. Typical examples of these include extra degrees of freedom, non-linear coefficients, and coupling other motions with rolling. See, for example, Refs. **(7)** to **(10)**. In general these studies have been motivated by advances in computer simulation techniques and the availability of more versatile computer hardware and software.

b) Fresh Approaches: Typical of these are approaches involving the application of Catastrophe Theory, **(11)**, and the Chaotic Principle, **(12)**. The basis of these treatments is the identification of situations in which a ship is likely to capsize. There is no doubt about the attractiveness of such treatments, and efforts in this direction should be encouraged. In the next decade these approaches should be adopted for the identification of potential hazards, but considerable effort is necessary, together with other improved and more practical methods, if the results are to be employed in practice.

c) Practical Approaches: Work in this category is based on acceptance of the limitations of theory and attempts are therefore made to employ the available knowledge in sing the freeboard of an existing Ro-Ro ferry, for example, would increase its residual stability.

The main advantage of such an approach is its ability to improve the stability of an existing vessel without the need to make major structural alterations. The main disadvantage, however, is a major one. This approach often leads to a significant loss of load-carrying capacity. It is estimated, for example, that increasing by 50% the freeboard requirements of a typical Ro-Ro ferry operating on the Dover-Calais route could reduce its earnings by 20%. Operators would be unhappy about such a move as, likewise, would operators of semisubmersible support vessels, which are always short of payload capacity.

Further Research Approach

It can be argued that the most likely causes of failure are a lack of knowledge, or a specific engineering or operational factor, or an intereaction of these. Progress can only be made by bridging the knowledge gap through doing research on the problems identified.

The main advantage to be gained from doing appropriate research will be the assurance that designs, stability regulations and operating procedures have all been put on a sounder foundation.

The main difficulty here is associated with the applicability of the results. It is all too easy to do research in a very narrowly-defined area, without making provision for testing how practical implementation of the research findings will affect other parameters of the vessel.

Design Modification Approach

Since the stability of ships and ocean vehicles is usually determined at the design stage it has been argued tha many disasters could be avoided if safety were a major factor in the designer's minds. This would overcome the problem of designers simply responding to operators' pressure regarding cost or practical efficiency.

The main advantage of this approach is its positiveness, and it is indeed true that, within the regulatory requirements, the designer can achieve a good solution early in the design stage. The main drawback, however, is that such an approach is only really suitable for new designs.

It is possible to identify many other approaches, but none are likely to meet practical needs. It is therefore necessary to consider a fresh treatment of this problem.

5. SAFETY AND THE DESIRED GOAL

Before putting forward an approach that has the potential to meet the practical needs which were identified earlier and also a general applicability for dealing with the wider issue of safety, it is essential to define clearly what is meant by "safety" and what should be the goal of ship and offshore operators.

"Safety" is a word in frequent use in many different contexts, but the users' understanding of its meaning can vary widely. This can range from the dictionary definition of "freedom from danger", to the commercial one of "not losing money". Safety however, is not an absolute quality, and its interpretation varies with the differing perceptions of those in the particular situation. For example, an experienced seafarer would not be worried about his safety in a ship with a ten-degree roll in heavy seas, but a novice in the same circumstances might well be very alarmed.

Safety in the present context may be defined as follows:

> "Safety is a perceived quality that determines to what extent the engineering and operation of a system would be free of danger to lives, property and the environment".

The desired goal of operators should be to incorporate into a design both operational safety and minimised cost. Their goal can therefore be stated as follows:

> "To meet the operator's specifications to an agreed standard while at the same time satisfying the demands for cost effectiveness and an acceptable level of safety."

There are three separate and distinct criteria to be met here, i.e., operational specifications, cost, and safety. Each on its own can usually be readily satisfied but the challenge is to meet all three simultaneously. Technologically this can be regarded as a multi-level and multi-variable optimisation problem Unfortunately, the relationships between the variables cannot be defined by analytical equations or numerical relations because they are closely inter-related and highly complex. In addition, some of the elements involved are qualitative rather than quantitative.

6. A PREVENTION FRAMEWORK FOR TOTAL SAFETY

Achieving the goal indicated above is a very challenging task but the use of the prevention concept does offer a logical way of dealing with many practical situations. The basis of this concept can be readily appreciated if stated as follows:

> "To anticipate potential or expected failures by taking active steps to minimise the likelihood of their occurrence or to limit their effects."

In applying the concept in the marine context much can be learned from the practice of preventive medicine. In the latter case the key activities are as follows:

- Predicting likely illnesses
- Research into their likely causes and consequences
- Evaluation of possible actions by the individual concerned
- Investigation into ways of eliminating or minimising the risks
- Preparing for possible emergency action as required
- Providing guidelines and good procedures to extend the range of effective prevention
- Encouraging the adoption of a healthier lifestyle.

Closer examination of this list reveals two important factors. Firstly, such an approach could bé applied to technological situations, and secondly, such an application need not be limited to stability but could embrace the total safety of the ship, or ocean vehicle, or indeed any system at all.

Three elements of management, technology and operation are needed for such a total safety framework. Management will determine the safety objectives and offer general support. Technology provides the methodology and the means for achieving the objectives. The tasks of operation are practical application and acquiring feedback of experience gained. This paper will focus on the technology element and use the PREVENT-IT strategy as a basis for safety improvement in the 1990's. The approach is a generalisation of 20 years' research: see its application to offshore installations, **(16).**

7 THE BASIC STEPS OF THE PREVENT-IT STRATEGY

The PREVENT-IT strategy consists of nine steps, and the tasks involved, illustrated in Fig. (1), are highlighted here. It should be noted that although the emphasis is on ships as such, the term "ship" here may also imply an ocean vehicle or any floating offshore installation.

STEP 1: Predicting Potential Hazards

Once the basic specifications of a ship are defined or the specifications of an existing ship are known, the potential hazards should be predicted for a number of practical scenarios. For a vessel such as a Ro-Ro ferry, the potential hazard-factors would include:

- Stability and possible capsize
- Fire and smoke
- Flooding

situations.

The practical implementation of this strategy is summarised in a case example in Appendix 1.

8. SOME IMPLICATION OF APPLYING THE PREVENT-IT APPROACH

We have argued that the concept of prevention and the PREVENT-IT strategy is the approach to be implemented in the 1990's and beyond. It would be useful, therefore, to seek answers to seven significant questions.

Can Designers Meet Specifications by Using the PREVENT-IT approach?

At present ship designers have to comply with a given set of regulations. Some of these regulations are based on firm scientific findings and sound practical experience while others are less well-founded. To a large extent designers have left the responsibility for safety in the hands of the legislators, even when they know, from their own practical knowledge and experimental evidence that the regulatory requirements are unrealistic. In view of the fact that, for good reasons, legislators are always behind technological advances the present practice gives designers little scope for initiative. The application of the PREVENT-IT approach, however, will allow designers to exercise their skills and experience to the full and to explore the opportunities for innovation. Because the framework is logical and based on a systems approach it can act as an aid to designers in their explorations, while the legislator still has the ultimate responsibility for ensuring safety in Step 8.

Would the PREVENT-IT Strategy Be Cost-Effective as Well as Safe?

The answer to this is a firm "Yes". Adopting the PREVENT-IT strategy means that although identical standards are demanded for all ships, variations in the risks to, and the operating conditions of, individuals are taken into account. Thus investment will be directed at the points where safety is most necessary. Typical examples of this would be:

- The use of robotic hardware combined with appropriate software to replace human operators in very routine tasks where boredom may lead to errors.
- Providing regular training for personnel in critical areas such as ferry-operation in busy traffic zones.
- Installing advanced collision avoidance systems as a back-up to navigation systems would do much to improve the operational safety of ships, and thus reduce overall costs.

Should Safety Not Really Be Left to the Operators?

Matters of ship safety have traditionally not been left to the operators for a number of reasons. The keys ones are the variations in standards imposed by different maritime nations, the fact that ships may operate in any part of the world, and the tendency of operators to adopt the minimum requirements possible in order to cut down operating costs and increase profit. It would, however, be a positive step forward to put the general responsibility for ship safety firmly in the hands of the operators, but leave the ultimate judgment to the legislators (Step 8) with the support of national governments and the co-operation of IMO. In the offshore industry it is accepted practice for operators to have higher standards of safety than those laid down in the rules and regulations.

What About the Owners?

Not all owners have specialised knowledge of shipping or ship operations, and so they are expected to leave technical decisions to those with the appropriate expertise. In general owners are wise investors and would accept the logic that "Prevention is better than cure and will also cost less in the long run!"

How Would Legislators View PREVENT-IT?

We can expect those involved with ship

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18. Risk Analysis for Offshore Structures and Equipment. ASTEO. Graham and Trotman, 1987.

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20. Offshore Installations: Formal Stability Assessment. UK Dept of Energy. PEA 89/93/4. October 1989.

APPENDIX 1
PREVENT-IT STRATEGY FOR RO-RO FERRIES

In this appendix the Ro-Ro passenger ferry has been selected as an example for demonstrating application of the PREVENT-IT strategy. Each of the nine steps is summarised to show how such a strategy can be readily applied in order to improve the quality of vessel safety.

STEP 1: Potential Hazard Identification

Some potential hazards connected with Ro-Ro passenger ferries are shown in the Table, together with an indication of their likely occurrence.

