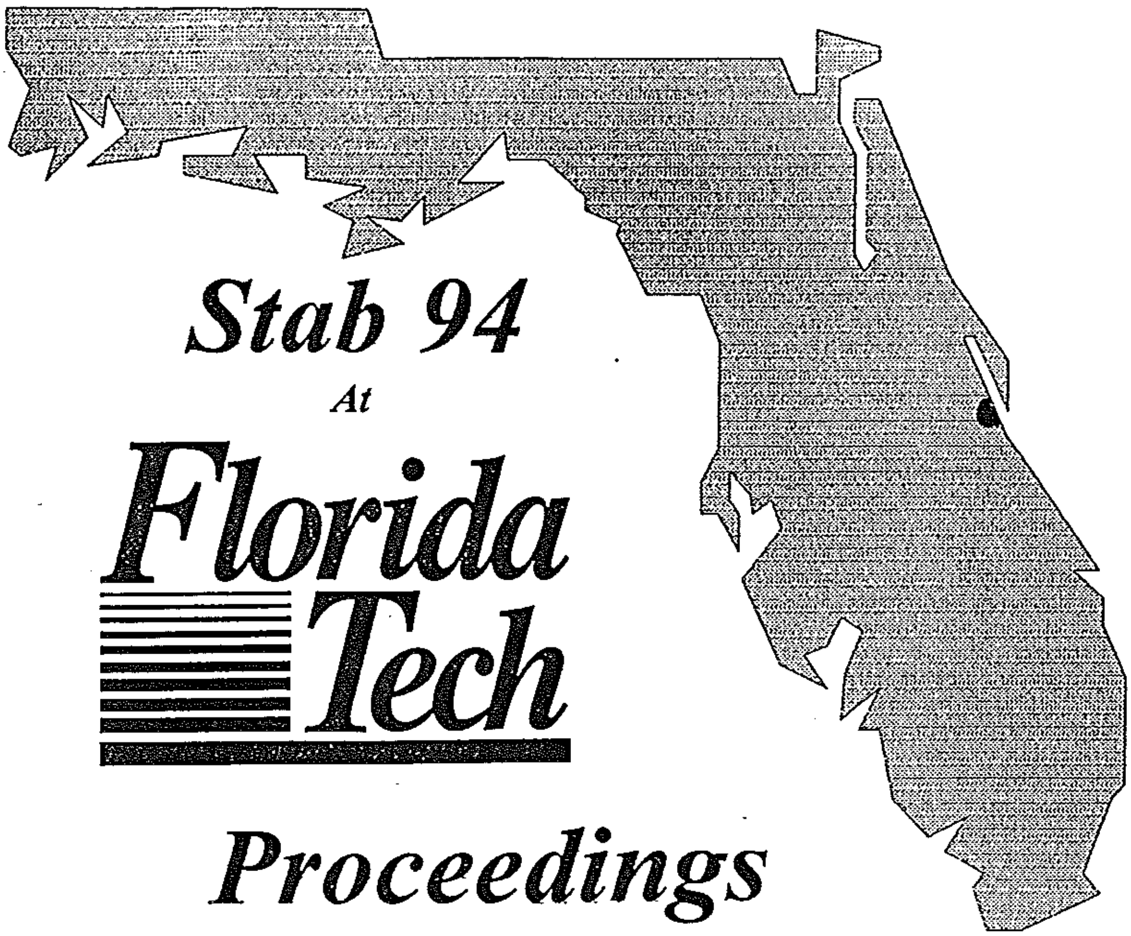
FIFTH INTERNATIONAL CONFERENCE ON STABILITY OF SHIPS AND OCEAN VEHICLES

NOVEMBER 7-11, 1994



Stab 94

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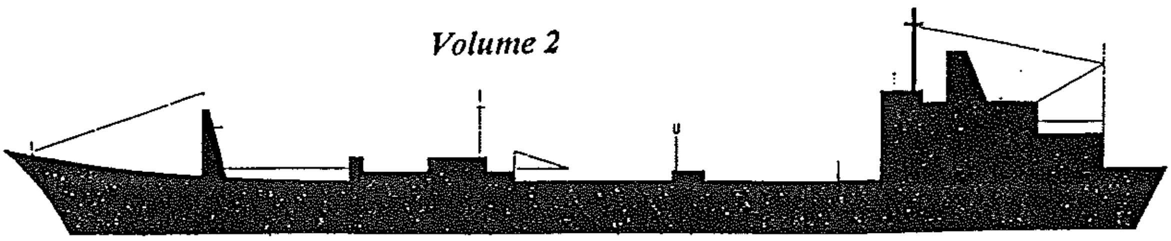
Volume 2



150 W. UNIVERSITY BLVD.
MELBOURNE, FL 32901-6988

PURPOSE OF STAB 94

STAB 94 had been offered to promote a full exchange of ideas and methodologies regarding STABILITY OF SHIPS AND OCEAN VEHICLES and to provide an opportunity to professional naval architects, capsize prevention researcher, regulatory agencies, inspection and certifying authorities, ship owners, consultants and ship operators to present, discuss and listen to improvements in capsize prevention for all types and sizes of ships.

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STAB 94

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University of Strathclyde

CAPSIZE THRESHOLD
(intact and damaged;
all ship sizes and types)

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(continued)

and

(concluded)

RESULTS OF THE 1993 U.S. COAST GUARD STABILITY SYMPOSIUM ON REGULATORY DEVELOPMENT

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U.S. Coast Guard Research and Development Center

Patricia L. Carrigan
U.S. Coast Guard Headquarters (G-MTH)

June 1994

Final Version

ABSTRACT

This paper highlights the 1993 U.S. Coast Guard Symposium and the information gained from the participants on the status of worldwide stability research. The paper examines in detail the work accomplished by the Symposium Workshop participants and the research objectives that were arrived at and the need for the establishment of an international stability committee to coordinate stability research.

INTRODUCTION

Statistically, stability problems result in more serious ship casualties worldwide than structural problems. Most current stability regulations are based on traditional hull forms in still water. Today, factors such as novel ship designs, ageing fleets, and reduced safety margins create doubt as to whether current static standards still provide an adequate safety margin. Research in the United States and internationally has thus far been not well coordinated resulting in data that are generally not correlated, and for the most part not usable to reevaluate current international stability criteria. Stability research is also extremely expensive and often requires extensive model tests or major software development and computer simulation. For these reasons, the U.S. Coast Guard recognized that it must take an active role in developing a plan for coordinated stability research.

Discussions of the need to develop some type of "Stability Forum" or "International Committee on Stability (ICOS)" go back to at least the mid-1980's within the U.S. Coast Guard. Specific discussions of the need for such a forum internationally, reached a high at the Fourth International Conference on Stability of Ships and Ocean Vehicles (STAB'90) in Naples, Italy during 1990, where there were discussed a number of proposals to initiate the groundwork for such an International Committee.

The success of the quadrennial STAB Conferences stand as proof of the strong worldwide interest in stability research. However, the STAB Conference's program are directed by the papers submitted rather than product-directed where the purpose of the meeting is to solve a particular stability problem. The usefulness of the STAB Conferences could be greatly enhanced if they are used to evaluate and recommend research in a cooperative manner.

Because of the numerous points-of-view aired at STAB'90 and the fear that some of the proposals might actually stifle rather than encourage cooperative stability research, further action internationally was stalled. However, the U.S. Coast Guard considered it very necessary that some interim progress be made before the next STAB meeting.

Based upon discussions during STAB'90, the decision was made by the Coast Guard to assemble a team of internationally recognized stability experts in a symposium/workshop forum to develop a Coast Guard stability research plan. By soliciting international expertise, it was hoped that whatever plan was developed by the Coast Guard would be beneficial to the rest of the international community. Such efforts in the past were very successful in the structures area, which provided a foundation for research that has provided the necessary tools for the Coast Guard to use in setting minimum standards. Coordinated structures research has clearly minimized the duplication of research efforts, thus saving the U.S. many dollars. Similar results are expected if the same level of coordination is brought to stability research.

In August 1991, a Technical Steering Committee of several highly recognized stability experts was convened at Coast Guard Headquarters in Washington, DC, to develop the detailed planning necessary for the Symposium. The Symposium was planned for the interim period between STAB'90 and STAB'94 in order that the results of the Coast Guard Symposium could be available for STAB'94. The Coast Guard Academy in New London, Connecticut, was chosen as the Symposium location because of its proximity to the Coast Guard Research and Development Center in Groton, Connecticut, which executed the Symposium, as well as the potential for professional interaction between Academy staff and stability researchers.

U.S. Coast Guard Stability Symposium 1993 Initial Technical Steering Committee	
CAPT Ted Thompson	U.S. Coast Guard HQ, Wash, DC, Chair
Dr. Chengi Kuo	University of Strathclyde, Glasgow, Scotland
Mr. Ivar Manum	Norwegian Maritime Directorate, Oslo, Norway
CAPT John Maxham	U.S. Coast Guard Academy, Groton, CT
Prof. Seizo Motora	Foundation for Shipbuilding Advancement, Tokyo, Japan
Prof. John Paulling	University of California, Berkeley, California
Dr. John Spencer	ABS Americas, Houston, Texas

Briefly, the goal of the Coast Guard Vessel Stability Symposium/Workshop was to identify specific research topics which were most likely to benefit development of vessel stability and safety regulations. The plan was to have three days of technical papers presentation followed by a one-day workshop. The theme of the Symposium was the impact each author's research had on regulatory development. During the one-day workshop, material drawn from the papers presented in the three-day Symposium would be used to provoke discussions about how useful that research could be in improving vessel stability.

SYMPOSIUM ORGANIZATION

The Symposium was organized into several sessions which dealt with different aspects of the stability problem. Session A, entitled "Stability and Regulatory Needs," was chaired by CAPT "Ted" Thompson, the Chief of the Coast Guard's Marine, Technical, and Hazardous Materials Division, which is responsible for administering stability research. The focus of the first session was to identify to researchers just how their research can be used in development of national and international stability standards. Several papers dealing with how regulations develop were presented by authors from three countries.

Session B, "Engineering," and Session E, "Engineering II," each addressed the role specific engineering or scientific advances had in improving stability research. Session B was chaired by CAPT John Maxham who is the head of the Coast Guard Academy's Engineering Department. Session E was chaired by Mr. Otto Jons, Advanced Marine Enterprises. Authors within these sessions presented papers on a wide variety of topics including: results of research in areas such as risk analysis, numerical prediction of ship capsizing and the effect of bulwark submergence.

Session C, "Management," looked at the big picture of how management affected stability and stability regulatory development. Session C was chaired by Dr. Chengi Kuo from the University of Strathclyde.

Session D, "Operations and Human Factors Aspects," chaired by Mr. Peter Fisher from Matson Navigation Company, dealt with stability and its impact upon safety from an operator's point of view. The roles of qualifications, education, training, and accident analysis were explored in the context of how they might contribute to stability regulation.

Even though the attendees had to brave a major east coast snow storm and many didn't make it, there were still over 100 persons from 10 different countries in attendance. They represented academic and research organizations, government and class societies, design and shipbuilding firms, and ship owners and operators. Varying perspectives on the need for specific research that would lead to reasonable regulatory requirements were well–represented among the attendees. These diverse inputs led to insightful and thoughtful discussion of the practical issues when trying to bridge the gap between research and regulation.

SYMPOSIUM PRESENTATIONS

A total of twenty–four papers was published in the Symposium/Workshop Proceedings, and represents the spectrum of stability research from computational fluid dynamics to human factors analysis. The variety of paper topics clearly demonstrates that the field of stability research is broad, and a complete presentation of all aspects of stability research is impossible in a three-day Symposium setting. Perhaps the most prevalent theme was the connection between the terms "ship safety" and "ship stability." Reference (1) identified "safety" as "safety is a perceived quality that determines to what extent the management, engineering and operation of a system is free of danger to life, property and the environment." In this context, safety is inextricably involved with management, engineering, and operations. The point is that a ship may be well engineered, but may be unsafe due to a poorly trained crew or mismanagement.

The term "stability" has a much narrower definition. In the classic engineering sense, "ship stability" is considered in a systems context. "A system is regarded as 'stable' when, disturbed from its normal position by excitational forces, it returns to that position after a reasonably short interval of time without sustaining damage or having its intended function adversely affected." The terms "safety" and "stability" are often used interchangeably, yet it is clear from many of the papers presented that stability may be impacted by many other factors more commonly considered under the umbrella term, safety.

Another theme, prevalent through many of the papers, was the role of human factors upon stability and safety. It is clear that development of improved regulations alone will not improve ship stability unless proper consideration is given to such human factors as technical competence of operators, education, training, their interaction with hardware, software and the work environment.

Another trend emerging from several papers presented is the promising future of advanced numerical simulation techniques. With the advent of greater capacity computers, the ability to perform higher order simulations is no longer just a tool of the academic researcher, but is becoming available to ship designers as well. These numerical simulations can examine the influence of various geometric configurations under designated wave conditions. Together with tank model data, a better understanding of ship behavior including its stability in non-linear environments, can result in more seaworthy designs.

At the same time, real-time monitoring devices are available to measure vessel motions and stresses while underway. This data, collected by "black boxes," together with better and more wave/wind data, will eventually allow ship designers to observe how their designs behave during real-life conditions.

Another theme brought out during the Operations session, was how well vessel routing to avoid heavy weather has done. While ships will still need to be designed to meet the most severe environments anticipated for their routes, it is sound practice to avoid severe storms.

WORKSHOP

Based upon the papers presented in the three–day Symposium and the questions and answers immediately following each paper, a Workshop convened to discuss the relevance of the reported research to development of improved regulations. The Workshop was essentially by invitation only. All of the authors were invited to attend as well as a number of the attendees who expressed a deep interest in contributing their time and effort to accomplish the Workshop goals. Approximately forty of the attendees participated in the three working group sessions that were held on the day following the Symposium. Among the attendees, there was a good breadth of knowledge and understanding of the problems being addressed.

The workshop was organized into three working groups. Group A, the Standards Development Group, was led by Mr. Otto Jons of Advanced Marine Enterprises, Inc., of Arlington, Virginia. Group A undertook the topic of how standards need to be developed with respect to current research. Group B dealt with the role of Engineering and Technology in stability regulatory development and was led by Professor Kuo of the University of Strathclyde in Glasgow, Scotland. And the last, Group C, explored the role that Vessel Operations and Human Factors should play. This Group was led by Dr. Jack Spencer of ABS Americas in Houston, Texas.

The Groups were charged with evaluating the stability research areas of interest and determining their priorities. To accomplish this they were given 4 ratings. The first was importance to marine safety by estimating benefit versus cost and time to achieve results. Each of the topics were to be evaluated according to the following criteria:

<u>Cost</u>
High (H) – over $500,000
Medium (M) – $100,000 to $500,000
Low (L) – under $100,000

<u>Time to Conduct</u>
Long–term (LT) – over 3 years
Near–term (NT) – 1–3 years
Short–term (ST) – less than 1 year

The Groups also were requested to note for each of their research topics, whether or not existing technology was feasible as opposed to developing new information, what is the promise of significant results, and whether or not the specific research is already underway.

Group A, Standards Development, examined the relationship between standards development and stability research and what was necessary to best use research results to develop good feasible safety standards for ships. The Group developed a framework that could be used to define the area of stability. This framework is shown in Figure 1. The elements of the framework could be addressed independently but with consideration for their mutual interaction.

For each of the elements, specific research projects were delineated in accordance with the instructions the Group had been given. Finally, the Group began rating the projects in accordance with the time and cost criteria set for the Workshop. In addition to the cost and time to conduct rating criteria discussed above, the Group added a rating that considered relative benefit gained from the research and development of better standards in particular project areas.