REF No	POTENTIAL HAZARD	LIKELIHOOD OF OCCURRENCE
1.1	Structural Failure	*
1.2	Equipment Malfunction	*
1.3	Operational Accident	* * * *
1.4	Fire in Machinery	* *
1.5	Fire in Accommodation	* * * *
1.6	Fire in Cargo Hold	* *
1.7	Explosion	*
1.8	Capsizing	* * *

Where: **** = Very High
*** = High
** = Average
* = Low

STEP 2: Research into Causes and Consequences

Using the capsizing of a Ro-Ro passenger ferry as a sample hazard, the causes of this may be either one or more of the following factors:

REF. No.	CAUSES
1.8.1	Loading Conditions
1.8.2	Cargo Shifting
1.8.3	Water on Deck
1.8.4	Flooding
1.8.5	Collision Damage
1.8.6	Design Fault
1.8.7	Human Error

A fuller review of causes has been compiled by W A Cleary. See "Subdividing Stability Liability" in Marine Technology, Vol. 19, No. 3. 1982.

STEP 3: The Required Level of Human Involvement

It is important at this stage to establish the level of human involvement in the identified potential hazards. Hence provision can be made for protecting crew or passengers as necessary, and for reducing the involvement of human beings in tasks that are repetitive or prone to error. Checking that bow doors are closed before cast-off, for instance, can be done more effectively by electronic means. Speedier response is also possible to a video-monitor on the bridge than to the signals of a crew-member positioned on the bow. In areas such as these, every effort should be made to use non-manual methods leaving human beings to concentrate on supervision and cross-checking.

STEP 4: Scope for Design Modifications

Possible design modifications could include:

- A lower position for the vessel's centre of gravity
- Increased freeboard
- A flared hull form
- Additional buoyancy units
- Incorporating a fast drainage system
- Automatically operated watertight doors.

STEP 5: Engineering Suitable Containment Systems

A number of possible containment systems can be introduced, including the following:

- Permanent ballast, or inflatable buoyancy units to keep the vessel afloat for a given period following damage
- Regular inspection and testing of equipment and facilities
- A more efficient cargo-lashing system.
- Reducing the possibility of different factors acting together to cause a major incident.

STEP 6: Nominating Emergency Provision

It is important to make emergency provision in case the containment systems do fail to cope with the results of an accident. These would include:

- Life boats launchable in all conditions
- Effective and speedy methods of evacuating passengers.

STEP 7: Transmitting Quality Requirements

Many accidents could be avoided or their effects minimised if the top management required high quality at every level. The benefits of this would include higher efficiency, greater savings and minimising of the risk of accidents. Requirements would include:

- The introduction of high standard operating procedures
- The employment of effective methods for measuring quality of performance
- The acquisition of the most effective hardware and software for each task.

STEP 8: Interface with Rules and Regulations

Steps 1 to 7 can be regarded as the "good practice" of any ship operator, but every vessel must also satisfy statutory rules and regulations, e.g., SOLAS 90 for new designs. It is at this stage that the procedures and assessments used by the operators should be found to satisfy regulatory requirements.

STEP 9: Training of Staff

There is a need to extend safety training programmes to the entire crew and aspects deserving special attention include:

- The implementation of safety procedures
- How to gather and feed back safety-related information
- Regular updating improved practices.

Panel Discussion I: **Operational Safety and Avoidance of Accidents**

Panelists:

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PLEADING FOR COMPARATIVE STUDIES IN THE FIELD OF SAFETY AGAINST CAPSIZING

Peter Blume [1]

In the meantime there are several proposals for mathematical models describing the capsizing phenomenon or even for new stability criteria. None of them is widely accepted. They are not even really discussed because there is no scale for the judgement of the methods. In hydrodynamics it is common practice to conduct comparative studies. I believe it is the time to do similar investigations for this topic and to find a generally accepted scale for the judgement of new methods.

Whatever method is used the final result of each method must be a limiting value for the height KG of the centre of gravity in a certain environment. We cannot expect that different methods come up with the same results. But that is not necessary. If the trend within different hull forms is in line with the experience, it is only a question of scaling.

At HSVA we have tested over the last 10 years 8 different models at different draughts and in various seaways always in the same procedure. So we have an unique treasure of experience which could be used for validation. Everybody who wants to contribute to the problem of a better stability assessment could determine the limiting KG for a number of selected hull forms. Depending on the KG some simple parameters preferably the maximum righting lever GZ_m and the total area E_0 could be calculated and compared. Due to our experience both parameters are suitable to distinguish between safe and unsafe. Then the trend within the different hull forms decides on the worth of the proposed methods. For a good method the trend should be in coincidence with the trend determined in our model experiments.

I will give a short example. I calculated for 7 hull forms maximum righting levers GZ_m for the limiting KG-values according to three methods:

1. Hull form factor concept proposed by the Federal Republic of Germany (FRG) in the modified form reported on this conference
2. Method proposed by German Demokratic Republic and Poland using righting lever curves on a wave crest (DSRK)
3. IMO-Recommendation A 167

For these 7 cases we also have experimental results all valid for the same conditions. The Figure 1 shows the maximum righting levers determined with 3 methods related to the maximum righting levers GZ_{mT} found from the tests.

The ideal case would be that all columns representing the same method have the same height. In reality we cannot achieve this due to an unavoidable scatter in the test results and simplification and insufficiencies in the methods. The standard deviation related to the mean value could be a measure of the quality of a method. The following table gives on overview of the mean values, the standard deviations and the latter related to the mean value (s_R):

Method	FRG	DSRK	A 167
GZ_m/GZ_{mT}	0.636	0.364	0.354
s	0.070	0.148	0.147
s_R [%]	11.0	40.7	40.5

The relative standard deviations show the superiority of the FRG-method. That follows from the fact that this method was conceived on the base of the experience with the largest systematic tests series I know which is used here as a scale, too. I believe the test results reflect the relative merits of different hull forms in a real environment. For both other methods the relative standard deviations are much larger indicating that these methods do not fit very well the behaviour of different hull forms.

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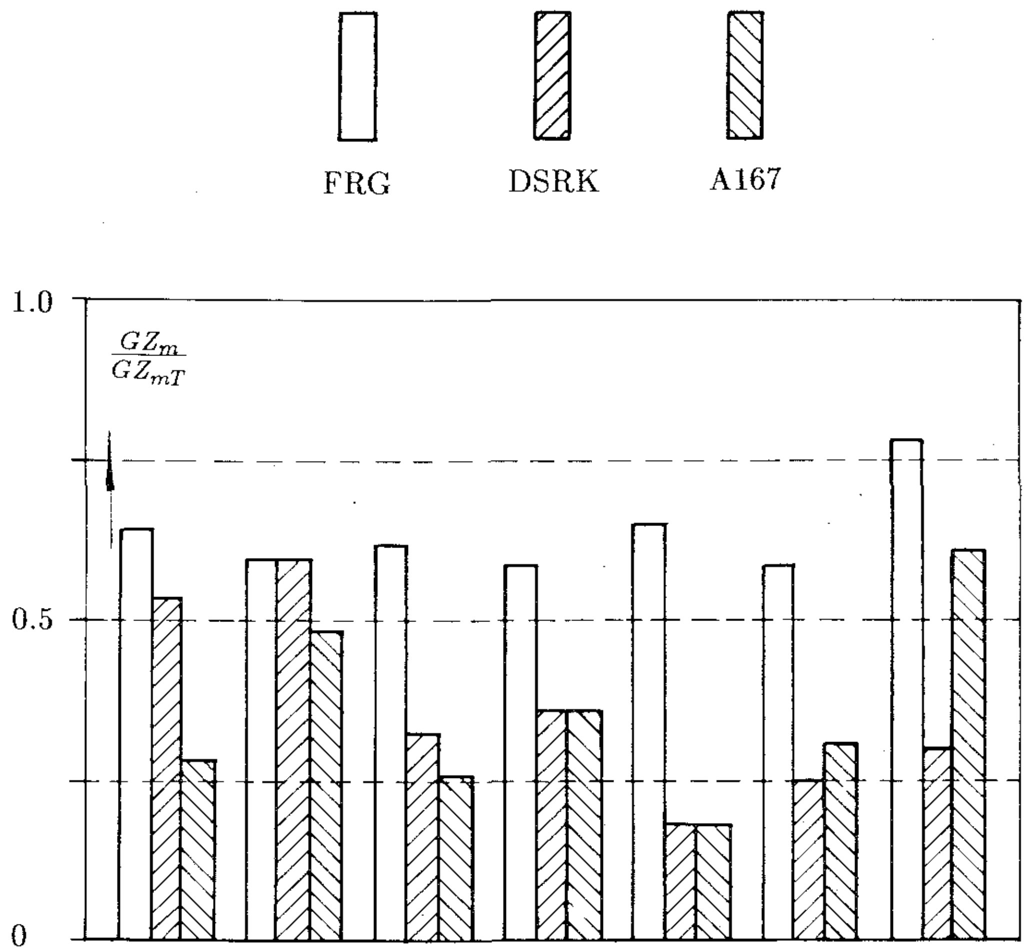


Model	A	C	E_1	F	F	G	G
L_{pp} $[m]$	135.0	133.3	135.0	135.0	135.0	135.0	135.0
B $[m]$	23.0	22.63	24.75	23.0	23.0	23.0	22.06
T $[m]$	8.20	9.28	7.80	7.70	5.00	7.70	5.00

Figure 1. Comparison of different methods
for stability assessment

PANEL DISCUSSION

ON

OPERATIONAL SAFETY AND AVOIDANCE OF ACCIDENTS

"Basic thoughts on operational safety in follwing waves."

by Seizo Motora Japan Foundation for Shipbuilding Advancement

Let us consider about intact stability of ships.

There are roughly two groupes of accidents related to ship stability.

Case 1. Difficult to avoid capsizing by operation.

Case 2. Capsizing may be avoided by operation in some exent.

Typical example of case 1 should be avoidance of capsizing in beam wind and waves. It may happen that a ship loses its means of control by stop engine etc, and unavidably exposed to severe beam wind and waves for a long time.

In this case, only means that crew can take are to close opnings to avoid flooding or fixing movable loads, ro lower heavy materials to improve stability. Since such condition is possible to occure, a ship must be desiged so that it can survive in an extreme severe beam wind and waves which are expected to ecounter in the life time.

IMO's stability criteria such as A167[1], A168[2], and A562[3] are the criteria based on the above mentioned idea, and are cailed "beam sea criteria". Though A167, A168 are the criteria based on experience and A562 is based on physical consideration of capsizing phenomena, the basic principle is to require ship builders to design a ship so as to survive in severe beam wind and waves for a long time duration.

In other words, in the case 1, ship should be protected for capsizing mainly by design, and operational sffort is regarded to be rather subsidiary.

On the other hand, typical example of case 2 would be stability in following waves. In this case, it has been well known that if a ship runs in following waves, there is specified range to conbination of ship speed, encounter angle and sea condition in which range, a ship will be faced to danger of capsizing, and out of which range, ship will be safe.

It has also been known that main cause of copsizing in following wave would be;

a) pure loss of stability

b) broaching to

c) parametric resononce

and that usually capsizing will take place by combined effect of the abobe causes. There is also an opinion[4] that period dubling bifurcation, which is an omen of a caos, is more impotant cause than parametric resonance.

Anyway, in case of stability of ships in following waves, a ship is supposed to maintain its mobility, and that's way there is a possibility of capsizing by running at a dengerous conbination of speed, direction and sea condetion, and also that's why a ship can avoid capsizing by operation. This situation is completely defferent than the case 1.

Then, how to avoid capsizing in following waves? There will be two extreme ways to zvoid caprizing in following waves:

a) To give the ship master a precise informattion in which dangerous range of combination of wave height, wave length, ship speed and angle of encounter are clearly shown. And let the ship master to carefully avoid to operate the ship within the dangerous range. In this case, the ship master is totally responsible to avoid capsizing in following waves. Obcourse, the ship is assumed to be capable of survioing in head seas and beam seas condition.

b) To design a ship so that a ship does not capsize at any combination of wave height, wave lenght, ship speed and angle of encounter. In other wards, a ship is designed to be fail safe, and the shipbuilder is totally responsible for capsizing.

In practive, a compromize between a) and b) should be neccessary. The author's opinion is as follows;

i) First of all, crew should be well educated for the danger of capsizing in following waves, and the presence of the dangerous range of combination of sea condition, ship speed and direction should be well acknowledged.

ii) Then, precise information as to the dangerous range of combinations of sea condetion, ship speed and angle of encounter for each ship should be given to the ship master. Most effective way of escaping from this dangerous range, such as slow douwn the ship speed, should also be included in this information.

iii) The ship master is obliged to carefully operate the ship out of the dangerous range.

iv) However, it may happen that it will take some time before the ship master can judge that the ship is now in the dangerous range, and that it will take some time to escape from the dangerous range by slow down or changing course. It may also happen that the ship master is obliged to change course and unavoidably operate the ship in the dangerous renge for some time.

v) Therefore, a ship must be designed so that not to capsize if it is exposed in dengerous range of following wave condition for a while, but not long time. In other words, too extreme severe sea condition together with highest ship speed should not be taken as the basis of a stobility criterion in following waves.

vi) If an international stability criterion in following waves is established with above mentioned philosophy, it will also be used as a good tool for designing ships, and operating ships.

Three have been two proposals for international stability criteria in following seas made at the stability , Load lives and Fishing vessels safety subcommcttee of IMO. One is the proposal made by Federal Republic of Germany[5] and the other is the proposal made by German Democratic Republic and Polland[6]. The former is based on extenrive model experiments[7] and the letter is based on theoretical consideration of pure loss of stability.

Hamamoto[8] has applied the both criteria together with A167 and A562 to 16 Japanese vessels as shown in Table 1 and Table 2. As seem in Table 1, the both following wave criterea seem to be more stringent than bean seas criteria A167 and A562.

Ob course, it may be possible that a ship which complies with the beam seas criteria fails to comply with the following wave criteria. However, it would be preferable that, as a whole, the bean sea criteria and possible following wave criteria are the same level of stringency except those ships of which GM varies excessibly in longitudinal waves.

It is not the purpose of this paper to discuss relative merits of FRG and GDR - Polland proposals, but the author is really looking forward to seeing a final version of the IMO critesion for stability in following waves in reasonable form. In that criterion, the author hopes that too much obligation is not required to ship designers, and operational effort such as reducing ship speed is reflected in it so that ship owners can indicate ship masters to reduce speed at specified rough sea condition.

Summarizing the above, it can be concluded that;

1) Copsizing in beam wind and waves should be mainly avoided by design of ships. Operational effort would be subsidiary.

2) Capsizing in following waves should be mainly avoided by operational effort, provided crew are well educated for the danger of capsizing in following waves, and precise information as to dangerous range of sea condition, ship speed and angle of ecounter for each ship is giben to the ship master. Design of ships would be subsidiary as far as the ship is designed so as to comply with beam sea stability criteria, and to withstand in moderate following waves, at moderate ship speed.

Peference

[1] IMO Resolusion A167, 28 Nov.1968

[2] IMO Resolusion A168, 28 Nov.1968

[3] IMO Resolusion A562, 22 Nov.1985

[4] S. Kan et al "Capsizing of a ship in Quatering seas, (Part 1. Model Experiments on Mcechanism of Capsizing) Jounal of the Society of Naval Architects of Japan Vol.167, 1990

[5] "Proposal by the Federal Republic of Germany on Stability criteria." Annex 4, SLF 31/22, IMO, 1986.

[6] "Proposal by the German Democratic Republic and Polland for Guidance on a Method of Calculation of the Intact Stability of ships in Following Waves." Annex 5, SLF 31/22, IMO, 1986.

[7] "Report on Stability and Safety against Capsizing of Modern ship Design." Submitted by the Federal Republic of Germany. IMO SLF/34, 5 September 1984.

[8] "Sample calculations of the proposed criteria ap plied to Japanese vessels." IMO, ANNEX of SLF33/INF.24, 4 July 1988.

Table 1 Principal Particulas & Caluculation Conditions

SHIP No.			1	2	3	4	5	6	7	8
KIND OF SHIP			5000 M3 LPG TANKER	CONTAINER CARRIER	TANKER	CARGO & PASSENGER	PASSENGER BOAT	FERRY	CONTAINER CARRIER	CONTAINER CARRIER
NAVIGATION AREA			CREATER COASTING	CREATER COASTING	COASTING	CREATER COASTING	CREATER COASTING	CREATER COASTING	OCEAN GOING	COASTING
GROSS TONNAGE (t)			3,600.00	2,680.0	699.28	5,489.45	3,540.27	4,886.01	10,381.50	1,665.42
DEAD WEIGHT (t)			3,793	2,999	1,780	—	—	3,119	13,102	2,374
LENGTH(B.P.P.) (m)			92.00	92.00	65.00	115.00	100.00	128.00	145.00	79.00
BREADTH (m)			16.00	17.20	10.80	17.20	15.20	20.50	23.00	14.20
DEPTH (m)			7.00	7.50	5.00	6.80	8.20	11.90	13.00	5.80
DEPTH(TWEEN DK) (m)			—	—	—	—	—	7.10	—	—
DRAFT (m)			5.40	6.10	4.70	5.50	4.75	5.35	8.65	4.45
C_B			0.738	0.540	0.725	0.549	0.522	0.561	0.642	0.682
FULL LOAD	DEP.	Δ (t)	6,042.90	4,779.3	2,464.0	6,491.00	3,882.0	8,072.49	18,984.98	3,376.28
		G_0M	0.91	1.44	1.22	1.15	1.10	1.27	0.77	1.28
	ARR.	Δ (t)	•5,948.67	4,481.9	2,374.69	•6,354.00	•3,568.0	7,551.62	•17,382.21	3,154.36
		G_0M	0.99	1.07	1.20	0.80	1.08	0.99	0.41	0.99
BAL-LAST	DEP.	Δ (t)	4,819.32	3,341.2	1,992.19	6,017.00	3,809.0	6,723.28	12,277.36	2,346.66
		G_0M	2.34	1.71	1.68	1.63	1.63	2.20	2.69	3.34
	ARR.	Δ (t)	4,432.20	3,043.8	1,902.88	5,340.0	3,462.0	6,089.45	9,946.80	1,942.04
		G_0M	2.59	1.35	1.71	1.10	1.55	1.99	3.11	3.48
LIGHT WT (t)			2,250.00	2,330.5	684.26	4,414.00	2,964.0	4,953.63	5,890.96	1,176.23
GM			3.44	0.70	3.05	0.66	1.10	1.53	2.80	4.64