Benefit
Great (G) - great increase in safety or casualties prevented
Medium (M) -some increase in safety or casualties prevented
Nearly none (N) - small increase in safety or casualties prevented

In the short time available to discuss these research projects, the Group could not reach full agreement on the priorities, however a good start was made to highlight those stability areas where better standards are necessary. The unfinished results of Group A's discussions are given in Table 1.

FIGURE 1

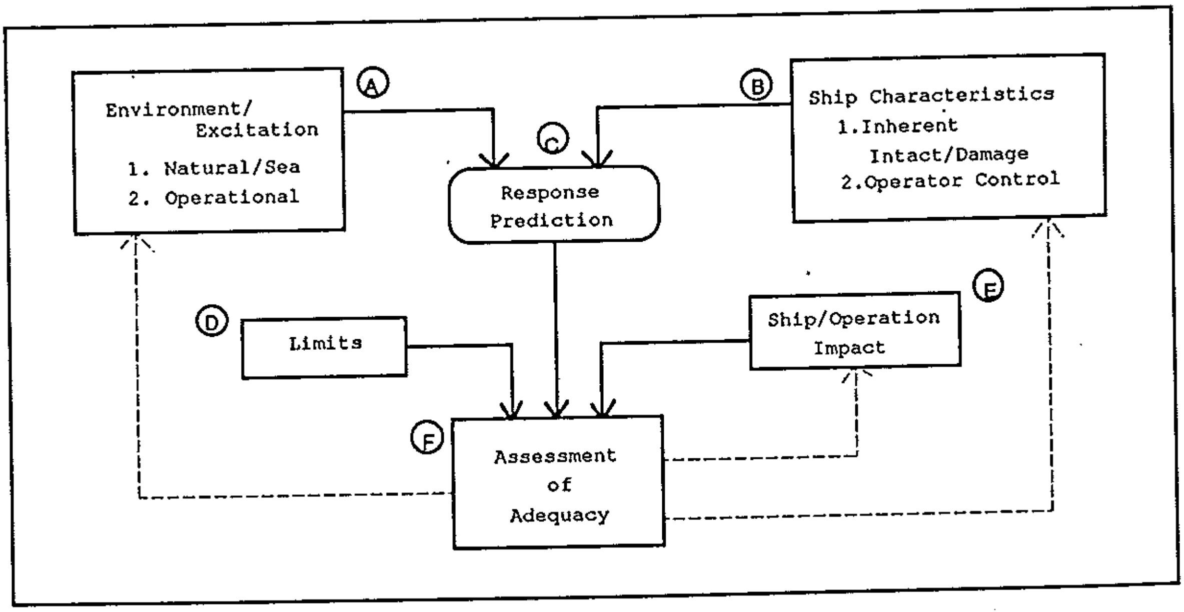

TABLE 1

Research Element	Time	Cost	Benefit
A. Environment/Excitation			
1. Standard definition of a seaway (also for varying regions; "wave atlas"			
2. Definition of Standards (for analysis & testing purposes) – regular/irregular waves – degrees of steepness			
3. "Climatic" seaway definition for analysis purposes	LT	L	G
4. Instrument vessels to measure response and correlate with actual hydrostatic forces	LT	M	M
B. Ship Characteristics			
1. Standard hull form data format	NT	L	M
2. Small ships – simple screen test for possible design problems			
3. Test criteria (in transit) standard seaway			
C. Response Prediction/Testing			
1. Analysis approaches/tools physics based – Systematically assess – Run test cases – Compare results – Set standards	LT	H	G
2. Develop standard approaches for validation/testing and recording of results			
3. Capture/record/document past tests			
D. Limits			
1. Survey/compile/assess existing standards as basis for limits			
– variation for different operating areas develop criticality matrix (ship type vs hazard) – levels of acceptable risk			
2. Review casualty data bases – develop standard casualty format – compile – provide for electronic access			
E. Assessment of Adequacy			
1. 3-D analysis of approach to current standards (development/vessel type vs hazard vs risk – casualty rate)	ST	L	M
2. Amalgamation of current standards (SOLAS/ ICLL etc)			
3. Revisit current Rahola based criteria based on current technology and hull forms	NT	L	N
4. Operations manuals; format and motivation for use			
5. Training of "existing" crew			
F. Ship / Operation Impact			
TBS			
1. Response prediction certification			
– correct physics	NT	M	G
– correct mathematics	ST	L	G
– verified by experiment	NT	H	M
– verified by full scale	LT	H	G
2. Interrelation of stability			
– securing of cargo	NT	M	G
3. Priorities/casualty statistics	ST	L	G
4. Produce guidance notes for criteria (to improve consistent use of criteria)			
5. Logical aspects, national/Int'l regulations (retroactive application)			

The goal of Group B, Engineering and Technology, was to examine what role engineering and technology should play in stability development. Specific engineering research or technological developments discussed during the preceding Symposium were examined and evaluated. Ultimately this Group developed a stability goal: "To provide a relevant and cost-effective engineering/technical capability that will contribute positively to clients' needs in relation to ship stability." Group B was charged by Professor Kuo to come up with a "wish list" of research topics they each would like to pursue and then evaluate each according to the rating criteria given above.

Eighteen research topics were then discussed and evaluated. Table 2 lists these topics. Although there was extensive debate and not always universal agreement among Group B participants, consensus was reached on the relative costs and time to conduct these topics which will be discussed below.

TABLE 2

No.	THEMES/PROJECT	RATING	
		Cost	Time
1	Understand the mechanism of capsize	M	LT
2	Devise methods of applying available research results	L	ST
3	Carry out time-domain simulation of capsizing	H	LT
4	Acquire full scale/model data for correlating with theory	H	LT
5	Study the role of control surfaces in enhancing ship stability	L	ST
6	Treat ship behavior as a dynamic system	L	ST
7	Develop methods for sharing technical information	M	LT
8	Determine critical parameters	M	NT
9	Evolve probability safety measures	M	LT
10	Determine the cost of achieving a specific level of stability	L	LT
11	Devise new criteria (to include dynamics)	M	LT
12	Study non-hull-form-related factors affecting stability	M	NT
13	Define risk level in stability criteria	M	NT
14	Interface engineering approaches with operational management	L	NT
15	Devise methodology for treating stability as an aspect of safety	L	NT
16	Identify hazards and do risk assessment in ship stability	L	NT
17	Improve the integration of engineering with human factors	M	NT
18	Prepare a technical definition of undesirable features of stability	L	NT

Group C, Operations and Human Factors, examined the role of Operations Safety and Human Factors research and its part in regulatory development. Group C deviated from defining specific research projects and concentrated on identifying things that would prevent dangerous situations from occurring and defining the boundaries of proper corrective actions. The Group identified many issues and a framework around which research projects can be more easily developed.

Among the issues that the Group determined needed more examination are: Major hazard identification (compilation of casualty history, in-service records and research literature), identification of areas that the designer can alleviate through his ship design or information provided to the operator on corrective actions (relations between dangerous scenarios and various corrective actions, use of expert systems, and operator training and experience), importance of planning for contingency situations, the management aspect and its effect on safety, role of simulators in training, and the need for cost effective research.

Again for Group C, as with all the others, time was against significant progress. However, this Group's discussions demonstrated that the need for coordination and planning is of the highest importance, if we are to effectively target research that will have the greatest impact on improving vessel safety in hazardous stability situations.

CONCLUSIONS

In spite of the "Storm of the Century" which delayed or prevented many people from attending the Symposium/Workshop, over one hundred did register. Twenty-four papers were presented or submitted representing several countries. Seven ship owners or operators had attendees and three navies were represented. Topics presented represented the broad spectrum of ship stability-related research and provided the basis for extensive discussion at the Workshop. Over forty experts attended the workshop and the groundwork was accomplished to establish a mechanism for developing a stability research plan by the U.S. Coast Guard and, hopefully, the international stability community.

In the Workshop, Group A, "Standards Development," stressed the importance of having a framework which can be used to define broad areas of interest before identifying suitable topics for vessel stability research. One theme carried through Group A's discussion was the need to use "standard" methods for collecting, processing, assessing and applying data relating to ship stability.

In Workshop Group B, "Engineering/Technology," each participant's research idea was reorganized into a set of eighteen themes and projects. An attempt was made to assign cost and time priorities to each and the findings are summarized in Table 2.

And Workshop Group C, considered vessel operations and human factors with the emphasis on safety rather than just stability. To achieve the best results, the Group advocated joint action by all the parties concerned such as designers, regulatory bodies and ship operators.

Symposium papers and Workshop Proceedings have been published in a two-volume Coast Guard Research and Development Center Technical Report which will be available through the National Technical Information Service (NTIS).

Following the workshop, the raw notes kept by each Group leader were refined. An attempt was made by Group B's leader, Professor Kuo, to develop broad research themes into which the research "wish" topics could be assigned. This attempt was based upon all ideas drawn from all Symposium papers, Group discussions, and questions and answers. Seven themes were identified and are described below:

a. Fundamental Studies, whose objective is to perform fundamental research studies on key issues which will provide improved understanding of stability in the broad context of safety includes:

(1) Devise methodology for interfacing stability with safety

(2) Apply safety "case" approach to ship stability

(3) Define risk levels in stability criteria

(4) Understand the mechanism of capsize

b. Role of Human Factors, i.e., to examine how research advances in human factors can be incorporated into consideration by everyone directly or indirectly involved with ship stability, whether owning, designing, operating, regulating or supporting. This theme includes topics:

(1) Understand the impact of human factors or stability

(2) Improve collaboration and communication between users and providers

(3) Enhance the education of designers and operators in risk assessment techniques

(4) Overcome non-compliance with operational procedures.

c. Stability Information, whose objective is to devise methods for enabling the best use to be made of the stability information available from all sources, and to improve access to this information. Includes:

(1) Develop methods for sharing technical information

(2) A technical definition of undesirable features of stability

(3) Build up a database on accidents due to various forms of stability deficiency

(4) Assemble wind load data for use in meeting stability criteria

(5) Standardize the collection of full-scale and model data

(6) Devise computer graphics for use in stability studies

d. Theoretical Studies, whose objective is to acquire a better understanding and new knowledge in theoretical aspects of ship stability. This theme includes topics:

(1) Perform time domain simulation of capsizing

(2) Treat ship behavior as a dynamic system

(3) Determine critical parameters of ship stability

(4) Study the role of control surfaces in ship stability

(5) Develop a national capability for assessing vessel stability based upon computing tools

(6) Devise probability methods of assessing loss from damaged stability

(7) Develop linear models of gravity waves

(8) Develop stational stochastic models for non-linear waves

(9) Study more realistic modeling methods for roll motion with stability assessment

(10) Consider ship capsize via a non-linear system dynamic analysis approach

(11) Determine more accurately the excitation forces affecting stability

e. Experimental Studies with the objective to perform carefully designed experiments to obtain information on stability that will both enhance an understanding of the theory and improve the methods of applying the available knowledge. Includes:

(1) Acquire full–scale/ship model data for correlating with theory

(2) Devise methods for calculating floatability and recoverability of damaged vessels

(3) Develop model testing methods to provide increased statistical reliability in fewer runs

(4) Examine the use of on–board facilities to overcome stability–sensitive deficiencies

f. New Ship Designs, the objective of which is to seek solutions to identified problems and potential difficulties for new ship designs which have recently come into operation or are expected to be introduced to the industry in the near future. Topics include:

(1) Identify key stability features deserving attention

(2) Apply stability research findings to small craft

(3) Devise new stability criteria for small craft

(4) Examine the damage stability of high–speed catamarans

(5) Consider stability aspects of new designs that will influence safety

g. Application of Research Studies, whose objective is to ensure that advances made in various research areas can be used by practitioners in their daily activities. The emphasis here should be on both the quality and user–friendliness of the research results. Includes:

(1) Devise effective methods for applying research results

(2) Devise new stability criteria to include effects of dynamics

(3) Evolve probability–based safety measures

(4) Relate the level of a vessel's stability with its cost

(5) Test the applicability of existing regulations for identified seaways using appropriate methods

(6) Validate computer software for stability calculations

(7) Enhance international cooperation on vessel stability

While insufficient time existed within the one-day workshop to fully debate the merits of promoting each individual research topic, the framework presented by the Group B moderator, Dr. Kuo, provides a starting point for organization of future research. Further discussion is needed by the international stability community to identify those stability topics offering the most promise.

FUTURE OF COORDINATED STABILITY RESEARCH

The accomplishment of this Symposium reinforced our belief that some type of international forum is needed to try to work together to prioritize and accomplish necessary expensive stability research projects. The planning and coordination of such research can be greatly improved by the formation of an international committee whose major aim would be to direct relevant research on the stability of ships and marine vehicles.

This committee would meet on an agreed timetable to discuss common stability research and exchange research results. The committee could be used to develop recommendations for further research where uniformity is needed and where multiple countries are involved. Major meetings of such a Committee could be held in conjunction with the STAB Conferences with other smaller meetings being held as necessary between the STAB Conferences. The Committee goal should be for the improvement of maritime safety with a membership that would represent equally industry, academia, and governments.

This proposed Committee framework is open to change and was first presented by Dr. Spencer at STAB'90 (Reference 2). However, we believe it is vital that actions be taken by the leaders in international stability research to provide an opportunity for all those involved in stability research to meet and exchange information and use their combined knowledge to direct cooperative research to solve specific stability challenges. This need is as yet, unfulfilled.

ACKNOWLEDGMENT

The authors wish to thank CDR(sel) Randall R. Gilbert of the U.S. Coast Guard and Dr. John S. Spencer of ABS Americas for their dedicated belief in the true need for this Symposium on stability research. Without their initial efforts and continuing strong

support for the Symposium, this step forward for cooperative stability research would probably never have been taken.

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STABILITY ASSESSMENT FOR FLOATING STRUCTURES IN REALISTIC SEAS

by

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May, 1994

ABSTRACT

The main objective of this paper is to present a reliable and easy-to-apply method for the assessment of a floating structure's (ships and floating platforms) transverse stability at sea. The theory of Markov processes will be used to derive the equations for the random decrement and the autocorrelation function of the nonlinear rolling motion for a ship sailing in a random oblique sea. The wave exciting moment, used as an input to the roll equation, is obtained as the output of a narrow band filter whose input is a White noise Gaussian process. Thus, the theory of Markov processes can be used to model the roll motion of a ship acted upon by a wave exciting moment, which has a realistic spectral density. The random decrement and the autocorrelation functions will be used to obtain estimates for the nonlinear righting moment of the ship. The identification method proposed uses only information obtained from roll measurements at sea. Results of numerical simulations, model tests and a full scale fishing vessel will be presented.

A method for the assessment of the margin of safety against capsizing will be developed. The method can be used to provide warning when a dangerous situation is encountered by the ship. It will also help naval architects develop reasonable safety criteria.

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[1]Professor of Naval Architecture

[2]Graduate Student

I. INTRODUCTION

Ship safety against capsizing is a major quest for every naval architect. Designing safety against capsizing is not a simple task. Several factors may contribute to the loss of a ship at sea. Ship survivability is not only a function of the hull form, but is also a function of loading and environmental conditions. While transverse stability can be investigated for a certain ship at extreme loading conditions, there is no control over extreme environmental conditions to which a ship can be subjected to. Human factors also play a major role. Statistics for shipping accidents reported by the Transportation Safety Board of Canada [1] show that human factors played a primary role in 55 percent of the shipping accidents over the past five years. There is a need for a simple and reliable method for the assessment of ship's stability at sea to help the captain make the right decision when faced with a dangerous situation. We may differ about the definition of stability and how it is related to the safety of ships against capsizing. However, what a ship's captain needs is a simple measure of how his\her ship will perform in a certain condition. The metacentric height has been used in the past, and it will continue to be used, at least for some time to come, as a measure of the transverse stability of a ship or a floating structure.