SHIP No.			9	10	11	12	13	14	15	16
KIND OF SHIP			TANKER	REEFER	CONTAINER CARRIER	TANKER	ORE CARRIER	TUG BOAT	MULTI PURPOSE	BULK CARRIER
NAVIGATION AREA			OCEAN GOING	OCEAN GOING	OCEAN GOING	OCEAN GOING	OCEAN GOING	OCEAN GOING	OCEAN GOING	OCEAN GOING
GROSS TONNAGE (t)			2,478.14	2,384.27	4,790.51	49,197.21	25,311.39	2,161.64	9,966.46	19,900
DEAD WEIGHT (t)			3,316	4,194	6,691	76,572	40,210	1,920	14,941	34,494
LENGTH(B.P.P.) (m)			92.00	97.00	114.00	232.00	183.00	79.00	136.02	170.00
BREADTH (m)			13.80	16.00	18.50	34.80	30.00	14.60	21.02	28.40
DEPTH (m)			7.10	9.55	8.50	20.80	15.00	7.20	12.20	15.00
DEPTH(TWEEN DX) (m)			—	7.05	—	—	—	—	9.05	—
DRAFT (m)			5.70	6.56	6.633	13.85	10.51	6.00	9.00	10.30
C_B			0.644	0.645	0.685	0.806	—	0.575	0.757	—
FULL LOAD	DEP.	Δ (t)	4,783.3	6,754.70	9,815.1	92,431.8	48,401.0	4,083	20,163	42,368.0
		G_0M	0.88	1.04	1.17	2.92	3.23	1.60	3.95	2.22
	ARR.	Δ (t)	4,617.7	6,114.99	9,020.00	91,074.5	•47,799.0	2,5[illegible]	19,511	41,928.0
		G_0M	0.83	0.35	0.92	2.91	3.20	[illegible].24	3.12	2.15
BAL-LAST	DEP.	Δ (t)	2,458.8	3,698.38	6,183.5	46,489.0	23,232.0	4,083	—	20,349.0
		G_0M	0.41	1.63	3.50	6.50	8.60	1.58	—	3.43
	ARR.	Δ (t)	2,209.1	3,058.67	5,889.1	45,131.6	22,620.0	2,802	—	19,719.0
		G_0M	0.51	1.50	3.50	6.68	8.98	1.16	—	3.33
LIGHT WT (t)			1,467.1	2,056.70	3,124.8	15,860.0	8,191	2,163	5,221	7,874.0
GM			1.36	—	3.99	—	—	1.13	—	—

• Loaded with ballast

Table 2 Results of Sample Calculations for 16 Vessels of Japan.

Ship No.	item	D − D Full Dep.	D − D Full Arri.	D − D Ballast Dep.	D − D Ballast Arri.	D − D All Cond.	(FRG ; D−D′) (GDR, Fn=0.15) Full Dep.	Full Arri.	Ballast Dep.	Ballast Dep.	All Cond.	A 167 All Cond.	A 562 All Cond.
1	FRG	×	○	○	○	×	×	○	○	○	×	○	○
	GDR	○	○	○	○	○							
2	FRG	○	○	○	○	○						○	○
	GDR	○	○	○	○	○							
3	FRG	○	○	○	○	○						○	○
	GDR	×	×	○	○	×	○	○	○	○	○		
4	FRG	×	×	○	×	×	○	○	○	×	×	○	○
	GDR	○	○	○	○	○							
5	FRG	×	×	○	○	×	○	○	○	○	○	○	○
	GDR	○	○			○							
6	FRG	○	×	○	○	×	○	○	○	○	○	○	○
	GDR	○	○	○	○	○							
7	FRG	○	○	○	○	○						○	○
	GDR	○	○	○	○	○							
8	FRG	×	×	○	○	×	×	×	○	○	×	○	○
	GDR	○	×	○	○	×	○	○	○	○	○		
9	FRG	○	○	×	×	×						○	○
	GDR	×	×	○	○	×							
10	FRG	○	○	○	○	○						○	○
	GDR	○	○	○	○	○							
11	FRG	×	×	○	○	×	×	×	○	○	×	(Full) ×	○
	GDR	×	×	○	○	×	$\times(\theta_3)$	$\times(\theta_3)$	○	○	×		
12	FRG	○	○	○	○	○						○	○
	GDR	○	○	○	○	○							
13	FRG	○	○	○	○	○						○	○
	GDR	○	○	○	○	○							
14	FRG	○	○	○		○						○	○
	GDR	○	○	○		○							
15	FRG	○	○			○						○	○
	GDR	○	○			○							
16	FRG	○	○	○	○	○						○	○
	GDR	○	○	○	○	○							

WHAT HAVE GUIDED INTERNATIONAL ACTIVITIES ON INTACT STABILITY PROBLEMS SO FAR? ARE CHANGES NEEDED? IN WHAT DIRECTION?

Ivar A. Manum

In considering safety against capsizing there will be contributions from both the design and the operation of vessel. We all know that 100% safety can never be obtained. There will always be a risk left, which have to be catered for by preparedness.

Through decenniae the Universities, the Research institutions, the Administrations, IMO and recently these International Stability Conferences have done their utmost to improve on the intact stability criteria for the design of vessels including interpretations, calculation methods and description of operating conditions in relation to such criteria.

This has resulted in adequate international minimum intact standards for design against capsizing for types of vessels.

However, this biuld-in safety in the vessel and its equipment must be maintained through-out the lifetime of the vessel. This will always to an extensive degree be dependant upon the level of competency and safety commitment amongst the vessels crew and commitment by owners to response to the needs stressed by the Master.

In this relation, as well as in the operation and the preparedness as mentioned, the human element is, in my opinion, a decisive factor.

It is acknowledged that investigations of ship casualties in general clearly establish that the human error is the source of 60 - 80% of all marine accidents.

I belive that an investigation of capsized vessels separately would show an even higher per centage.

I belive it is now time for a change in the research and the administrative activities on stability problems. In order to reach at a more balanced vies than up to now, we must concentrate much more of our effort on how to reduce the probability of human errors.

(In a paranthesis I would like to draw your attention to the UK public opinion and their political system which have changed attitude after the tragic casualty with "Herold of Free Enterprise".

Basic build-in particulars with influence on the stability as light-weight and centre of gravity will be regularly confirmed. Further the responsibility of the companies operating ships is put on the ship safety agenda.

Although the "Exxon Valdez" casuality has no connection with stability, the recently decided US-laws on pollution prevention stress the same change in attitude. Responsibility of companies and means to reduce the probability of human errors are new regulatory areas).

We must thoroughly analyse the casualties in order to find the primary source behind these so-called human errors.

Such primary sources could be:

- lack of procedures
- lack of information
- lack of assisting means
- lack of stability knowledge
- lack of easily understandable operational manuals
- lack of safety policy and motivation amongst owners and crew
- fatique-problems
- etc.

In my opinion we must find ways and means to fight the sources behind human errors. We must find ways and means, possibly through new regulations to ensure compliance with all the mandatory requirements on stability rather than to continue our effort on improving the criteria for design with its conditions, interpretations, etc.

Verification of Dangerous Situations in Quatering seas by Use of Model Tests Results

Y. TAKAISHI

1. INTRODUCTION

The most substantial cause of the danger in quartering seas is found in the case of the ship navigating with the speed as same as the group velocity of main part of irregular waves. In this case the so-called dangerous group wave phenomena of encounter waves occours so that almost the highest waves attack the ship repeatedly. This phenomena was found by the author 1 and the diagram has been proposed to identify the dangerous combination of encounter angle, ship speed and wave period.

In this paper, the verification of the diagram is tested by using the model test results of capsizing experiments in model basin at SRI. It will be concluded that the diagram will be useful provided some degree of spreading of dangerous situations considered.

2. EXPLANATION OF THE DIAGRAM

Fig. 1 shows the diagram of polar expression. The vertical axis (=0) means the direction of wave propagation and the abscissa means the direction of wave crest. The ship course is represented by the radial lines with the scale indicating ship speed in knot by concentric circles. The horizontal lines parallel to the abscissa indicate the wave period in second corresponding to the peak frequency of the wave spectrum. Any point on the diagram shows the dangerous combination of three quantities i.e the ship speed, encounter angle and the dominant wave period, at which the ship speed is equal to the group velocity of waves havinh main energy part in irregular waves. This relationship is represented by the following formula:

$$V_c \cos\chi = g T_p / 4\pi = C/2 = C_g \qquad (1)$$

where Vc: the critical ship speed, :the encounter angle, Tp: the wave period of the peak of the wave spectrum, C: the phase velocity of the wave, and Cg: the group velocity of wave corresponding to Tp. It should be noticed that this dangerous condition occurs at the half speed of the phase velocity of wave. Since the surf-riding or broaching-to phenomenon occurs in such cases that the ship approaches to the phase velocity of wave, the dangerous situation here mentioned should be recognized as the other phenomena than surf-riding.

An example showing such situation of the encounter waves is shown in Fig. 2. It is remarkable that the ship will encounter to extremely high waves successively. To avoid this danger the ship speed should be reduced significantly low.

3. INTERPRETATION OF MODEL BASIN TEST RESULTS BY THE DIAGRAM

In the diagram, Fig. 3, the capsized cases in the experiments are indicated for three ship models. The test conditions of ships are shown in Table 1.

The container ship G was tested by Kan et al. 2

The points in the diagram show the ship speed and wave period in Table 1. The encounter angle of the points are derived automatically by using the ship speed and wave period. However, these points don't necessarily indicate all capsized cases, i.e. capsizing occurred at a wider range of encounter angle around the point. This means that the dangerous situation is not represented only by a point but by a certain area around the point, because all quantities are not strictly determined by one value, but have some deviations or spreading, i.e. stochastic features.

4. CONCLUSIONS

The running in following or quartering seas with the speed that the ship travels with the wave group is one of the most dangerous situations in oblique waves, because the extremely high attacks repeatedly and the worst events such as heavy rolling, shipping water, stability reduction, etc. can occur. It seems that not only capsizing of model ships but also many accidents of actual vessel occurred under such a situation. The diagram proposed will be used effectively as a guideline of navigator to prevent the worst condition to waves.

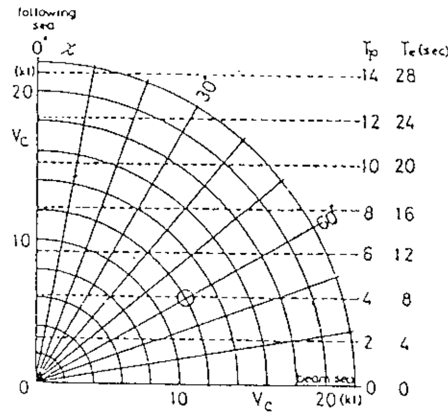


Fig.1 Diagram Indicating Dangerous Situations in Quartering Sea [1]

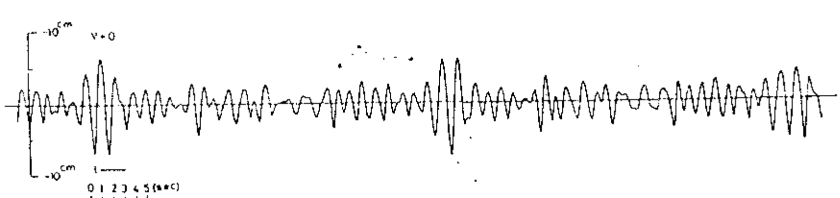
REFERENCES

1 Takaishi, Y; Consideration on the dangerous situations leading to capsize of ships in waves, Proc. *STAB'82, Tokyo, 1982*

2 Kan, M. et al.; Model tests on capsizing of a ship in quartering waves, STAB'90, Naples 1990

Table 1 Ship Conditions

No	Kind of ship	Lpp(m)	B(m)	D(m)	dm(m)	trim	T_R(sec)	GM (m)	V(kt)	Tp(sec)	$H_{1/3}$(m)
a	Drift Netter	14.4	3.5	1.38	1.20	1.08	5.07	0.34	6	4.7	1.7
b	Container:F	135	23	11.5	8.37	0		1.77	22.7	13.4	10
b'										11.2	9.6
c	Container:G	135	24.3	11.5	8.37	0		0.76~1.06	19~24	13.4	10



(a) Record of Irregular Wave Elevations Measured at the Fixed Point in Basin

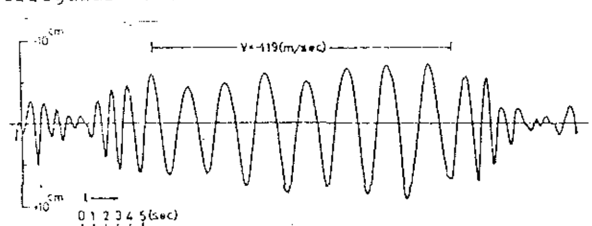


Fig. 2 (b) Record of Encounter Wave Elevations at Critical Velocity [1]

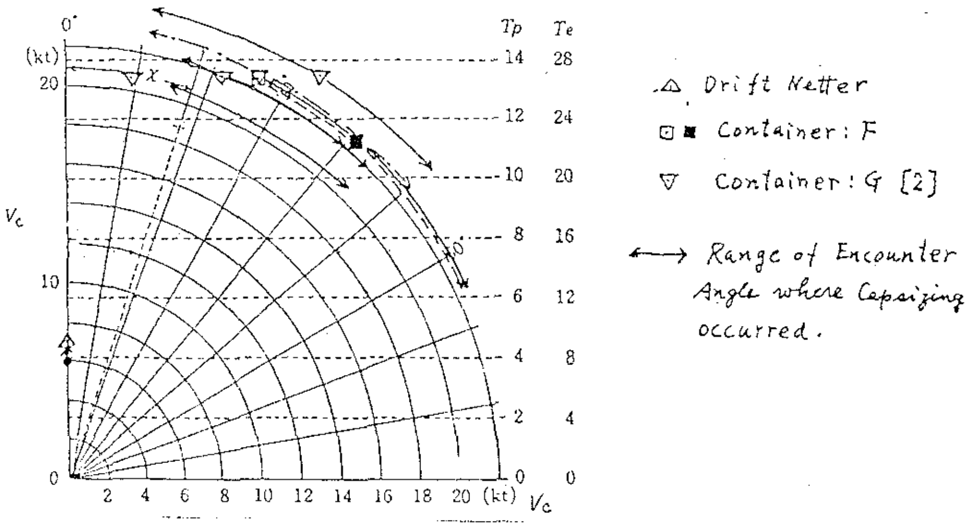


Fig. 3 Verification of Dangerous Conditions by means of Test Results Plotted on the Diagram

Consideration for Operational Stability

J.S. SPENCER

Only one year ago the International Maritime Organization adopted Resolution A.647 "IMO Guidelines on Management for the safe Operation of Ships and for Pollution Prevention", which addresses these important considerations. According to that resolution, care must be taken to ensure that stability information is correct, understandable, and simple enough to be usable by the ship operator: contingency planning must be done and a plane of action for emergency situations be written up in a concise format; and each company should publish operating instructions that are explicit and well written, taking into consideration the human element involvement in every task.

Operational safety must be considered each time before the ship leaves its berth. The Master must ensure that cargo is properly secured and hatches and doors are closed tight. He must know that all cargo and consumable weights have been accounted for, calculated drafts have been verified with the actual draft marks, and necessary adjustments made so that the ship is in compliance with the applicable intact and damage stability standards.
The Herald of Free Enterprise disaster demonstrated the importance of positive reporting of critical matters, no matter how routine, trivial, or obvious they may seem to be.

If operational safety is a cornerstone of good management during design, construction, equipping, maintaining and loading of the ship, then while it is underway in normal conditions adequate stability should be easily maintained. The Master normally would have only to ensure that procedures for properly using consumables and fuel are followed and thet watertight doors remain closed when they are not actually in use. In times of heavy weather or when the ship is involved in an accident the Master's job is more difficult. It is essential that he is competent and prepared to make the coorect decisions and lead his crew in emergency situations. However, the Master may fail in his job if he is not fully conversant with the tools and procedures available to him, or if he has not been given the authority and flexibility to perform his duties properly, safely and thoroughly.

The stability criteria used in the design, construction and operation of a ship are a foundation to operational safety. We who have taken up the challenge ti improve safety by improving our knowledge and understanding of intact and damage stability must continue to work together. It requires keeping an open mind to each other's ideas, with an understanding that stability is varied and complex, often having incompletely definied variables and novel formulations. Let us not lose sight of our common goal to improve the safety of ships and that each of us should strive to do our part.

OPTIONAL STABILITY AND ADVOIDANCE OF ACCIDENTS
SOME THOUGHTS AND QUESTIONS

by

H. HORMANN

To prevent stability related accidents en route is a task which almost entirely has to be performed before the ship leaves port. This is different to many others areas of accident prevention, where continuous precautions or alert are integral parts of safeguarding the ship's adequate level of safety during the voyage.

Once the ship left port there is very little possibility to influence the stability properties - mainly the option to take in additional ballast; this, however, requires spare freeboard. Furthermore, if at all possible, the overall effect of additional ballast is not necessarily positive. Over and above this there are no more means, what is left then is only to react nautically towards reducing the impact of the external forces, i.e. altering the course.

Any sound stability related decision have as a prerequisite the knowledge of key figures and their possible margins. This puts extreme importance on the master (or the officier responsible) to receive timely the information on the weight of cargo.

It is a general problem that the decision steps to secure sufficient stability for the voyage ahead have to be undertaken during a period which for those in charge is densely occupied with demands for various other important decisions. In consequence two key questions are:

- how can the officier in charge be given sufficient time for stability forecast and stability check before the ship puts to sea?

- how can the quality of information available before loading is commenced be improved?

An obvious improvement of the entire situation is the provision of means on board to quickly and reliably perform inclining tests toward or after the end of the loading process. But what happens, if the result of such test is negative? What can be done to establish a "net" which would prevent a master to be almost automatically dismissed, if he decides that the load, he just took on board, was too much and part should be put back on shore?