In this work we would like to report on a method for the estimate of the metacentric height of a ship at sea. The method is based on the analysis of the measured random rolling motion of the ship. Most available parametric identification techniques are based on relating input and output signals. Very few studies exist in the literature which describe parametric identification techniques applied to ships at sea, where it is difficult to measure the input to the system. Roberts et al [2] used a method of equivalent linearization to determine the values of the parameters in the nonlinear equation of motion for a ship rolling in random beam waves. The excitation is assumed to be White Gaussian noise. This method can be applied for systems with weak nonlinearities. The authors reported that the method yielded good results when tested using digitally generated data. Results using experimental data was not presented. The concept of the random decrement was extended to the case of nonlinear rolling motion in beam seas by Haddara [3]. It was shown that the expected value of the rolling motion excited by stationary White noise random process approximately satisfies the free decay roll equation. It was also shown that the autocorrelation function of the same process satisfies a linearized free decay roll equation. The method yielded good estimates for the damped nonlinear natural frequency of rolling motion which can be used to estimate the magnitude of the meta-centric height at sea. An equivalent damping coefficient was also obtained. The method was also used in conjunction with a function modulation technique, Shinbrot [4], to identify various parameters in the free, forced and randomly excited rolling equation (see Haddara et al [5] and [7] and Zhang and Haddara [6]). In these studies, the wave excitation was assumed to consist of a White noise, Gaussian random process.

The main objective of this study is to develop a method for predicting the parameters in the differential equation describing the nonlinear roll motion of a ship excited by random beam waves. The method uses the roll motion record as the only input. The excitation is assumed to have a spectral density given by the standard JONSWAP form. The method is capable of estimating the linear and nonlinear coefficients of the restoring and damping moment. The method is applied to both digitally generated and experimental data.

II. EQUATION OF ROLL MOTION IN RANDOM BEAM SEAS

The equation of motion for a ship rolling in random beam seas can be written as (see Haddara [3]):

$$\ddot{\phi} + N(\dot{\phi},\phi) + D(\phi) = \nu K(t) \tag{1}$$

where ϕ is the roll angle, $N(\dot{\phi},\phi)$ $D(\phi)$ and ν $K(t)$ are the nonlinear damping, nonlinear restoring and the wave exciting moments per unit virtual moment of inertia of the ship, respectively. A dot over the variable indicates differentiation with respect to time.

Roll damping is traditionally considered a nonlinear function of the roll velocity which can be expressed as the sum of two terms: a linear term and a quadratic term. It has also been expressed in the literature as a nonlinear function of the roll angle and roll velocity. In this study we will assume that $N(\dot{\phi},\phi)$ is given by:

$$N(\dot{\phi},\phi) = 2\zeta\omega_o(\dot{\phi} + \epsilon|\dot{\phi}|\dot{\phi}) \tag{2}$$

where ζ and ϵ are the nondimensional linear and nonlinear damping coefficients, respectively. ω_0 is the natural frequency of the linear roll equation. The method of analysis introduced here can deal with any of the known forms of roll damping.

The restoring moment is a function of the form of the underwater part of the ship hull and the weight distribution. It is usually expressed as an odd series in the roll angle. Thus, $D(\phi)$ can be expressed as:

$$D(\phi) = \omega_o^2(\phi + \alpha_1\phi^3 + \alpha_2\phi^5) \tag{3}$$

where α_1 and α_2 are constants.

The excitation $K(t)$, is assumed to be a narrow band Gaussian process which satisfies the following equation:

$$\ddot{K}(t) + \gamma\,\dot{K}(t) + \Omega^2\,K(t) = u(t) \tag{4}$$

where $u(t)$ is a random Gaussian process which satisfies the following conditions:

$$\begin{aligned} \langle\, u(t)\, \rangle &= 0 \\ \langle\, u(t_1)\; u(t_2)\, \rangle &= \delta\,(t_1 - t_2) \end{aligned} \tag{5}$$

where $\langle\ \ \rangle$ means the ensemble average of the process and δ is the Dirac delta function. γ is the band width and Ω is the central frequency for the narrow band process $K(t)$. Equations (5) imply that $u(t)$ is a White noise random process with zero mean and unit variance.

(i) Expected Value of Roll Angle

The expected value of the roll angle can be obtained as:

$$\mu(t) = \langle\, \phi(t)\, \rangle = \frac{1}{N}\sum_{i=1}^{N} \phi_i(t) \qquad 0 \le t \le t_{max}$$

If the measured record of the roll angle is divided into portions of equal length and having the same starting conditions, and the process of ensemble averaging is applied to these portions, one gets an equation describing the propagation of the expected value of the roll angle as a function of time, as:

$$\ddot{\mu}(t) + 2\zeta\omega_o\dot{\mu}(t) + \omega_o^2\mu(t) + 2\zeta\epsilon\omega_o\langle\dot{\phi}|\dot{\phi}|\rangle + \alpha_1\omega_o^2\langle\phi^3\rangle + \alpha_2\omega_o^2\langle\phi^5\rangle = 0 \quad (6)$$

In deriving equation (6), we have assumed that $K(t)$ is a zero mean Gaussian random process which is justified because of the relationship given in equation (4).

It is thus seen that the expected value of the roll response of a ship, excited by a narrow band random process, satisfies a differential equation similar to the one that describes the free rolling motion of the ship. This derivation is a generalization of that given by Haddara and Wu [5], where it was assumed that the ship is excited by a White noise process. A linearized form of Equation (6) has been referred to in the literature as the random decrement. The derivation given here extends the concept of the random decrement to the case of nonlinear, time invariant dynamic systems subjected to stationary narrow band zero mean Gaussian random processes.

(ii) Autocorrelation Function

It has been shown by Haddara [3] that the autocorrelation function of a nonlinear, time invariant system excited by a zero mean, stationary, Gaussian, White noise random process satisfies the differential equation describing the free motion of an equivalent linear system. This linear system has equivalent linear damping and restoration which can be obtained from an equivalent linearization scheme. In this work, we would like to extend this to the case when the excitation is a narrow band random process. Multiplying equation (1) by $\phi(t+\tau)$, and performing the ensemble averages, one can obtain the following differential equation:

$$\ddot{C}_{11}(\tau) + 2\zeta\omega_o\dot{C}_{11}(\tau) + \omega_o^2 C_{11}(\tau) + 2\zeta\omega\epsilon f(\tau) + \alpha_1\omega_o^2 g(\tau) + \alpha_2\omega_o^2 h(\tau) = \nu C_{21}(\tau) \quad (7)$$

where $C_{11}(\tau)$ is the autocorrelation function of the roll angle given by:

$$C_{11}(\tau) = \langle\phi(t+\tau)\phi(t)\rangle$$

and

$$f(t) = \langle\ \phi(t+\ \tau)\dot{\phi}(t)\,|\,\dot{\phi}(t)\,|\ \rangle$$

$$g(\tau) = \langle\ \phi(t+\tau)\phi^3(t)\ \rangle$$

$$h(\tau) = \langle \phi(t+\tau)\phi^5(t) \rangle$$

$$C_{21}(\tau) = \langle\ \phi(t+\tau)K(t)\ \rangle$$

Dots over $C_{11}(\tau)$ indicate differentiation with respect to τ. In writing equation (7), we have used the definitions for the first and second derivatives of the autocorrelation function of a stationary random process.

Multiplying equation (4) by $\phi(t+\tau)$ and taking the averages, one gets:

$$\ddot{C}_{21}(\tau) + \gamma\dot{C}_{21}(\tau) + \Omega^2 C_{21}(\tau) = \langle\ \phi(t\ +\ \tau)u(t)\ \rangle \tag{8}$$

where $C_{21}(\tau)$ is the cross correlation between the roll angle and the excitation $K(t)$.

There are several methods in the literature that can be used for the estimation of the parameters of equation (6). In Zhang and Haddara [6], the authors used the modulating function technique for the estimation of the parameters in the free decay roll motion equation. The same analysis can be applied to this equation after evaluating the averages in the equation. This can be done numerically since the roll response is known. However, using equation (7) in estimating the values of the parameters of the nonlinear damping and restoration is more complicated than using equation (6). In the case of equation (7), we have to find a way to calculate the cross correlation between the input and output. We will discuss this in the next section.

III. MODULATING FUNCTION TECHNIQUE

Zhang and Haddara [6] presented this technique for the estimation of the parameters in the equations describing the free decay roll motion and the forced roll motion excited by regular waves. Using this technique, the authors were able to estimate the parameters from the roll response only. The technique has been originally developed by Shinbrot [4] to estimate the parameters of dynamical systems using input output relationships. Here, we will present the method briefly.

Define an operator $\Psi_k[f(t)]$ such that:

$$\Psi_k[f(t)] = \int_0^T f(t)A^k(\tau)\, dt, \quad k=0,\ldots,n. \tag{9}$$

where

$$\begin{aligned} A^k(\tau) &= \exp(-\tau^2/2)\, H_k(\tau) \\ &= (-1)^k \frac{d^k}{d\tau^k}[\exp(-\tau^2/2)], \quad k=0,\ldots,n. \end{aligned} \tag{10}$$

and $H_k(\tau)$ is Hermite polynomial of order k and

$$\tau = \frac{t}{T}(T_e + T_s) - T_s$$

T_e and T_s are chosen such that $A^k(T_e) = A^k(-T_s) = 0$.

The functions $A^k(\tau)$ satisfy the following orthogonal relationship:

$$\int_{-\infty}^{\infty} \exp(\tau^2/2) A^m(\tau) A^n(\tau)\, d\tau = n! \sqrt{2\pi}\, \delta_{mn}$$

where δ_{mn} is Kronecker delta. They also satisfy the following recursion relationships:

$$\tau A^{n}(\tau) = A^{n+1}(\tau) + nA^{n-1}(\tau)$$

$$\frac{dA^{n}(\tau)}{d\tau} = -A^{n+1}(\tau)$$

Equations (6) and (7) can be rewritten in the following form:

$$\ddot{x}_k(t) + B_o\dot{x}_k(t) + A_o x_k(t) + B_1 p_k(t) + A_1 q_k(t) + A_2 s_k(t) = v_k(t) \tag{11}$$

where

$$x_1 = \mu$$
$$x_2 = C_{11}$$

$$p_1 = \langle \phi|\phi| \rangle$$
$$p_2 = f(t)$$

$$q_1(t) = \langle \phi^3(t) \rangle$$
$$q_2(t) = g(t)$$

$$s_1(t) = \langle \phi^5(t) \rangle$$
$$s_2(t) = h(t)$$

$$\nu_1(t) = 0$$

$$\nu_2(t) = \nu\, C_{21}(t)$$

Applying the operator Ψ_k to equation (11), one gets:

$$\begin{gathered} \Psi_k[\ddot{x}_j] + B_o \Psi_k[\dot{x}_j] + A_o \Psi_k[x_j] \\ + B_1 \Psi_k[p_j] + A_1 \Psi_k[q_j] + A_2 \Psi_k[s_j] = \nu\ \Psi_k[\nu_j], \\ j=1,2 \wedge k=0,\ldots,m. \end{gathered} \tag{12}$$

Equation (12) can be rewritten as:

$$\begin{gathered} \beta B_o \Psi_{k+1}(x_i) + A_o \Psi_k(x_i) \\ + B_1 \Psi_k(p_i) + A_1 \Psi_k(q_i) + A_2 \Psi_k(s_i) - \nu\ \Psi_k(\nu_i) \\ = -\beta^2 \Psi_{k+2}(x_i), \qquad i=1,2 \wedge k=0,\ldots,m \end{gathered} \tag{13}$$

where

$$\beta = \frac{(T_e + T_s)}{T}$$

Equations (13) are an m number of simultaneous algebraic equations in five unknown parameters. If we limit m to 5, the equations can be easily solved to find the magnitude of the five parameters B_0, A_0, B_1, A_1 and ν. We can also take m to be a large number and use a regression technique to the values for the five unknowns. In this study, we found no need to consider more than five equations.

IV. RESULTS AND DISCUSSION

The method described above was tested using both numerically generated and experimental data. A random roll angle record is generated by numerically integrating equations (1) and (4). The values for the central frequency and the bandwidth are equal to 3.0

rad/sec and 0.5, respectively. The random response is then used to calculate the random decrement and the autocorrelation function. The values for the parameters used in the simulation and their estimated values obtained from applying the identification technique to equations (6) and (7) are shown in Table 1.

TABLE 1: Values of the parameters from numerical simulations

	B_o	B_1	A_o	A_1	A_2	K
Actual	0.1	0.1	9	1	0	10
Auto Cor	0.103	0.097	9.056	0.888	0	9.710
Ran Dec	0.106	0.096	9	1.137	0	10.153

Experimental roll response in random seas was obtained for a 1.5 meter model. This is 1 to 6.8 scale model of a fishing vessel.

TABLE 2: Parameters obtained from model experiments

	B_o	B_1	A_o	A_1	A_2
Free Roll	0.12	0.166	11.217	9.108	-0.057
Auto Cor	0.119	0.004	11.396	1.335	-0.008
Ran Dec	0.115	0.003	11.444	2.236	-0.014

Table 2 shows the values for the roll equation parameters for the model obtained from the analysis of the free roll response of the model, the autocorrelation equation, and the random decrement equation. It is seen from the Table that the estimates obtained for the linear damping and natural frequency agree well with the values obtained from the free roll response of the model. A better way of comparison between these methods is to use the estimated parameters to generate roll simulations. Figures 1 and 2 show such comparison. In Figure 1, we have a comparison between a roll decay curve obtained using the parameters estimated from the free roll decay curve and that obtained using the parameters estimated from the autocorrelation equation. In Figure 2, the comparison is between two simulations obtained using the parameters estimated from the free roll decay curve and the random decrement equation, respectively. The fact that the agreement in these two Figures is excellent indicates that the slight differences in the estimated values of the nonlinear parameters are not critical to the response prediction. However, experimental results for severe motions should be analyzed before this question is settled completely. It should be mentioned that, in using the autocorrelation function approach, we have assumed that the cross correlation function C_{21} is zero. The good agreement shown in Figure 1 indicates that this approximation is permissible.

One of the main objectives of this study is to find an index for the transverse stability which can be used by the captain at sea. The technique presented in this paper is successful in obtaining accurate estimates for the natural frequency of the rolling motion. One can then use the natural frequency as a measure of the transverse stability of the ship. Figure 3 shows the relationship between the square of the natural frequency of rolling motion, as predicted by the method of analysis described in this work, and the actual values of the metacentric height for the ship model. The linear relationship indicates the suitability for using the square of the natural frequency of rolling motion as a measure for the metacentric height.

The rolling response of a 33 meter fishing vessel at sea is also analyzed using the method described in this work. Figures 4 to 6 show a comparison between measured and predicted autocorrelation functions for three cases. Figure 7 shows the relationship between the square of the natural frequency of rolling motion and the metacentric height of the vessel. Unfortunately, the range of variation of the metacentric height is not wide enough to show the nature of the relationship. However, it is clear that the method of analysis is successful in the parametric identification of the rolling motion equation in realistic seas.

CONCLUSIONS:

In this work, we have presented two methods for the identification of rolling motion parameters of a ship from its roll response in a realistic sea. One can use the random roll response to calculate either the random decrement or the autocorrelation function. Any one of them can be used for finding estimates for the damping and restoring parameters. No knowledge of the excitation is required. In the present paper, we have considered the excitation to be a narrow band random process with zero mean. This seems to be a reasonable assumption. Good agreement is obtained between measured and predicted autocorrelation functions, even when the cross correlation between the response and the excitation is set to zero.

The square of the natural frequency of the rolling motion can be used as a measure for the metacentric height at sea. However, more data need to be examined for this result to be conclusive. A wider range of variation for the metacentric height, as well as responses in more severe seas, are needed.

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Figure 1

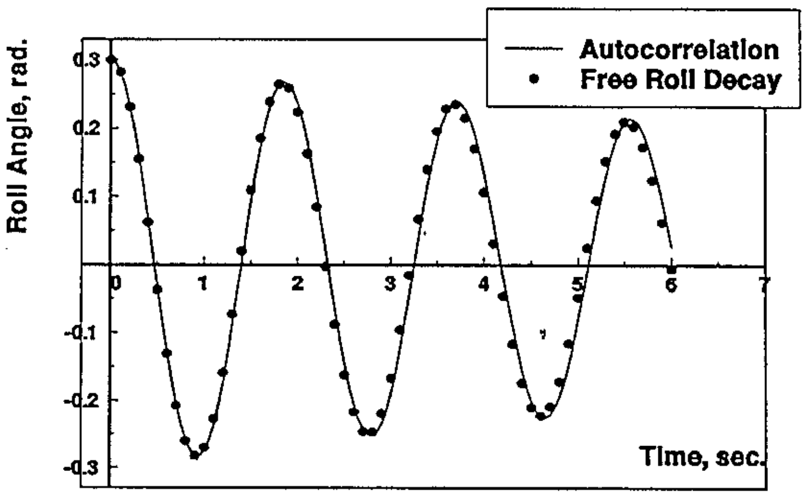


Figure 3

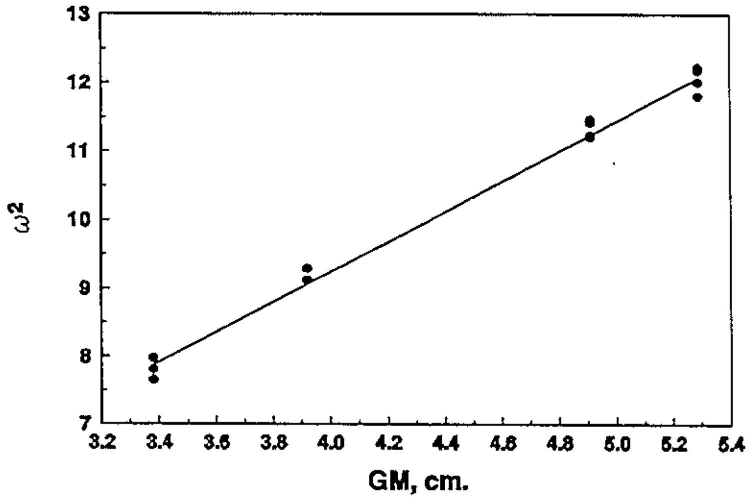


Figure 2

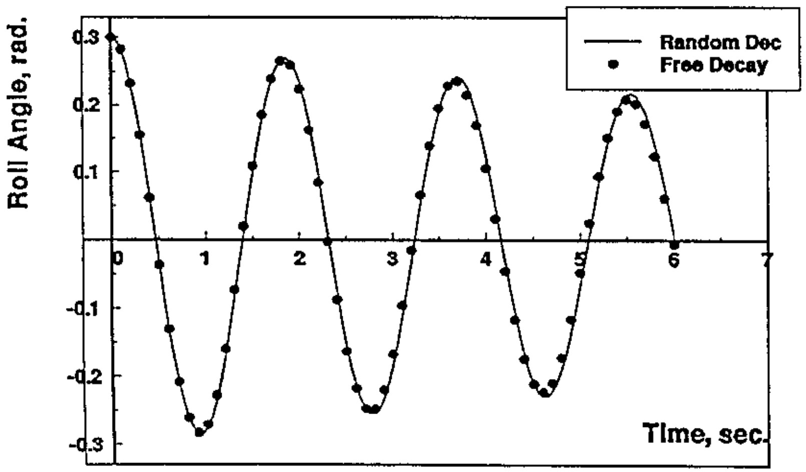


Figure 4

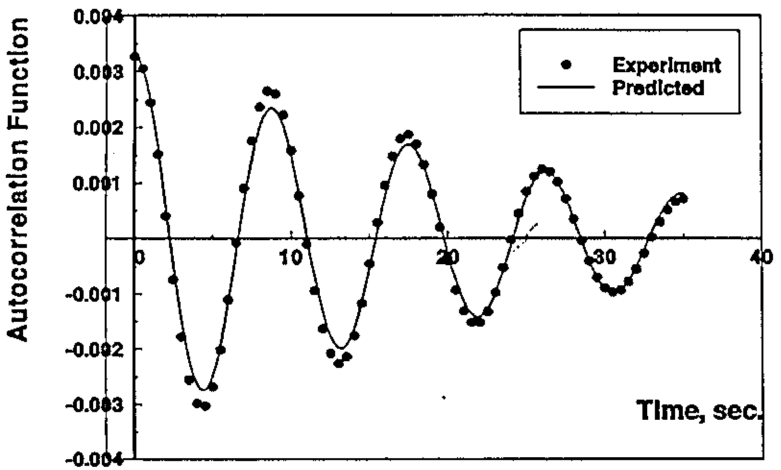


Figure 6

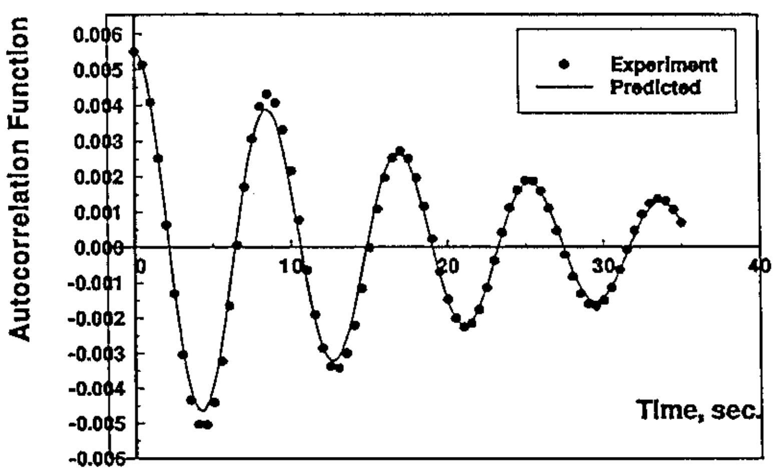


Figure 5

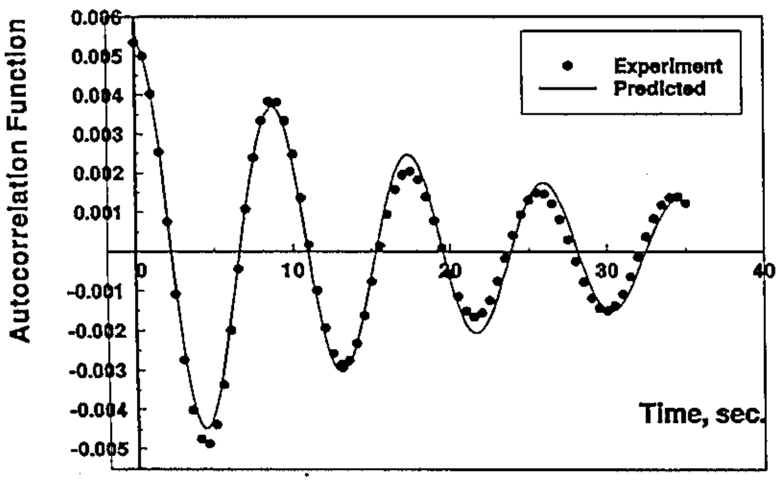
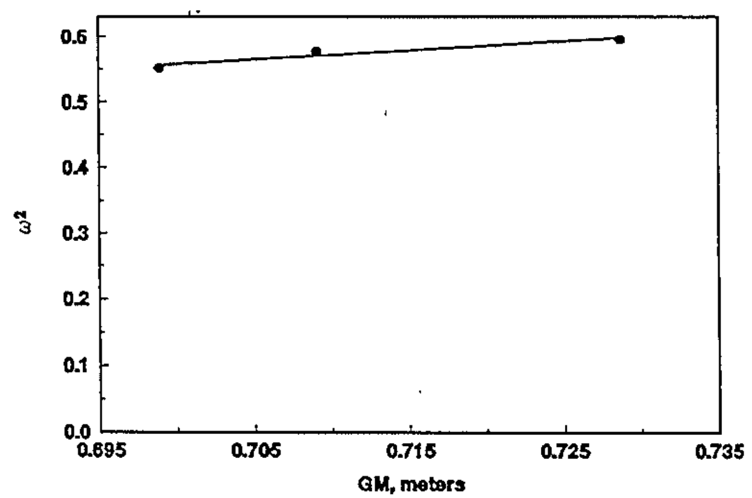


Figure 7



ANALYSIS OF I.M.O. STABILITY CRITERIA BY SYSTEMATIC HULL SERIES AND BY SHIP DISASTERS

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ABSTRACT

The problem of ship safety against capsizing is still now a pre-eminent one, especially for small vessels, as it was dramatically pointed out in the last year by the casualties of fishing vessels and ferries with loss of human lives.

By means of the regression analysis and of the geometrical similarity laws applied to the systematic hull series, first aim of this paper is to show the possibility to analyse methodically the influence of the ship's main dimensions and of the different loading conditions on the intact stability and to show also the possibility to have cases of bad seakeeping characteristics in order to verify all the stability indices.

Second aim is to investigate the recent disaster of a ferry and to point out some operational safety aspects of these ships as well as the possible consequences of loading and ballasting errors.

Last aim of this paper is to propose an integration and a reduction of the I.M.O. stability indices in order to rationalise and simplify the actual Regulations.

NOMENCLATURE

A_S - superstructures area;
B - moulded beam;
C_B - block coefficient;
C_p - prismatic coefficient;
C_∇ - loading conditions coefficient ($C_\nabla = \nabla/\nabla_0$);
D - moulded depth;
E - restoring energy; $\Delta E = E_{40°} - E_{30°}$
f - design freeboard at midship;
$\overline{KG}$ - vertical position of the centre of gravity;
$\overline{KG}^*$ - maximum allowable vertical position of the centre of gravity;
GM - metacentre height;
GZ - righting lever;
h_S - vertical distance from the centre of windage area to the centre of the underwater lateral area;
L - length between perpendiculars;
T - moulded draft;
∇ - actual displacement volume;
∇_0 - design displacement volume;
φ_x - angle of heel at which the righting lever GZ is maximum;
φ_d - flooding angle;
b/a - areas ratio (I.M.O. Weather Criterion).

INTRODUCTION

The problem of safety against ship capsizing is very important especially for the relatively small ships, which are more sensitive than the bigger ones to the environmental conditions as dramatically pointed out by casualties with loss of life recorded in last years.

As it is well known, nowadays I.M.O. suggests using for the intact stability of all the ships, included fishing vessels having length of 24 m and over, both the statistical intact criteria (Resolution A.168) and the severe wind and rolling criterion (Resolution A.562).

Many theoretical and experimental researches carried out during the last thirty years allow nowadays a good understanding of the various causes of ship capsizing, but until now it has not yet been possible either to transfer the results of these researches into a new simple criterion or to use these results to check easily the intact stability both in the design and in the operative stage.

The aim of the actual I.M.O. stability criteria is to verify ship's intact stability conditions by means of simple regulations in order to obtain a standard probability level of ship's safety against capsizing by considering both the geometrical characteristics (of the hull and of the superstructures) and the operative conditions of the ships.

By means of the geometrical similarity laws and the regression analysis the I.M.O. stability indices were calculated by the authors for about nine thousands different combinations of hulls and loading conditions for a methodical hull series of fishing vessels.

The results so obtained, allowed us to carry out the next steps:

- to analyse methodically the influence on the intact stability both of the geometrical and form parameters and of different loading conditions of the fishing vessels;
- to make a comparison among the various stability indices;
- to point out the excessive GM value obtained in some cases to verify the most severe stability indices.

By taking into account the results obtained the authors think that the I.M.O. stability criteria could be applied also in the future in order to verify ship's intact stability, but with suitable modifications.

STABILITY CRITERIA FOR THE SYSTEMATIC SERIES HULLS

According to the geometrical similarity laws applied to the systematic series hulls, we obtained the following functional relations:

$$\frac{I}{B} = \left(C_\nabla, C_B \left(\text{or } C_p\right), \frac{B}{T}, \frac{f}{B}, \frac{\overline{KG}}{D}\right) \qquad (1)$$

$$C_{\varphi_x} = \frac{B}{T}\tan\varphi_x = \left(C_\nabla, C_B \left(\text{or } C_p\right), \frac{B}{T}, \frac{f}{B}, \frac{\overline{KG}}{D}\right)$$

where I is the generic statistical stability index $\left(\overline{GM}, \overline{GZ}_{30°}, E_{30°}, E_{40°}, E_{40°} - E_{30°}\right)$ and φ_x is the angle of heel at which the righting lever GZ is maximum.

The dependence of the generic index I from length-displacement ratio $L/\nabla^{1/3}$ and from the design displacement volume ∇_0 is expressed by the relation:

$$B = \left[\nabla^{2/3} \cdot \frac{B}{T} \Big/ C_B \cdot \left(\frac{L}{\nabla^{1/3}}\right)\right]^{1/2} \qquad (2)$$

As far as the weather criterion is concerned, referring again to a systematic series hulls, the following relation gives the ratio b/a versus not only the same previous geometrical and loading parameters but also versus A_S/LT and h_S/T.

$$\frac{b}{a} = f\left(C_\nabla, C_B\left(\text{or } C_P\right), \frac{B}{T}, \frac{f}{B}, \frac{\overline{KG}}{D}, \frac{L}{\nabla^{1/3}}, \frac{A_s}{LT}, \frac{h_s}{T}\right) \tag{3}$$

By means of the regression analysis it is possible to develop the functional relations (1) and (3) in polynomial form:

$$C_{\varphi_x} = \sum_i a_i C_P^m C_\nabla^n \left(\frac{B}{T}\right)^p \left(\frac{f}{B}\right)^q \left(\frac{\overline{KG}}{D}\right)^r$$

$$I = \sum_i a_i' C_P^m C_\nabla^n \left(\frac{B}{T}\right)^p \left(\frac{f}{B}\right)^q \left(\frac{\overline{KG}}{D}\right)^r \nabla_0^s \left(\frac{L}{\nabla^{1/3}}\right)^t \tag{4}$$

$$\frac{b}{a} = \sum_i a_i^* C_P^m C_\nabla^n \left(\frac{B}{T}\right)^p \left(\frac{f}{B}\right)^q \left(\frac{\overline{KG}}{D}\right)^r (\nabla_0)^s \left(\frac{L}{\nabla^{1/3}}\right)^t \left(\frac{A_s}{LT}\right)^u \left(\frac{h_s}{T}\right)^v$$

Then the maximum allowable KG*/D value which fulfils each index can be obtained by solving the previous equations with respect to KG*/D and by substituting for I and φ_x the minimum value required in the I.M.O. recommendation A.168 or putting b/a equal to unity as suggested by I.M.O. resolution A.562.

So, these KG*/D values can be expressed by means of the regression analysis in the form:

$$\frac{\overline{KG}^*}{D} = \sum_i b_i C_P^m C_\nabla^n \left(\frac{B}{T}\right)^p \left(\frac{f}{B}\right)^q \tag{5a}$$

$$\frac{\overline{KG}^*}{D} = \sum_i b_i' C_P^m C_\nabla^n \left(\frac{B}{T}\right)^p \left(\frac{f}{B}\right)^q \nabla_0^s \left(\frac{L}{\nabla^{1/3}}\right)^t \tag{5b}$$

$$\frac{\overline{KG}^*}{D} = \sum_i b_i^* C_P^m C_\nabla^n \left(\frac{B}{T}\right)^p \left(\frac{f}{B}\right)^q \nabla_0^s \left(\frac{L}{\nabla^{1/3}}\right)^t \left(\frac{A_s}{LT}\right)^u \left(\frac{h_s}{T}\right)^v \tag{5c}$$

where the first KG*/D value satisfies the φ_x criterion, the second one satisfies simultaneously all the others statistical indices and the third one satisfies the weather criterion.