In all considerations of how to improve operational safety the securing of cargo need be addressed at the same time as stability itself. To provide sufficient stability has no positive effect, if the cargo is not secured against shiftin or falling overboard. There are harly any "clean" stability accidents, i.e. heavy lists or capsizings not caused or at least initially contributed to by shift of cargo. To improve in this area for one important ship type, the new concept of open - hatch - container ships appears to have a great potential.

Finally, are there are other means to achieve that the ships leave port with a reasonable margin in stability compared with the minimum permitted for the individual ship. Can we expect incentives in this respect from underwriters?

CONCLUSIONS OF THE PANEL DISCUSSION N.1

H. HORMANN

The intent of the panel discussion was at least threefold:

- induce thinking regarding the general context whenever new theoretical knowledge is achieved;

- try to illustrate repercussions of theoretical/numerical achievements in practice

- indicate possible ways to progress towards the aim: use of the knowledge on stability for improving safety at sea.

The introductory statements can be categorized as follows; three dealt with proctical problems, mainly, how is stability affected by operations; and one was a plea for validating the numerical results of calculations by calibrated thank tests.

The core of the discussions, which ensued upon the introductory statements may be summarized in a condensed form by the following paragraphs:

- As in other disciplines one has to look at safety in its entirity, i.e. the ship as a whole. This is applicable also for the problem area stability, we must not forget the operational and managerial aspects as well as the human factor, they all have a decisive influence on the question: is a ship safe stabilitywise?

- We have a sharing of responsibilities for stability between the designer and the master; communication between both is a vital to assess, how this responsibility is divided between them.

- Scientists should be more application conscious; frequently they terminate their efforts at a too early stage so that the final goal, i.e. benefit in practice is not reached. To this end the scientist - perhaps with the assistance of practice oriented experts - should start to validate, what he has achieved, at an early stage and continue to do so at intermediate steps.

- Everyone knows that fool-proof technical objects are no viable options. But it should be realized that the safety always gains with an increase in the built-in safety. This is very evident for intact stability.

- We have to put more emphasis on analysing the real sources of accidents and in particular investigate the patterns which bring about human errors. It is not enough to explain how an acciden-t/incident developed; the question as to the why has to be pursued more intensively.

- There appears to be still a potential for improving the way in which the information on stability is presented to the man on board. Better strategies are required, how to pass on the messages and how to safeguard the application of the material by him in the time available.

Finally I would suggest that in future Conference like this, more regard should be paid to, more emphasis should be laid on looking at adjacent or conplementary areas - better understanding of the human factor is only one example.

Panel Discussion II: **Inclusion of Theoretical Achievements in The Field of Stability in the Ship Design Process**

Panelists:

Changi Kuo (Chairman)
University of Strathclyde
United Kingdom

Sigismund Kastner
Bremen Polytechnc
Germany

Lech Kobylinski
Technical University Gdansk
Poland

Resat Ozkan
Deniz Ticaret Odasi
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Randolph Paulling
University of California
Berkeley - U.S.A.

Written Contribution by:

S. Kastner; R. Ozkan; R. Paulling; J.T. Stasiak

Inclusion of Theoretical Achievements on Ship Stability in Ship Design Process and in Ship Operation

Contribution to panel discussion by Sigismund Kastner [1]

Basic Definition of Stability

Generally, there is only one basic definition on the stability of any floating structure. We may see it as the ability of the vessel to recover from any heeled condition to the upright condition again.

Fig. 1 illustrates the basic pattern of ship stability (taken from my paper at the STAB'86 Conference in Gdansk). Any variation in the positions of either G (centre of gravity) or B (centre of buoyancy) due to external or internal forces acting on the vessel results in a variation of the uprighting ability, i.e. on ship stability. In Fig. 1 we see a condition with G' from shifting of cargo, and with B' varying in following waves, depending on crest (c) or trough (t) position. Stability is then represented by the corresponding righting lever G'Z'.

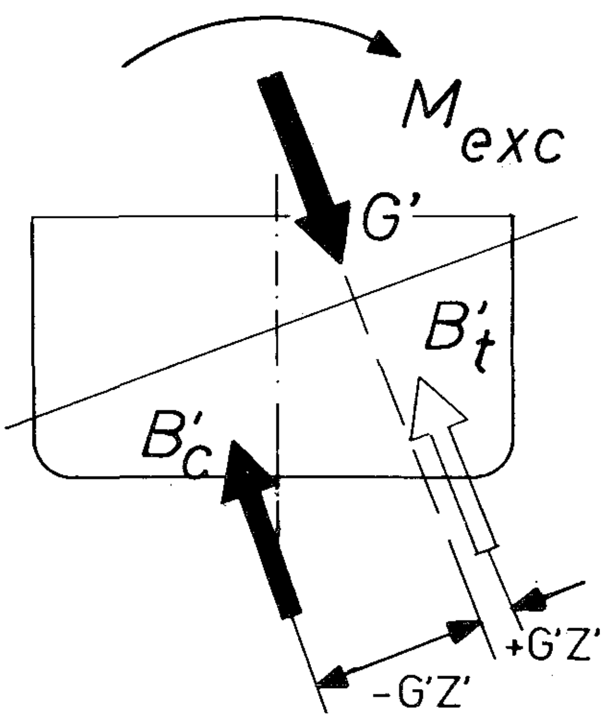


Fig. 1: Basic Moment Pattern of Ship Stability

1) Professor, Dr.-Ing., Bremen Polytechnic Neustadtswall 30, 2800 Bremen 1, F.R.G.

It is now a matter both of practical ship design and of ship operation, to allow the vessel to sail safely at the environmental conditions met on the Oceans. In order to ensure good stability and safety from capsizing, the ship must be designed accordingly with respect to main dimensions, hull form, and mass distribution. In ship operation, we must be able judge variations of G and B from loading, and from sailing in the Ocean environment. Fig. 2 illustrates the four-fold interaction of cargo and vessel, environment, and ship operation (taken from STAB'86).

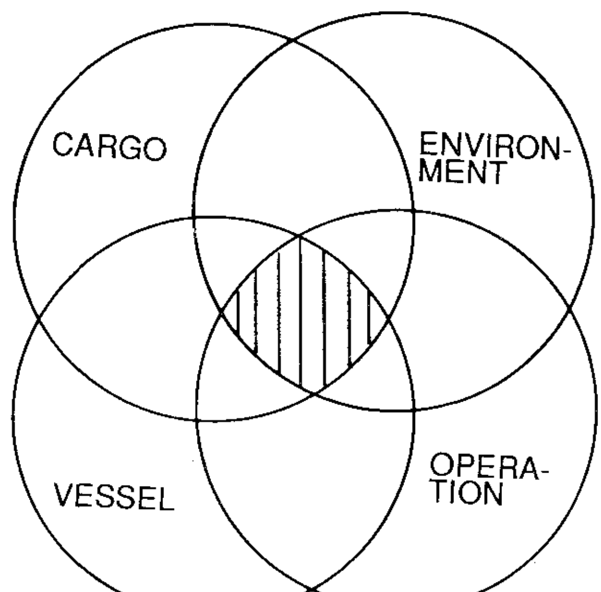


Fig. 2: Four-Fold Interaction of Cargo, Vessel, Environment, and Ship Operation

What is "Practical Stability"?

To reach sufficient stability at any ship condition, but without overdoing it, requires detailed knowledge and proper judgement. This is the practical field where all the problems with ship stability start.

We must certainly be sure to give the vessel at least a minimum stability. Will this always be enough for all circumstances? Certainly not. On the other hand, too large stability results in too large accelerations acting on vessel and cargo, and will in most cases be uneconomical. Problems arise from calculating difficult hydrodynamic patterns of the ship hull, but also from the need to know or to choose the right figures in loading and for environmental impacts for any particular ship condition. Therefore it is useful to speak in this context of "Practical Ship Stability". It can be split into three branches, see Fig.3:

- regulatory stability
- design stability
- operational stability.

Results of ship hydrodynamics can of course give guidance to all 3 branches of practical stability.

The Stability Range of Judgement

Regulatory stability sets minimum standards of required stability, and gives the best procedures to control the stability limits.

Although it is certainly necessary to devise sound international stability regulations (e.g. for ships of more than 100 m in length still overdue), it is important to look on design stability and on operational stability, too.

Unfortunately, ship designers, and even ship masters, have been looking for stability requirements from regulatory stability as guidance for ship design and operation. In order to meet all possible conditions at sea, certainly the minimum stability is not sufficient.