THE RIDGELY-NEVITT STANDARD SERIES

The previous procedures have been applied to the Ridgely-Nevitt series of fishing vessels hulls with the following ranges of non-dimensional parameters:

$C_P = .554 \div .700$ $\quad\quad$ $C_\nabla = .60 \div 1.0$

$\nabla_0 = 100 \div 750\ m^3$ $\quad\quad$ $B/T = 2 \div 3.5$

$A_s/LT = .70 \div 1.3$ $\qquad$ $h_s/T = .50 \div 1.1$

$f/B = .05 \div .25$ $\qquad$ $\overline{KG}/D = .60 \div .75$

The polynomial expressions obtained by means of regression analysis for each of the four C_p values of the series parent hulls, with a sample of thousands calculated stability indices, fit very well the data as shown by the values of the correlation coefficient R^2, of the mean values of the dependent variable I and of the standard errors S_E.

In the table I are reported, for example, these values obtained by fourth order equations applied to the sample of all b/a calculated values for each of the four C_p values, being n the total number of calculated values and k the number of terms in the regression equation.

TABLE I

C_p	n	k	S_E	I	R^2
.554	13986	177	.06192	1.78716	.9968
.597	14132	169	.05945	1.82472	.9969
.650	15196	115	.06938	1.75147	.9954
.700	15194	123	.06347	1.70635	.9957

INFLUENCE ON THE STABILITY OF THE MAIN DESIGN PARAMETERS

The maximum allowable value of the vertical position of the centre of gravity KG*, dimensionless with respect to the moulded depth D is the most suitable figure to compare the stability indices.

The curves of KG*/D calculated according to the equations (5) are shown in the figures from 1 to 8 as a function of the main design parameters. Drawing these curves we fixed:

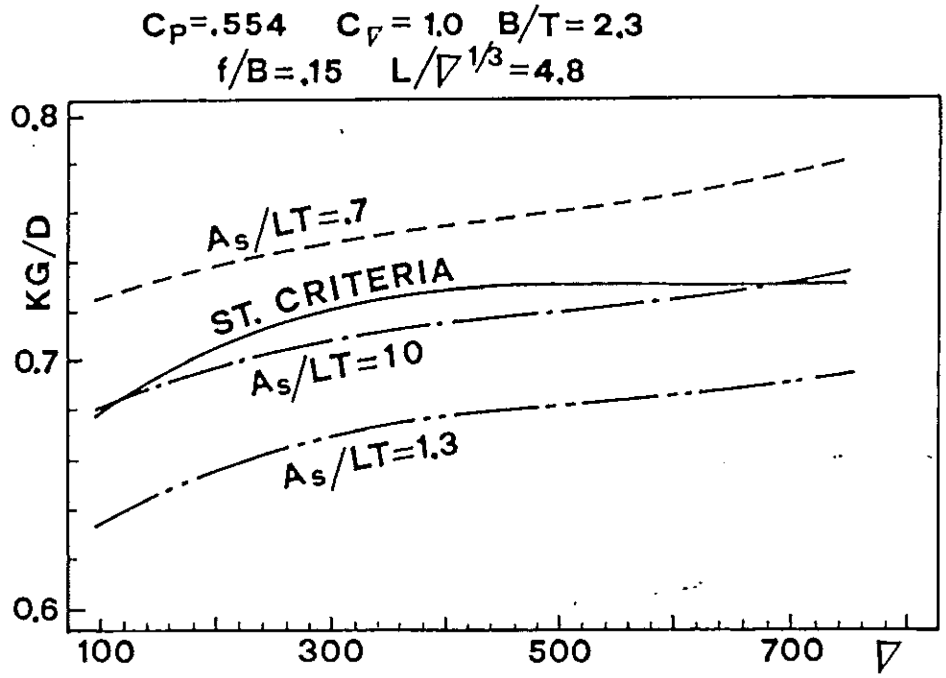
$$\frac{h_s}{T} = K\left(\frac{A_s}{LT}\right) = .70\left(\frac{A_s}{LT}\right)$$

This constant mean value has been obtained from the analysis of a sample of operating fishing vessels. In pàrticular the following considerations can be deduced:

a) the figures 1 and 2 show that KG*/D increases as ∇ increases; moreover the statistical criteria for $C_\nabla = 1$, $B/T = 2.3$ and $f/B = .15$ are almost equivalent to the weather one being A_s/LT equal to unity. Taking into account that the ratio A_s/LT is generally greater than unity for small ships and less than unity for big ships, it follows that the weather criterion is the most severe one for small ships, while for big ships the opposite applies.

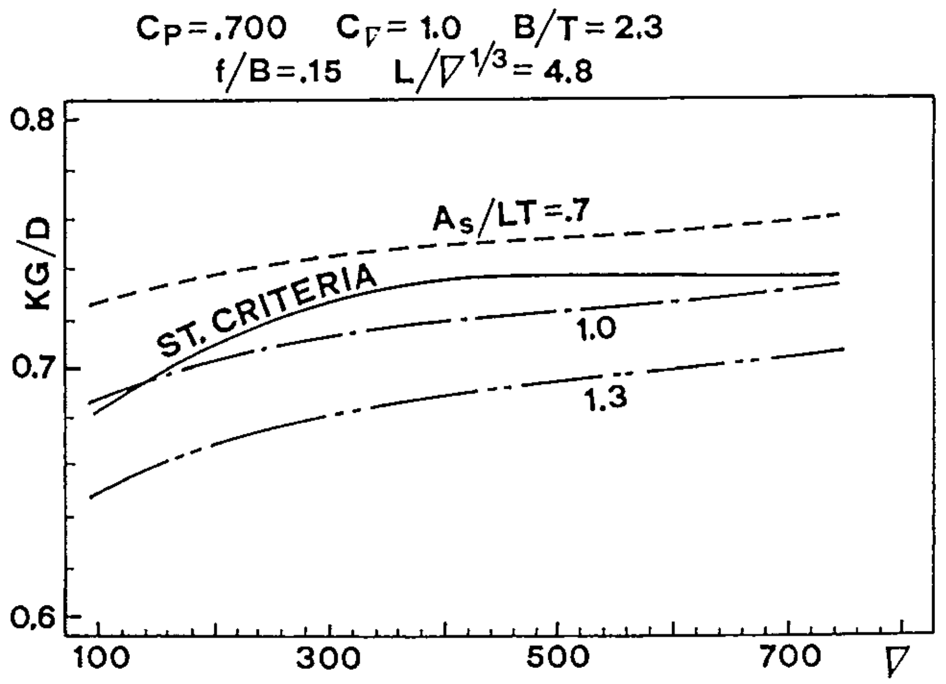
b) the figures 3 and 4 show that as C_∇ decreases, the KG*/D relevant to the statistical indices increases, while the opposite trend can be observed for the KG*/D curves relevant to the weather criterion; taking into account that for a given hull, the actual KG*/D value at light displacement is generally higher than at full load condition, the weather criterion is far the most severe one for low values of C_∇.

c) the influence of B/T on the stability is very remarkable (figs. 5-6). The KG*/D curves grow markedly with B/T; again it can be noted that the statistical indices are almost equivalent to the weather one being A_s/LT equal to unity, and that the φ_x index is the most severe one for very high values of B/T (say >3.0).



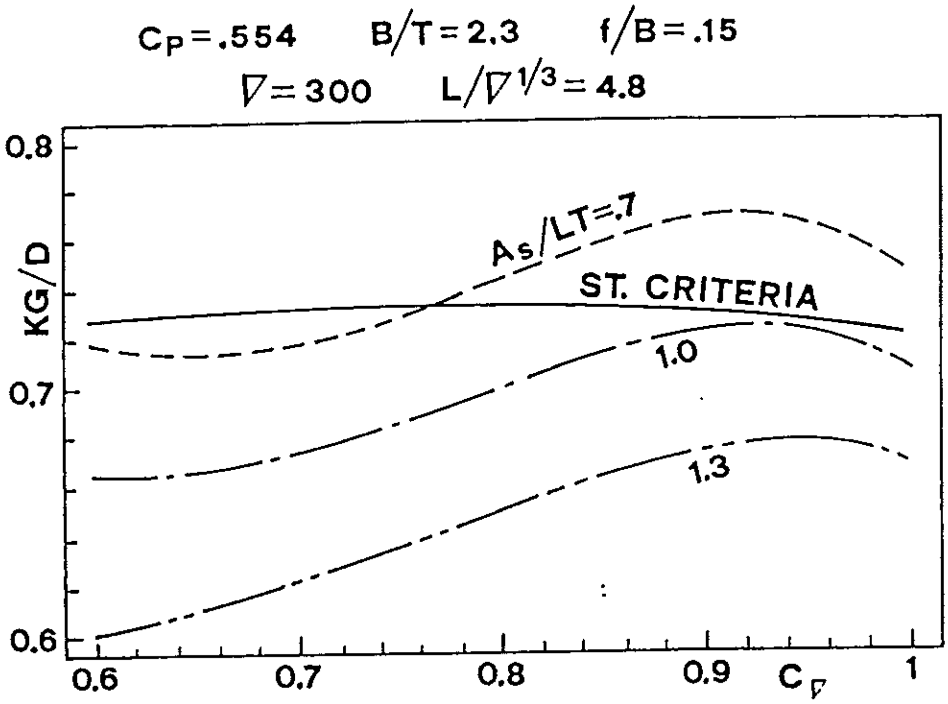
Cp=.554 C∇= 1.0 B/T= 2.3
f/B=.15 L/∇ 1/3=4.8
0.8
0.7
0.6
KG/D
As/LT=.7
ST. CRITERIA
As/LT=10
As/LT=1.3
100
300
500
700
∇

FIG. 1



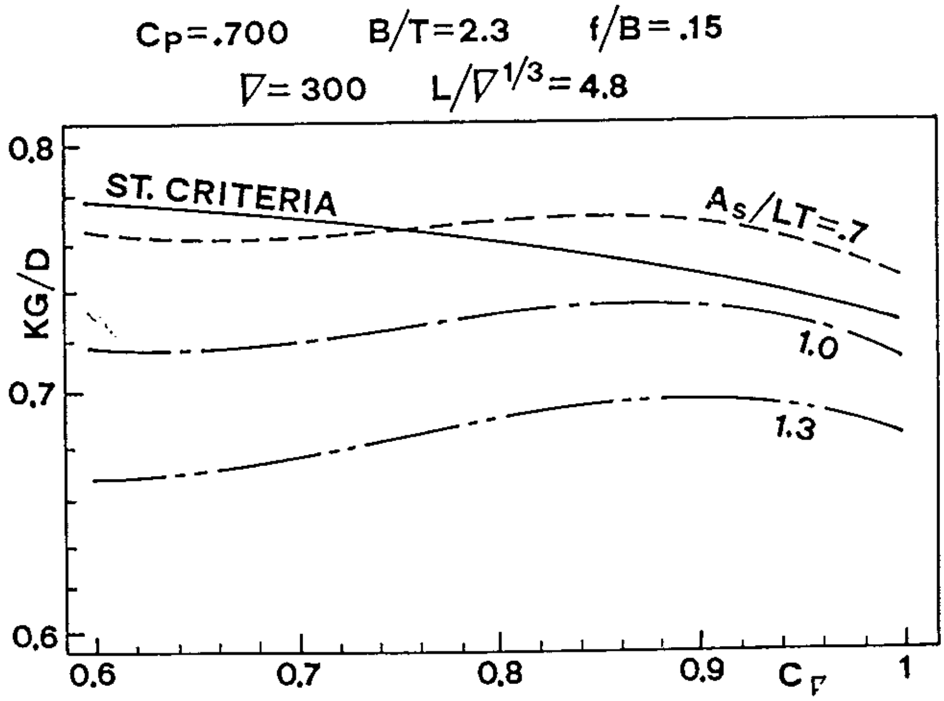
Cp=.700 C∇= 1.0 B/T= 2.3
f/B=.15 L/∇ 1/3= 4.8
0.8
0.7
0.6
KG/D
As/LT=.7
ST. CRITERIA
1.0
1.3
100
300
500
700
∇

FIG. 2



$C_P = .554$ $B/T = 2.3$ $f/B = .15$
$\nabla = 300$ $L/\nabla^{1/3} = 4.8$
KG/D
0.8
0.7
0.6
$A_s/LT = .7$
ST. CRITERIA
1.0
1.3
0.6
0.7
0.8
0.9
C_∇
1

FIG. 3



$C_P = .700$ $B/T = 2.3$ $f/B = .15$
$\nabla = 300$ $L/\nabla^{1/3} = 4.8$
KG/D
0.8
0.7
0.6
ST. CRITERIA
$A_s/LT = .7$
1.0
1.3
0.6
0.7
0.8
0.9
C_∇
1

FIG. 4

$C_P = .554$ $C_\nabla = 1.0$ $\nabla = 300$

$f/B = .11$ $L/\nabla^{1/3} = 4.8$

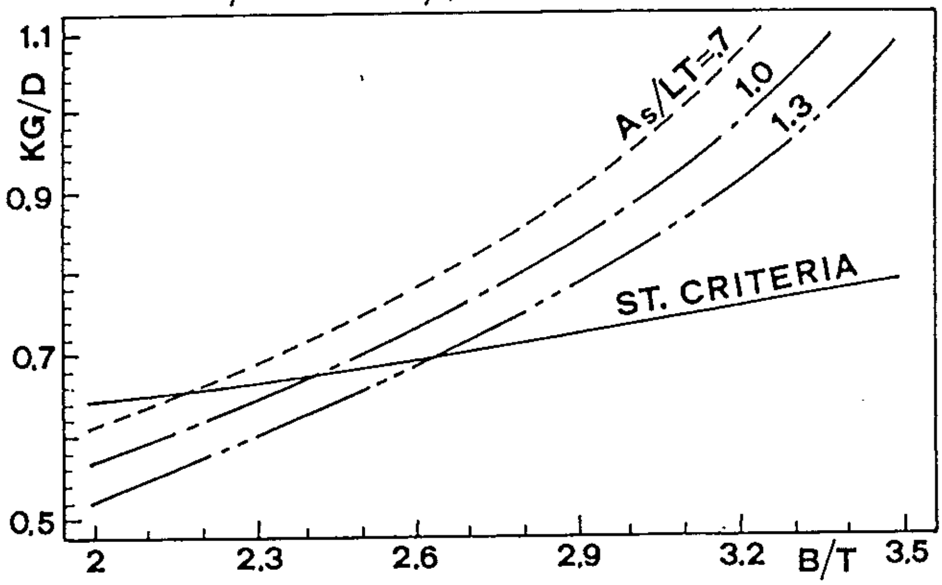


FIG. 5

$C_P = .700$ $C_\nabla = 1.0$ $\nabla = 300$

$f/B = .11$ $L/\nabla^{1/3} = 4.8$

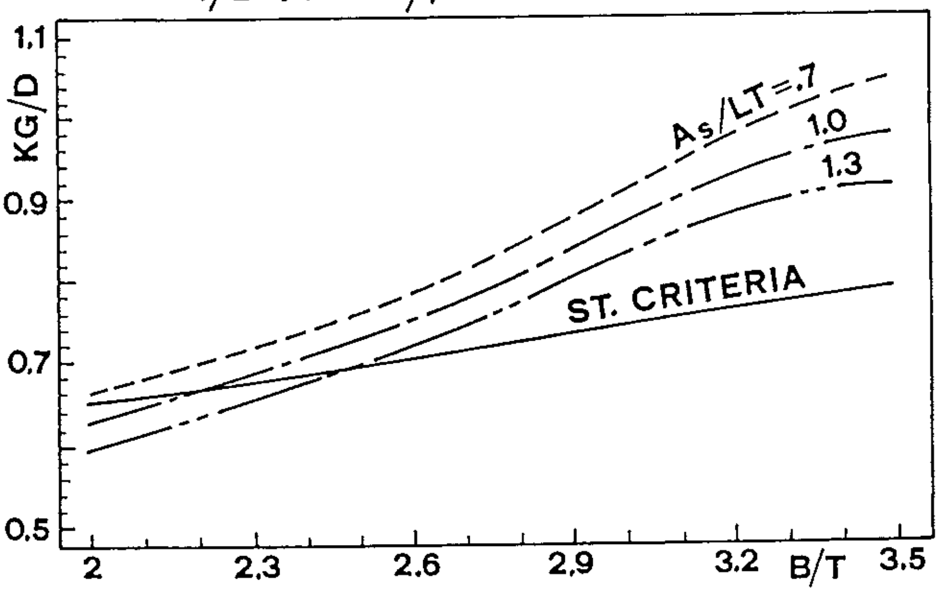


FIG. 6

d) the influence of f/B is similar for all indices (figs. 7-8): KG*/D at first increases, then it tends to decrease with f/B increasing. However, this trend is more marked for the statistical indices, because being f/B less than ≅.15at full loading condition the most severe statistical index is φ_x, which, at low f/B (say less than .1) is also more severe than the weather one. This is due to the fact that the well known favourable effect on dynamic stability of an increased freeboard is partly balanced by the contemporary raising of the centre of gravity and of the windage area, while on the other hand, the angle φ_x corresponding to the maximum righting arm is very sensitive to the freeboard value.

e) C_p has little influence on the stability indices except for low C_∇, where higher values of C_p are favourable;

f) the KG*/D values according to the weather criterion obviously decrease with A_S/LT increasing.

COMPARISON BETWEEN DIFFERENT STABILITY INDICES AND CRITICAL ANALYSIS OF I.M.O. CRITERIA

The KG^*/D calculated values for the different stability indices as a function of C_∇, C_p, B/T, f/B, $L/\nabla^{1/3}$ and ∇_0 show the range where each one of the different indices prevails; in other words that one for which KG^*/D is lower.

Making a comparison between these ranges (table II) the following considerations can be deduced, explained by the influence of f/B, B/T and C_∇ on the righting arm curve, considerations strictly true for Ridgely-Nevitt fishing vessels series hulls, but generally true for all fishing vessels hulls:

- the GM standard value, if B/T > 2.0, is always less severe with respect to the other indices. It can be important only for B/T < 2.0 because of lowering of the metacentre for narrow beam ships;
- E_{30} and E_{40} standard values are not influent for B/T > 2.5, but only within the following narrow bounded ranges:

$$E_{30} \text{ with } \begin{cases} B/T = 2.00 \\ \\ B/T = 2.50 \end{cases} \qquad f/B > .15 \qquad C_\nabla = .60 \div .90$$

$$E_{40} \text{ with } \begin{cases} B/T = 2.00 & \begin{cases} f/B < .10 & C_\nabla = .60 \div .70 \\ f/B > .15 & C_\nabla = .80 \div 1.0 \end{cases} \\ \\ B/T = 2.50 & \begin{cases} f/B < .15 & C_\nabla = .70 \\ f/B > .20 & C_\nabla = .90 \end{cases} \end{cases}$$

- GZ_{30} and ΔE indices affect enough wide ranges, but for B/T < 3.0 and C_∇ values increasing with B/T and f/B increasing;

- the suggested standard φ_x value is the most severe one over a wide range of C_∇. This range is of increasing width as for B/T=const and decreasing f/B, as for f/B=const and increasing B/T. Particularly, for B/T = 3.5, φ_x is the most severe index for any f/B and C_∇ values. This is understandable bearing in mind the influence of deck flooding angle on the righting arm curve, which depends only on f/B and C_∇.

TABLE II

B/T = 2

	f/B	.05				.10				.15				.20			
	∇ m³	100	250	500	750	100	250	500	750	100	250	500	750	100	250	500	750
C_∇	φ_x	.9+1.1	.8+1.1	.7+1.1	.6+1.1	1+1.1	.9+1.1	.9+1.1	.9+1.1	-	1.1	1.1	1+1.1	-	-	-	-
	GZ_{30}	.6+.9	.6+.7	-	-	.8+.9	.8+.9	.8	-	.9+1.1	.9+1	1	-	1.1	1.1	1.1	1.1
	ΔE	-	.6+.7	.6+.7	.6+.7	-	-	.8	.7+.8	-	-	1	-	-	-	-	1.1
	E_{30}	-	-	-	-	-	-	-	-	.6+.8	.6+.7	.6+.7	.6+.7	.6+.9	.6+.9	.6+.9	.6+.9
	E_{40}	-	.6	.6	-	.6+.7	.6+.7	.7	.7	.9	.8+.9	.8+.9	.8+.9	1.1	1+1.1	1+1.1	1+1.1
	GM	-	-	-	-	.6	-	.6+.7	.6+.7	.6	.6	.6+.7	-	-	-	-	-

B/T = 2.5

	f/B	.05				.10				.15				.20			
	∇ m³	100	250	500	750	100	250	500	750	100	250	500	750	100	250	500	750
C_∇	φ_x	.6+1.1	.6+1.1	.6+1.1	.6+1.1	.9+1.1	.8+1.1	.8+1.1	.8+1.1	1.1	1+1.1	1+1.1	1+1.1	-	-	1.1	1.1
	GZ_{30}	-	-	-	-	-	-	-	-	.7+1.1	.7+.8	.8	-	1+1.1	1+1.1	1	1
	ΔE	-	-	-	-	.7+.8	.6+.8	.6+.7	.6+.7	-	.9+1	.9+1	.8+.9	-	1.1	1+1.1	1+1.1
	E_{30}	-	-	-	-	-	-	-	-	.6+.7	.6	.6	.6	.6+.9	.6+.9	.6+.9	.6+.9
	E_{40}	-	-	-	-	-	-	-	-	-	.7	.7	.7	-	.9	.9	.9

B/T = 3

	f/B	.05				.10				.15				.20			
	∇ m³	100	250	500	750	100	250	500	750	100	250	500	750	100	250	500	750
C_∇	φ_x	.6+1.1	.6+1.1	.6+1.1	.6+1.1	.7+1.1	.6+1.1	.6+1.1	.6+1.1	1+1.1	1+1.1	1+1.1	1+1.1	1.1	1.1	1.1	1.1
	GZ_{30}	-	-	-	-	-	-	-	-	.6+.9	.6	-	-	.6+1.1	.6+1	.6+.7	.6+.7
	ΔE	-	-	-	-	.6+.7	.6	-	-	.8+1	.7+.9	.6+.9	.6+.9	1.1	.9+1	.8+1	.8+1

B/T = 3.5

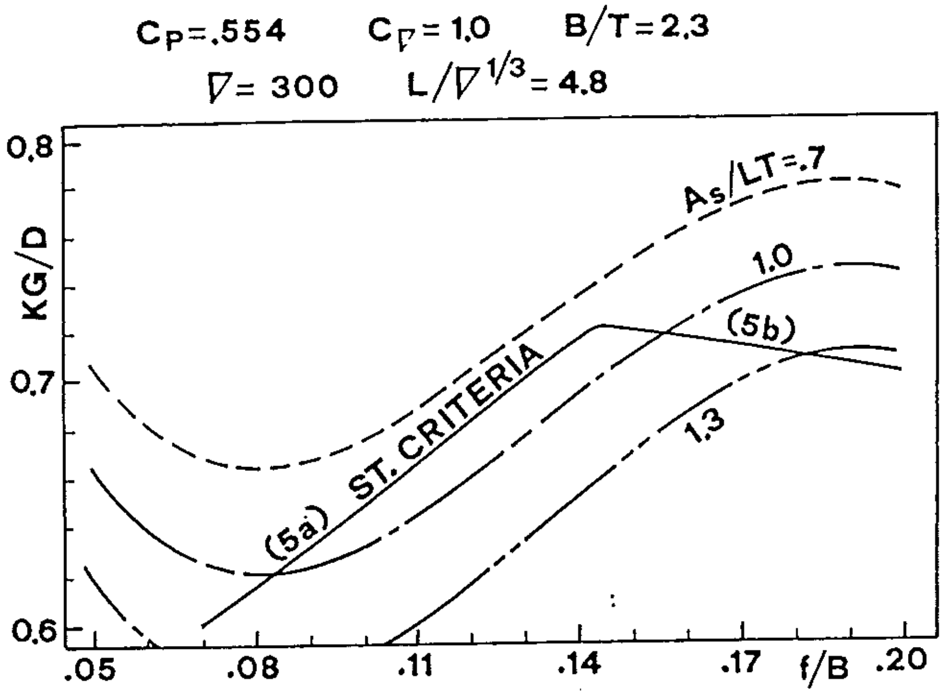
	f/B	.05				.10				.15				.20			
	∇ m³	100	250	500	750	100	250	500	750	100	250	500	750	100	250	500	750
C_∇	φ_x	.6+1.1	.6+1.1	.6+1.1	.6+1.1	.7+1.1	.7+1.1	.7+1.1	.7+1.1	1.1	1+1.1	1+1.1	1+1.1	1.1	1.1	1.1	1.1

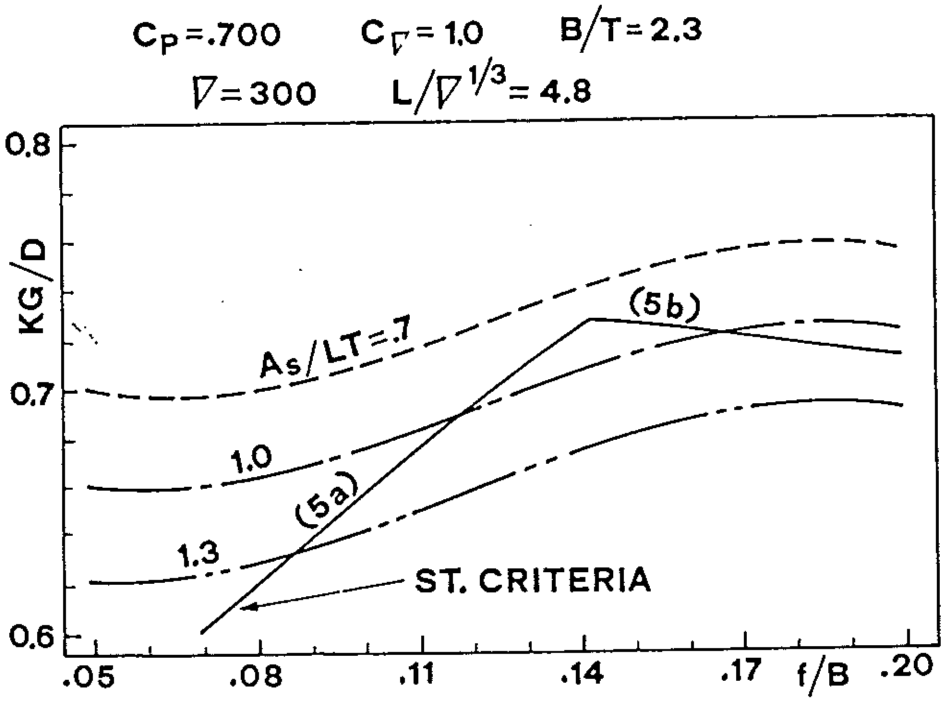
For $C_\nabla = 1$, the deck flooding angle φ_d is given by the relation: $\varphi_d = \arctan(2\ f/B)$

The index φ_x is generally small for low values of f/B and for high values of C_∇. Hence the need to lower the ship centre of gravity in order to satisfy the standard value φ_x^*.

From the above mentioned considerations it can be concluded that the most severe statistical indices for nearly all geometrical parameters of the hulls and for any loading conditions are φ_x, $(GZ)_{30}$ and ΔE.

By looking at the figures 9 and 10, it is very important to highlight the following further consideration:


Cp=.554
C∇=1.0
B/T=2.3
∇=300
L/∇^1/3=4.8
KG/D
0.8
0.7
0.6
As/LT=.7
1.0
1.3
(5a)
(5b)
ST. CRITERIA
.05
.08
.11
.14
.17
f/B
.20
FIG. 7


Cp=.700
C∇=1.0
B/T=2.3
∇=300
L/∇^1/3=4.8
KG/D
0.8
0.7
0.6
As/LT=.7
1.0
1.3
(5a)
(5b)
ST. CRITERIA
.05
.08
.11
.14
.17
f/B
.20
FIG. 8

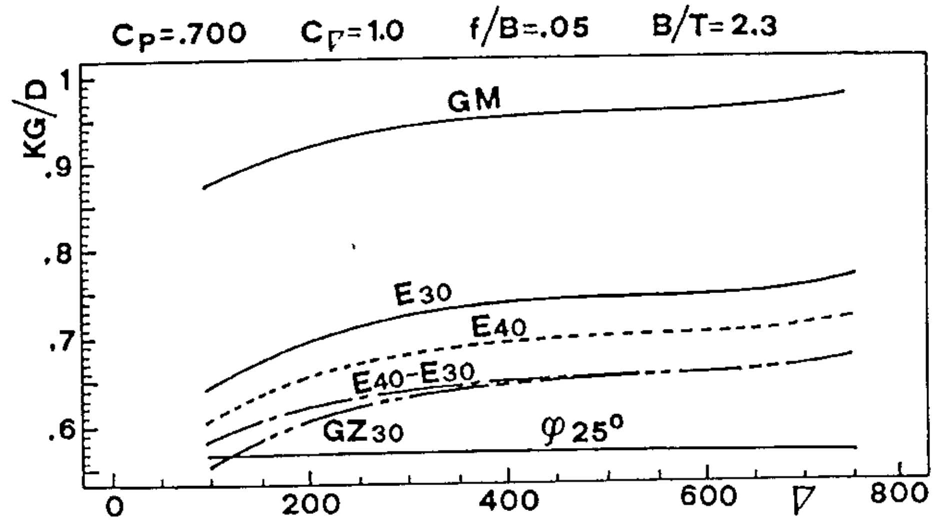


FIG. 9

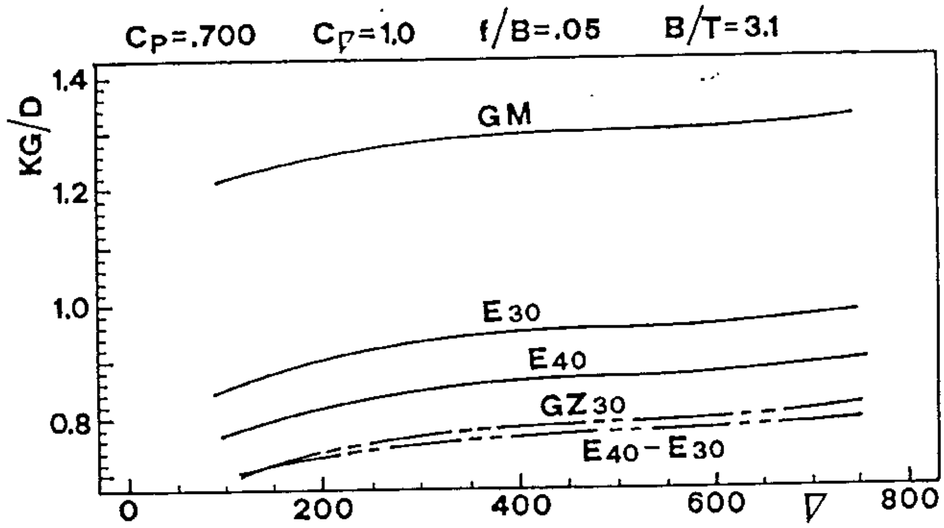


FIG. 10

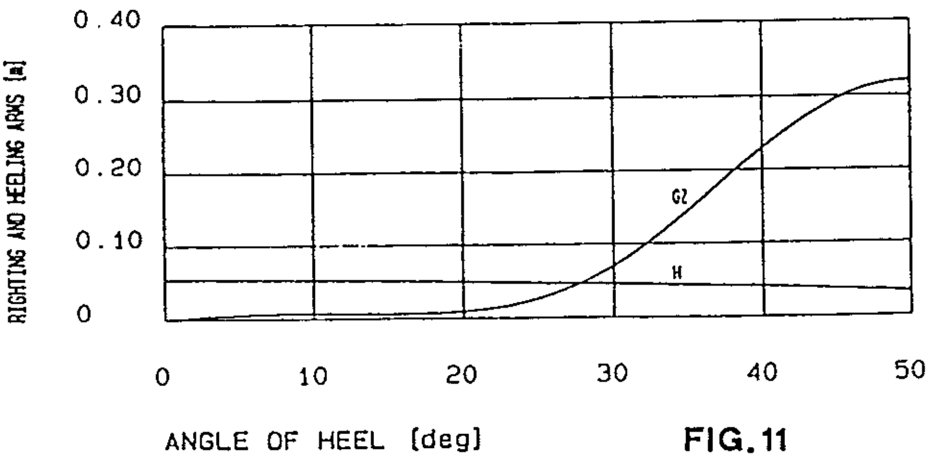


FIG. 11

- the GM, in order to verify all the stability indices, can assume excessive values with very low rolling period of the ship and the consequently very bad seakeeping characteristics and operational conditions on board, as shown by table III. The approximate values of the rolling period T and of the Kenpf number T_k reported in this table are obtained for two fishing vessels and are those ones necessary to verify the φ_x and ΔE stability indices, if the inertia radius ρ is given as the following percentage of the ship's beam: $\rho = c\ B \cong .38\ B$