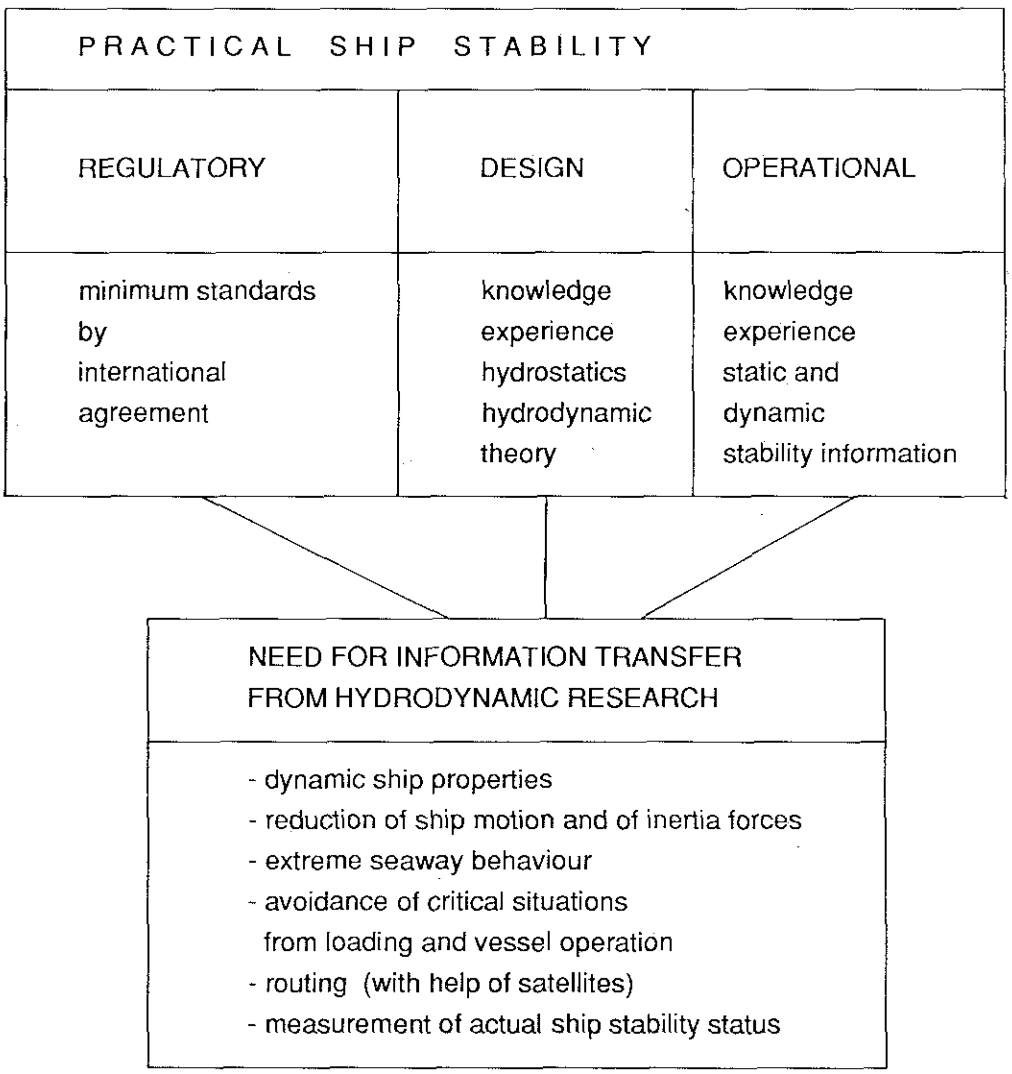


Fig. 3: Scheme of Practical Ship Stability

We may call the range of stability needed, above the minimum regulatory stability, to overcome safely any condition at sea, the "stability range of judgement" (see STAB'86).

The minimum stability standards do not state specifically, which excitation can be overcome. There is a strong need to transfer information from hydrodynamic research on the motion behaviour of vessels related with ship stability into the practical field of ship design and operation.

The more the designer is applying hydrodynamics, the better his ship may be, and the more information he can give to the ship master for safe ship operation.

At the moment, the ship master is responsible but left alone with difficult decisions on ship stability, if he meets extremely rare but dangerous conditions at sea. He needs information on the ship behaviour to be expected.

Transfer of Hydrodynamic Theory

How to use the "stability range of judgement" must be based on better information from ship hydrodynamics directly, and on particular dynamic ship properties during the design process.

Model testing procedures on stability and on roll behaviour in a seaway, together with corresponding computer simulation, can be extended to practical ship design. The application of such results in ship operation requires to transform hydrodynamics into simple procedures and guidelines.

Better information should be given on

- dynamic ship properties
- reduction of ship motion
- extreme cargo accelerations and effect on securing of cargo
- extreme seaway behaviour including capsizing.

Special measures to look at are:

- roll resonance at different headings and measures on resonance avoidance
- avoid rare extreme events at sea by routing
- control and measure GM (KG) aboard to know the actual stability status better
- measure ship reaction aboard vessel and indicate at bridge
- do not trap the master into relying on minimum stability curves solely, while at severe condition at sea
- give "code of safe practice" aboard vessel
- show ship master scenarios of extreme ship behaviour with his particular class of ship
- show the existing stability limits of any ship, even at high technology level, in advance, and train protective measures.

Written Contribution

by

Reset OZKAN

1. IMPORTANCE

Stability of a ship in a seaway is perhaps the most sophistacated problem of Naval Hydromechanics as it stands however, being handled in an extremely oversimplified fashion in today's practise. In brief, it can be named as the ability of staying away fro the phenomenon of capsizing that is to say having the quality of avoidance a catastrophic failure.

2. DEFINITIONS

a) nonlinearity

A ship in a seaway performs a general motion which is the resultant of a number of pure and coupled motions taking place at the interface of two different media. Among the others, rollimg mode of the motion appears to be the most dominant one, at the large angels of which the phenomenon of capsizing starts and develops therefrom. It is therefore the stability of a ship is undertood to be stability of rolling motion and that involved a series of mechanical and hydrodynamical interaction problems. Speaking of stability one should clearly and definitely admit that the motion under investigation is strictly a nonlinear one and therefore, no criteria can be evaluated in a global nature. It is a local concept and this is determined by the nonlinear features of the motion. In other words, one may say that the nonlinearity of the problem should be maintained in carrying out any stability analysis and it is therefore this requirement that puts the qualitative methods into the picture. It requires no further evidence that the methods of linearisation occupy a vital place in many applications of Naval Architecture and are very useful means in many problems however, it is of not much use in stability analysis of a large amplitude rolling motion of a ship as it can only provide some information to a finite number of set of initial conditions that are defined at the outset of the solution process. In addition to that they lack to provide any criteria which may be applied in a finite region as this region could only be defined via the use of nonlinear features of the model.

b) Degree of Freedom

Under the influence of mostly confused and interactive effects of the enviroment and itself a ship, in general and in real terms, makes n-degrees of freedom set of motion. Rolling motion is, obviously, the one whose stability should be investigated however the coupling effects of the others like pitching, heaving and yawing, as it happens to be in quartering and following seas, should be taken into account since they may become very influential on the eventual rolling characteristics.

c) Hydrodynamical Interaction

This problem, on its own, is a mayor one as it is nothing else than the evaluation of fluid active and reactive motions which determine corresponding forces exerted upon a ship by the environment. Despite this importance, one may say that the numerical data presently, used in place of these parameters are being employed with some degree of accuracy in today's stability analysis. However, this is a contradiction as this data is evaluated from the solution of a linear boundary value problem. For fluid active motion i.e., wave excitations, diffraction problem appears to be eliminated in applications and the use of Froude-Krylov theory can be adopted with some confidence.

d) Nature of Frequency

It is assumed that the incident environment develops with a well-defined frequency i.e., constant frequency for a certain sea state which permits the employment of ordinary differential equations in a frequency domain in modelling of motion. However this is not possible if one considers frequency to be a time-varying one as this would violate the uniqueness of the equation of motion therefore the use integro- differential equations in a time domain analysis is necessary.

3) METHODS OF ANALYSIS

a) Qualitative Approaches

In stability analysis of a ship, it appears to be that the nonlinearity of the equation representing the motion should be maintained. The use of qualitative approaches is a direct necessity imposed by this requirement and among these methods, the so-called Lyapunov's direct method together with its extentions and refinements has a great value. For a single degree motion a scalar Lyapunov function may suffice to investigate the stability conditions whereas for coupled motions VECTOR LYAPUNOV FUNCTIONS should be used.

b) Experimental Approaches

Experimental analysis is a very important way in solving many problems and it has the capacity of helping the solution of a stability analysis in great detail. In one hand, observation gained from experiments in forming the structure of the model of equation of motion as well as deciding on the degree of importance of various complex phenomena. On the other hand however, all experimental work claiming the evaluation of a stability problem should be planned and worked out in consequence of a thorough analytical work conducting of which experiments under what conditions ahould be decided before hand as it is an expensive and time-consuming approach. Otherwise, experimental results would be laking of giving any decisive generalities for the problem which is what is expected from a stability analysis.

4) CONCLUSION

Stability of a ship is a very difficult problem of naval architecture not only for its own definition but also it studies an extremely sophistacated mechanism of multi-degree motions of a ship.

It is, therefore, one should not expect to end up with results by using some easy wayouts. It is the belief of this author that we should extremely be prepared to have a totally impartial understanding of this problem, dedication and patience.

Inclusion of Theoretical Achievements in the Field of Stability in the Ship Design Process

J.R. Paulling

In an earlier presentation before this conference, it was observed that stability is that quality of a ship that has the *purpose of minimizing* the probability of a capsize accident. The stability referred to here, if it is to fulfil this objective, *must include ingredients based on* the properties of the ship, the environment in which the ship operates and the motivation and state of mind of the people who operate the ship.