TABLE III

SHIP	L_{bp}(m)	B(m)	T(m)	D(m)	C_p	B/T	f/B	stability index	KG/D	GM(m)	T(sec)	T_k
1	29.6	8.50	3.90	4.325	.65	2.3	.05	GM	.96	.35	10.9	11.7
								$\varphi_{25°}$	.57	2.035	4.53	4.8
								ΔE	.66	1.646	5.02	5.4
2	30.0	10.85	3.50	4.05	.70	3.1	.05	GM	1.37	.35	13.81	13.25
								$\varphi_{25°}$	.72	2.98	4.77	4.54
								ΔE	.94	2.10	5.69	5.4

SOME ASPECTS OF SHIP SAFETY DUE TO THE LOADING CONDITIONS

The I.M.O. stability criteria, as it is well known, should be complied with all the loading conditions.

A better understanding of the influence on the safety of the ships, and particularly on the ro-ro ships, of the loading conditions different from those ones required by I.M.O. criteria can be pointed out from a recent disaster of a ro-ro ship due to loading and ballasting errors, as shown from the conclusions of the Inquiry Committee.

In the spring of 1990 a ro-ro cargo ship, during a turning manoeuvre in still water and already in sight of the port of destination heeled heavily and rapidly capsized and sank with the loss of thirteen human lives.

The ship had a service speed of 15.5 kn. and the following main dimensions:

$L_{OA} = 112.84$ m $B = 18.67$ m $D = 13.5$ m $T = 6.34$ m $\Delta = 8314$ t

with three decks for trailers and one deck for cars.

The Inquiry Committee had the opportunity to ascertain the causes of the disaster.

In order to meet the I.M.O. stability criteria with the three uniformly loaded decks to transport the trailers, the Loading Manual imposed the carriage of water ballast and considered a theoretical total allowable weight of trailers of 1909 tonnes with trailers, having each a mean gross weight of 23 tonnes.

The paying cargo at the moment of the disaster was almost uniformly distributed on the three deck, but its global weight was about 3432 tonnes being the mean weight of each trailer 41.4 tonnes.

The mean overload of each vehicle was almost 80% in comparison with the weight of 23 tonnes, but only about 18% if reference is made to the usual 35 tonnes allowed for normal road haulage.

The water ballast was 109 tonnes and mean draught was 6.23 metres (i.e. with a margin of 0.11 metres to the freeboard mark, while this margin had been 0.08 metres at departure).

In this condition the check of stability based on I.M.O. criteria indicates extremely dangerous situation for the incautious loading of the ship, as shown by the righting lever curve (Fig. 11) and by the following consequently values of the stability indices:

$E_{30} = 0.008$ m rad $E_{40} = 0.033$ m rad $\Delta E = 0.025$ m rad

$$(GZ)_{30} = 0.07 \text{ m} \qquad GM = 0.09 \text{ m} \qquad \varphi_x = 50°$$

When the ship, in sight of the destination port, began to turn at her service speed of about 15.5 kn. the static equilibrium angle was almost 28° and the dynamic equilibrium angle was about 40°. In consequence of the very large first outboard inclination the ship began to flood (through the ramp gate on the weather deck imprudently left open) and in few minutes capsized and sank.

CONCLUSIONS

In this paper it has been shown that the ship's intact stability can be hardly conditioned by the design parameters and by the loading conditions.

When the ship form is obtained from a standard series by using the geometrical similarity law and afterwards the regression analysis we can obtain analytical relations in order to determine the maximum allowable vertical position KG according to the various I.M.O. stability indices.

These relations allow us to carry out a systematic analysis both of the influence of the loading conditions and of the geometrical parameters on the I.M.O. stability indices.

Therefore by means of these equations it is possible to analyse the influence on the stability of the design parameters and to compare the relative severity of the I.M.O. indices.

It is also possible to determine the limiting ship proportions to fulfil the stability requirements for predictable operational conditions.

In the present paper an application of the proposed methodology has been performed on the Ridgely-Nevitt fishing vessels series and from the results obtained we can deduce the following considerations:

- the check of minimum value of the initial GM metacentric height, that is the most frequent and often the only check performed for the small fishing vessels, is not significant;
- some indices ($(GZ)_{30}$, ΔE, φ_x) are always more severe than others and therefore these three indices could be taken in account in the statistical criterion for fishing vessels only, being very little significant the other indices;
- the GM value, in order to verify the most severe indices, could be excessive. Consequently it is necessary to fix a maximum allowable value of GM in order to avoid small rolling period, bad seakeeping characteristics and hard or impossible operational conditions on board.

From the analysis of the ro-ro cargo ship disaster we can deduce the following further considerations:

- the Loading Manual of the ship should always consider all the possible realistic and acceptable conditions and should foresee a safety margin in order to enable eventual loading or ballasting errors;
- the Loading Manual should be used by Officers who have a throughout knowledge of stability problems and can therefore adequately evaluate safety margins in any situation. To this aim could be necessary a better training of the Officers.

However the shipment of haulage units whose weights are unknown should always be forbidden, since reliable data are a prerequisite for any serious stability check.

Particularly ro-ro traffic should be allowed only if the ports are provided with efficient weight-houses or if the ships are equipped their own weighting machines.

ACKNOWLEDGEMENTS

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Paper for STAB94

Rational Examination of Stability Criteria in the Light of Capsizing Probability

by
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National Research Institute of Fisheries Engineering, Japan
and
Y.Ikeda
University of Osaka Prefecture

ABSTRACT

This paper describes a new procedure to examine stability criteria by using capsizing probability. The capsizing probability is calculated by the methods that were proposed by one of the authors, and was validated with model and numerical experiments. By using these methods, the risk levels that are guaranteed by stability criteria are identified in the form of probability. Then, comparing these risk levels, we can judge whether the stability criteria are effective or not. In some cases a guideline to improve criteria can be obtained. As applications of this procedure, the IMO weather criteria, a weather criterion for a new ship type and a design criterion for a ship in following seas are examined.

INTRODUCTION

It is one of the most important role for naval architecture to prevent capsize of ships in a seaway, and for this reason the International Maritime Organization (IMO) / governments provide international / national stability criteria. However, the rate of capsizing accidents are still high, although so many papers on ship stability have been published. The most presumable reason why results from stability research cannot prevent capsizing of actual ships is that a suitable methodology has not yet developed to transfer recent research fruits to stability criteria.

Before World War 2, statistical comparison between hydrostatic results and capsizing casualties was an only way to establish stability criteria. The typical examples of these empirical criteria are the IMO resolutions A.167 and A.168. These empirical criteria are not rationally applicable for a ship that is different from sampled ships.

After World War 2, a more physical approach became available with significant progress on ship dynamics. The weather criteria was developed by this approach, and authorized by the Japanese government as one of the stability criteria in 1956.(Watanabe et al.,1956) Later, the IMO also recommended the weather criteria as the Resolutions A.562 and A.685 in 1985 and 1991, respectively.

These criteria are based on physical phenomena but were adjusted with capsizing casualties in the form of the wind velocity. In other words, the wind velocity in the weather criteria does not represent the actual sea state and has rather empirical meanings. Since the weather criteria involve such an empirical factor, it is not easy to improve the criteria. That is, even if an excellent prediction method for roll is developed, the method cannot be adopted in the existing criteria without adjusting with capsizing casualties. Obviously it is quite troublesome or often impossible to find old casualties or collect new casualties.

Therefore, more physics-oriented approach is desirable for future stability criteria. The ultimate outcome of the physics-oriented approach is the probability of capsizing of a ship during its whole life. Because, wind and waves, which cause ship capsizing, are random events. In classical approaches this probabilistic nature of physical phenomena is involved in empirical factors. Thus, if we directly consider the capsizing probability for establishing stability criteria, the criteria become much more physics-oriented, in other words, more applicable for more ships beyond the limitation of sampled hull forms and casualties. The concept of capsizing probability approach itself is not new. It was firstly proposed probably by Firsev and then by Sevastyanov and Kobylinski. (Kobylinski, 1975) However, this concept has not yet used to establish stability criteria. (Kobylinski, 1990) Because, it is impossible to accurately calculate probability of all types of capsizing during a whole life.

To overcome this difficulty, this paper proposes a new procedure to use capsizing probability as a relative measures for establishing criteria. In the light of capsizing probability, stability criteria are examined and then improved on. The basic idea of this procedure was briefly mentioned by one of the authors. (Umeda et al., 1990) This paper also includes a review of our calculation methods for capsizing probability and some examples on examining stability criteria.

PROCEDURE

The use of the capsizing probability during the whole life of a ship requires calculation methods for all types of capsizing, reliable statistics of sea state and the operation practice. Since these three requirements are not available, the authors propose to calculate a short term probability of some dominant modes of capsizing instead of the capsizing probability during a whole life. The short term probability means the probability in a stationary state of sea, and does not require statistics of sea state and operational practice. In addition, it is convenient to examine the IMO weather criteria because the weather criteria are formulated also under a certain state of sea.

So far, several modes of capsizing have been identified. For example, capsizing in beam wind and waves, pure loss of stability in following seas, broaching, low cycle resonance, water trapped on deck, breaking waves, icing, fishing gear effects and so on. If some dominant modes of capsizing are prevented, the total number of capsizing can be greatly reduced.

For some of the capsizing modes, calculating methods for the capsizing probability are available and were reasonably validated with model experiments or numerical experiments. Since the accuracy of these calculating methods is not perfect, it is not desirable to use calculated results as absolute measures, which one can compare with the absolute risk level. However, the methods are still accurate enough to be used as relative measures, with which one can examine relationships between several criteria.

Nowadays, it is important for the IMO activities to establish stability criteria in following seas and establishing stability criteria for a new ship type. To do so, it is necessary to harmonize new established criteria with existing criteria, such as the weather criteria. Therefore, using the capsizing probability, the following procedure is proposed.

(1) many sampled ships are provided.

(2) Each ship is loaded to just meet the criterion that we intended to examine.

(3) For several stationary sea state, the capsizing probability is calculated, and assumed as the risk level that the criterion permits.

(4) Keeping the risk level of the criterion, the criterion is extrapolated for a new ship type or extended for other capsizing modes.

CAPSIZING IN BEAM SEAS

It is essential to calculate the probability of capsizing in beam winds and waves. Because, the IMO weather criteria were developed for capsizing in beam wind and waves. Pioneering work for this problem had been carried out by Caldwell and Yang (1986). A more realistic method was presented by the authors (Umeda, et al., 1992), validated with model experiments (Umeda, et al., 1993). Recently a rigorous method is obtained by Belenky (1993).

Here the method by the authors is reviewed briefly. It is based on the framework of the weather criteria but more rigorously formulated. The probability density of ship roll motion is calculated by an equivalent linearized method in winds and waves, which are considered as random processes. On the other hand, the safe domain on a phase plane of roll, shown in Fig.1, is determined making use of a nonlinear roll equation without time dependent external moments. Then, integrating the probability density of roll motion on the safe domain, the non-capsizing probability can be obtained. As can be seen in Fig.2, this method was well validated with model experiments of several ships in irregular beam seas.(Umeda, et al., 1993 and Fujiwara, et al., 1994) The assumption that the safe domain can be determined without time dependent external moments is in common with the weather criteria, and was theoretically validated by Belenky (1993). But, if we consider a time dependent external moments for the safe domain, chaotic

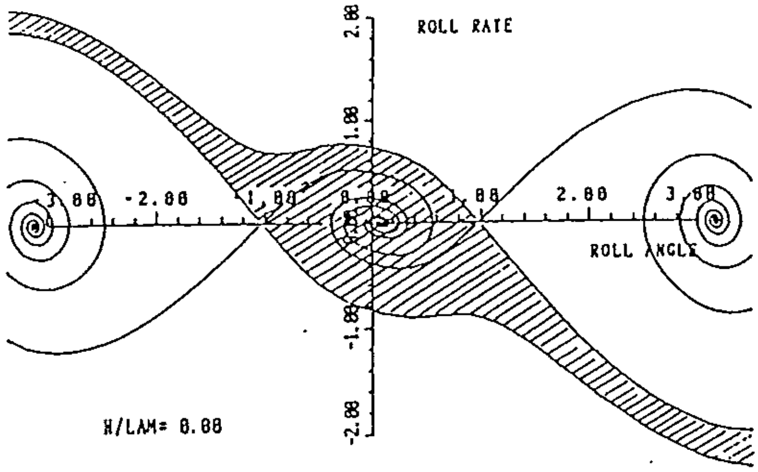


Fig.1 An example of Safe domain for a coastal trawler. (Umeda & Yamakoshi, 1992)

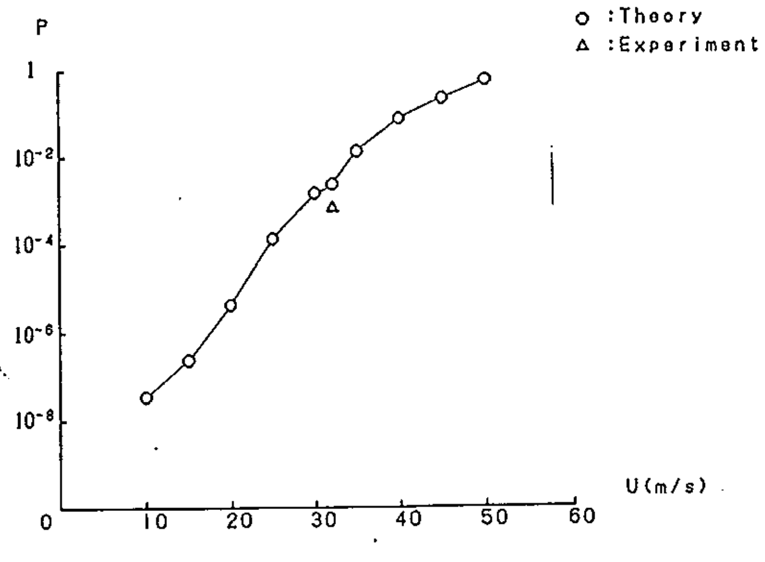


Fig.2 Probability of capsizing when a craft in beam seas per roll cycle. (Umeda, et al., 1993)