The properties of the ship include the geometry of the hull, the internal subdivision and the various conditions of loading that the ship will encounter. The stability is then expressed quantitatively by such measures as the metacentric height, the curve of righting arms and the dynamic stability or area under the righting arm curve. These quantities may be computed for all of the expected conditions of loading of the ship.
This aspect of stability has received the greatest attention from the designer and the regulator because it is simple to quantify *and amenable to straightforward* methods of computation.

The properties of the environment most often considered are the wind, waves and in special cases, icing. The waves may have two effects, one a geometric effect on the righting arms and the other effect felt in the dynamic motions of the ship. The unfavorable effect of waves on the righting arm has been known for some time but, for reason of expediency, we have persisted in computing the righting arms under an assumption of still water, including the effect of waves in a margin of stability. The dynamic rolling motion induced by waves, on the other hand, has been included, in a simplistic manner, in some stbility criteria for some time.

Much more difficult to account for are the actions of the ship's personnel. The effectiveness of inherent ship stability in preventing accidents depends to a very great extent on the ability and willingness of the operating personnel to make use of *information supplied by the de-* signer and regulator, and to operate the ship in accordance with that information. Great variations *in training, experience and mo-* tivation may be found here rancing from the owner skipper of a small fishing vessel to the master of a large seagoing passenger liner or container ship.
The importance of information transfer to these persons and their training in its effective use cannot be overemphasized.

Recent research in ship stability may conveniently be divided into five major areas.

(1) Hydrodynamic theory, focused on such topics as large amplitude ideal fluid dynamics, viscous effects (e.g. roll damping) sloshing of free water on deck and large amplitude (breaking) waves.
(2) *Classic and moder stability* theory chaos theory.
(3) Numerical, computer-based simulation of large amplitude *ship motions and capsize.*
(4) Experimental studies, including captive and free running model experiments.
(5) The accumulation of statistical information on such widely diverse subjects as causes of accidents and wind, weather and sea conditions.

Results developed in all of these areas have greatly enhanced our undertanding of the phenomena of capsize. The results of such research often seem to find their first application in <u>post hoc</u> explanation of casualties, and they are not always quickly incorporated into either design or regulatory practice. In order to overcome this limitation and delay, it is important for those conducting the research to assume a share of the responsibility for demonstrating the validity of the assumptions underlying their work and for showing the way in which their results may be applied to real world problems.

In carrying out the ship design, a check on the stability is often made only late in the design process after the designer has finished adjusting the principal dimensions and arrangements to satisfy other design requirements. Unadequate stabililty at this point requires additional cycles of the design spiral or, in an extreme case, added ballast or restrictions on the ship's loading distribution in order to achieve the required stability.

Modern computer-aided design theory offers the possibility of fully incorporating stability considerations in the ship design process from the beginning. In the typical design spiral, as used today, stability is checked at some point during each design cycle but this check consists only of a comparison of the stability as it exists at that point against the relevant criterion. Design is conceived as a "sequential" process and the sequence is repeated several times until convergence to a final is attained. The designer seldom considerd the stability as an important design target in the same sense as capacity or speed, but merely a side condition to be satisfied.

A fully-integrated design process, on the other hand, is a concurrent process in which seakeeping, loading, strength, stability and all other requirements of the design such as payload and speed can be treated simultaneously. These different considerations can be based upon theory, computer programs, data bases and experience, with virtually unlimited possibilities. Thus, a rational analysis of stability in all operational conditions of a loading could, in principle, be included in the same way that computations of seakeeping structural loading and response, powering and machinery characteristics and cargo variations are included.
The requirements of such a system are for suitable theories and software or databases and sufficient computing power.

Written Contribution

by

Janusz T. STASIAK

The stability is one of the most important aspects of seagoing ship safety which decides about essence and usability of these means of transport. At the same time this is a multi-aspect and "badly" organized-diffusion-problem of a navigation practice for which it is impossible to determine entirely true and simultaneously simple functional relationship occuring in physics as laws.

Hence, a search for so-called "rational" stability criteria i.e. the criteria based on the full sophisticated models of physics of a ship capsizing process creates only illusion to obtain proper solution having a utilitarian importance.

The fact of the matter is that, these attemps are not able to take into consideration a whole functional structure of ship and shipping technology which in fact essentially afects the ship stability. Particularly, they are not able to treat of a seagoing ship as a business concern, as a techno-economic object. They ignore a subject role of a master as a ship operator and very important phenomena such as flooding and cargo stability e.g.

Thus, **there is no option about the system approach to solution of ship stbility problem.**

This generally means that the stability must be treated as a system whose structure is made-up both respective questions and the coupling between these questions and an environment of the system.

One can easy to notice that **the system approach to the ship stability solutionis nothing else than a proper way of ship designing in general.** That is for an ensuring of safety is a fundamental task of designing of every engineering object. This is such a task especially than, when a safety is understood as an accepted compromise of permissible risk and usuability of created objects.

Generally speaking, the system approach to the solution of safety problem and designing are inseparable notions whose differentiate engineering problems from the ones of physical sciences.

Taking above into account, for the ship stabiliy solution to be effective at the ship designing it is necessary to work out:

- a body shape havinh as good seakeeping characteristics as possible for all live ship service conditions instead a body shape having only the best calm-water capabilities;

- an appropriate for a given ship a safety manual containing first of all, solutions of following questions: admissible vertical positions of ship mass centre, cargo stability problems and relationships between ship's sability measure and ship's exploitation parameters; between roll angles and ship's velocity especially.

In this way the system approach to the stability ensuring consists on an equivalent treatment of the hardware ship solution which suits to bulit-in ship stability, and of a software solution which determines a ship's adaptability, i.e. a ship's operation stability. Only such designed ship can be both safe and profitable.

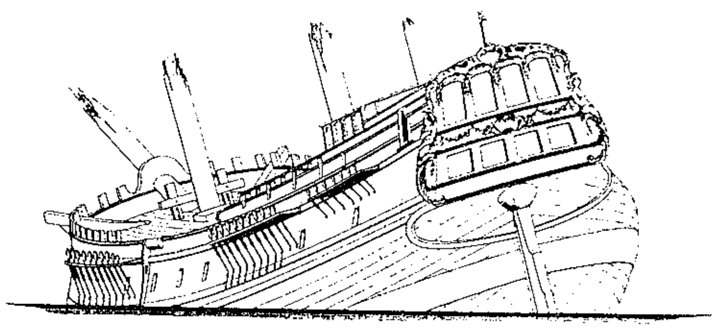
There are another two questions relevant to the matter in hand. Firstly, it is evident that a question of the improvement on ship's stability solutions resolves itself into **trasformation within the naval architecture.**

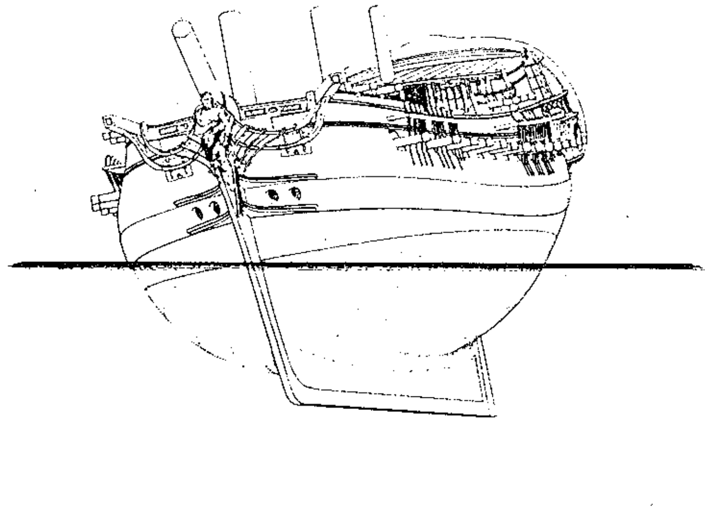
Strictly speaking, as long as a conventional piecemeal approach will be an obligatory method of the naval architecture the ship safety problem remains open. Improvement of the naval architecture first of all requires an overall approach to solution of hydromechanical ship problems and moreover a multi-disciplinary study as it is necessary both to have a sound understanding of functioning of ship sjstem and to have good knowledge of a safety theory.

Secondly, there is a problem of **changes of administrative and legislative regulations of navigation and shipbuilding.**
Seems to be impossible to solve properly substantial problems of ship, and the problem of ship stability particularly, by adapting them to the arbitrarily determined structures of a formal system. Certainly, the institutionalization of the ship designing and operation is a necessity. However, this necessity can easily change into an antineed, and becomes a source of artifical tensions and difficulties.

The point is that every formal system, and administrative one in particular, is inclined to generate its own needs which, in general, do not agree with the real purposes of a subject of interests. Unfortunately, the administrative and legal structures of marine are not free from this phenomenon. Therefore attemps should be made to elimination of these bureaucratic mechanism which are responsible for the essential and substantial questions of ship working to be "pushed aside" in competition with particular interests of respective formal institutions.

Recapitulating above, one should notice that an improvement of ship's stability solutions can be more effective achieved, first of all through a pragmatization of ship designing procedures and a rationalization of working of marine administration.
In other words, for the present, a task more important than a search about rigorous models of hydrodynamics of ship capsizing should be better utilization of existing seakeeping knowledge for to put a formal methods of ship designing and operation. Otherwise, the maritime world has to wait ages for do called "rational" stability criteria.





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INDEX

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