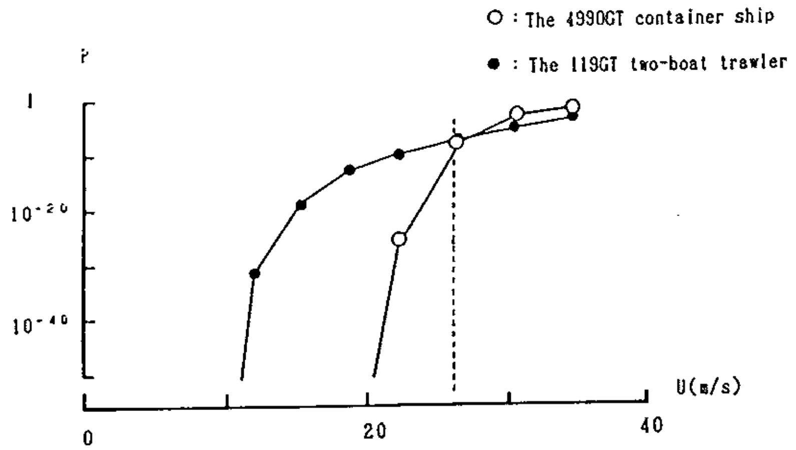


Fig.3 Probability of capsizing when ships, critically satisfying the IMO weather criteria, in beam seas per roll cycle. (Umeda, et al., 1993)

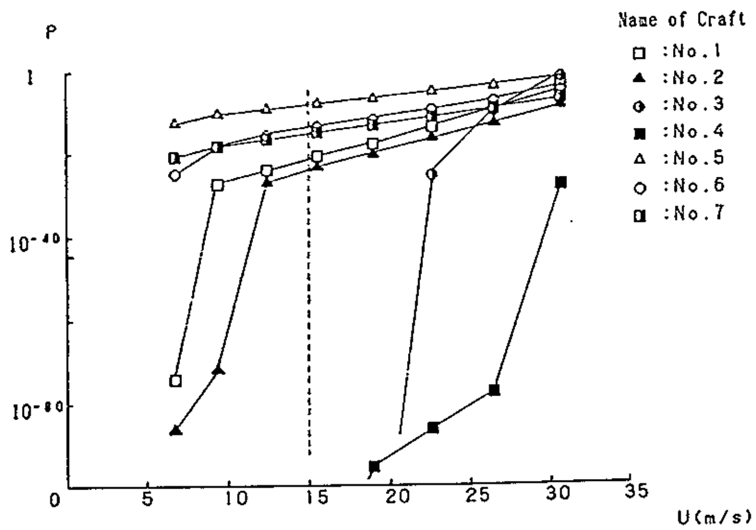


Fig.4 Probability of capsizing when craft, critically satisfying the Japanese weather criterion, in beam seas per roll cycle. (Umeda, et al., 1993)

behaviours can be observed in the roll motion (Thompson, 1991).

To examine the weather criteria, this calculation method was applied to some displacement-type ships that are loaded so as to just meet the weather criteria. As can be seen in Fig.3, the averaged wind velocity, U, represents sea states and the capsizing probability per one roll is obtained for each sea state. Since the weather criteria require that the averaged wind velocity is 26 m/sec, the risk levels permitted by the weather criteria are 10^{-9} for the container ship and 10^{-7} for the trawler.(Umeda, et al., 1992) These risk levels seem to be reasonable because 10^{-8} means that one capsizing is expected for about twenty years operation under a stationary state of sea.

To develop stability criteria for a semi-displacement craft, the present procedure is quite effective. Now Japanese government applied a weather criterion for some of semi-displacement craft. However, it is often not easy for these craft to obey this criterion, because the weather criterion was originally established from experience of displacement-type passenger ships in Japan. Numerical results of the capsizing probability are shown in Fig.4 for seven semi-displacement craft, loaded so as to just meet the Japanese weather criterion. Since the Japanese weather criterion requires that the wind velocity is 15 m/sec, the risk levels of the criterion for these craft are 10^{-9} to 10^{-218}. Except for No.5 craft, the Japanese criterion seems to be too stringent for these craft, because the risk level of the craft is much smaller than that of displacement-type ships.

CAPSIZING IN FOLLOWING AND QUARTERING SEAS

In following and quartering seas it is identified by model experiments that pure loss of stability and broaching are two major modes of capsizing. (Kan et al., 1993) Thus, a criterion should be established to make the risk level for each phenomena as small as the risk level of the IMO weather criteria for beam sea condition.

For the pure loss of stability, Krappinger(1970) firstly attempted to calculate capsizing probability and later one of the author have already proposed a method for calculating the probability of capsizing and validated with numerical experiments. (Umeda, et al., 1990 and Umeda & Yamakoshi, 1992) Other attempt was made by Vermeer(1990) and the Netherland(1991) to develop stability criteria by regression analyses with a empirical factor. In the method of the author, it is assumed that a ship capsizes when it meets a steep wave crest with large roll energy. Integrating joint probability density of GZ reduction, roll angle and roll rate on the safe domain that is determined by nonlinear roll equation with a GZ curve reduced by a wave crest, the non-capsizing probability can be calculated. As can be seen in Fig.5, this method was reasonably validated with numerical experiments. (Umeda & Yamakoshi, 1992) Here χ and F_n indicate wave direction from stern and the Froude number, respectively.

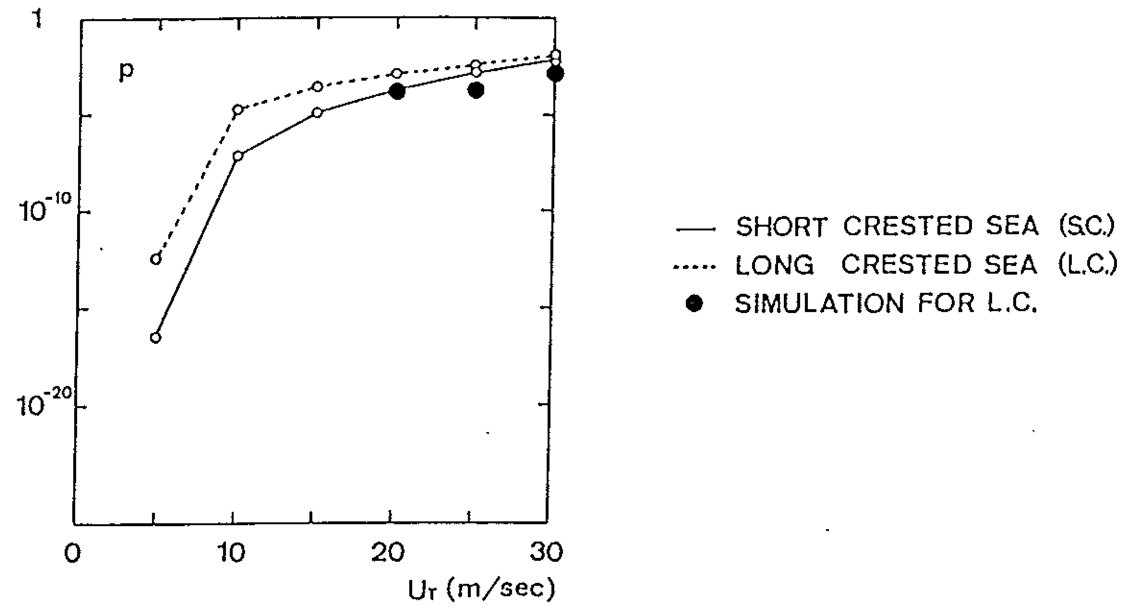
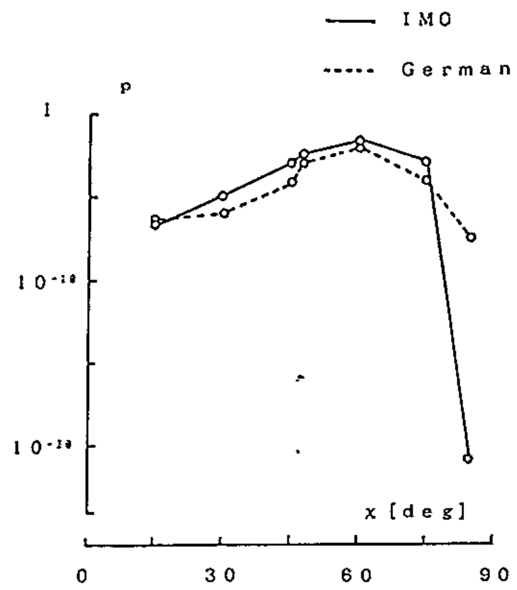


Fig.5 Probability of capsizing when a coastal trawler meets a zero cross wave with χ=30°and Fn=0.3.(Umeda & Yamakoshi, 1992)



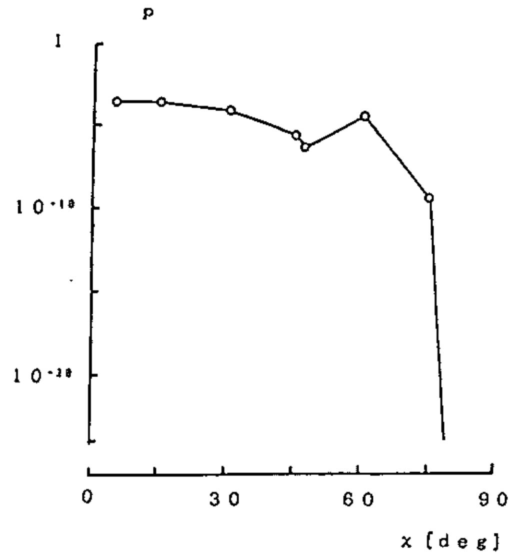


Fig.6 Probability of capsizing when a container ship meets a zero cross wave under the Beaufort scale No.6 where U=12.35m/sec, $H_{1/3}$=3m and Fn= 0.3.

Fig.7 Probability of capsizing when a trawler meets a zero cross wave under the Beaufort scale No.6 where U=12.35m/sec, $H_{1/3}$=3m and Fn=0.3.

Numerical results of capsizing probability are shown in Figs.6-7 for two displacement-type ships, namely, a container ship and trawler, that loaded so as to just meet the IMO weather criteria. The container ship is also loaded so as to just meet the German proposal of stability criterion in following seas. (Germany, 1990) As you see, when ships run in quartering seas, the capsizing probability becomes much larger than those in beam seas. As a result, the IMO weather criteria can permit too large capsizing probability. Furthermore, even the German proposal is not effective for quartering seas when the Beaufort scale is 6. These results demonstrate that rational design criteria for stability in quartering seas without any limitation of operational practices can be much more stringent for most of practical ships than the IMO weather criteria. Therefore, the operational guidance for avoiding dangerous situations will be a practical measure for capsizing in quartering seas.

For broaching, one of the author proposed a method for calculating the probability of surf-riding in place of that of broaching itself. Because, surf-riding is prerequisite for broaching and broaching is a more complex phenomenon than surf-riding. The details of the method have been already described at the stability conference. (Umeda, 1990) The method is useful for rationally establishing the operational guidance for avoiding surf-riding and broaching.

CONCLUSION

A practical procedure was proposed to rationally examine stability criteria in the light of capsizing probabilities. Some applications of this procedure showed that the IMO weather criteria are reasonable for conventional ships, while the Japanese weather criterion is more stringent than for semi-displacement craft. They also pointed out that a certain proposal to the IMO as a design criterion for a ship in following seas permits relatively large risk level.

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Nonlinear Dynamics of Floating Offshore Platforms

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School of Naval Architecture and Marine Engineering

As nearshore oil reserves are depleted, the search for oil is forced further offshore into deeper waters (D'Souza, 1994). In deepwater, the wave environment is much more severe and the capsizing risk dramatically increased. The current approaches to analyze the stability of floating offshore drilling units are for the most part derived from traditional ship stability criteria. However, due to their unique configuration, the dynamics of mobile offshore drilling units (MODU's) differ from that of mono-hull vessel hull forms. With this in mind, the recent American Bureau of Shipping Joint Industry MODU Stability Project (Shark, 1989) developed guidelines (ABS, 1990) based upon extensive comparisons to numerical simulations and model tests. Such a comparative approach may be inappropriate for predicting long-term survival behavior of certain unique offshore platforms configurations. An alternative to repeating these extensive numerical simulations is applied dynamical systems theory which can be used to more efficiently analyze the character of a vessel's extreme motions and predict critical (capsizing) behavior

Designing vessels which resist capsizing is the ultimate goal of all vessel stability analysis. However, vessel stability has long been analyzed using approximate static approaches; these approaches neglect the influence of external wave excitation and damping. The classical nonlinear vibration analysis approaches of perturbation methods and mathematical stability theory are also inadequate due to their restriction to weakly nonlinear systems and being difficult to apply. However, vessel capsizing is a large amplitude highly nonlinear transient (global) phenomenon requiring specialized analysis techniques. Over the last few years, dynamical systems techniques have been progressively developed and applied to various types of marine vehicles. These techniques are currently being extended to consider the large amplitude (nonlinear) dynamics of floating offshore platforms.

Motivation and Background

It is well understood that although the overall safety record of Mobile Offshore Drilling Units is substantially better than that of mono-hull vessels particularly the smaller vessels such as fishing vessels, MODU stability should still be studied. MODU stability should be studied for two reasons; first, there has been at least a few capsizings of damaged units (Johnson, 1983) and second, although MODU's have been previously used for exploratory drilling there has been limited application and increasing interest in utilizing MODU's for production drilling (D'Souza, 1994). As such these vessels have limited mobility and hence limited ability to avoid severe weather. However, knowing the predominate direction of severe storms and seas, it may be possible to position the vessel so as to minimize the motion. An example of this is the positioning of the Augur TLP so as to minimize the effect of Gulf of Mexico hurricanes (Schott, 1994).

Probably due to the wide variation in vessel configuration there have been extensive studies into MODU stability. Most notably and previously mentioned the American Bureau of Shipping's Joint Industry Stability Project (Shark, 1989). The ABS project resulted in a response based criterion (ABS, 1990) which serves as an alternative to traditional static righting arm versus wind heel criteria (ABS, 1991). As a result of this work, it seems as though much more extensive survival testing, analyses and prediction are undertaken during the design of such vessels (Schott, 1994). Although model tests or numerical simulations are quite general techniques of accurately predicting any vessel's performance in an extreme seaway, it is often quite difficult to determine critical behavior using either technique. What is proposed herein is an alternative which combines simplified global dynamical analyses with potentially more exact numerical simulations for verification. An additional step for further verification would be physical model testing if it were readily available.

Physical System Modelling

By considering Newton's laws of motion in a body fixed system (Figure 1a), the full nonlinear coupled equations of rigid body motion are obtained,

$$X = m[\dot{u}+qw-rv-x_G(q^2+r^2)+z_G(pr+\dot{q})]$$

$$Y = m[\dot{v}+ru-pw+x_G(pq+\dot{r})+z_G(qr-\dot{p})]$$

$$Z = m[\dot{w}+pv-qu+x_G(rp-\dot{q})-z_G(p^2+q^2)] \quad \textbf{(1)}$$

$$K = I_{44}\dot{p}-(I_{55}-I_{66})qr-I_{64}(\dot{r}+pq)-mz_G(\dot{v}+ru-pw)$$

$$M = I_{55}\dot{q}-(I_{66}-I_{44})rp-I_{64}(r^2-p^2)+mz_G(\dot{u}+qw-rv)-mx_G(\dot{w}+pv-qu)$$

$$N = I_{66}\dot{r}-(I_{44}-I_{55})pq-I_{64}(\dot{p}-qr)+mx_G(\dot{v}+ru-pw)$$

These equations are often called Euler's equation of motion and are derived in a variety of references, (see e.g. Abkowitz, 1969). Also associated with these equation is a set of angular kinematics called the Euler angle rotation sequence (Figure 1b). Although, various representations of the forces (X,Y,Z) and moments (K,M,N) on the right hand side of this system of equations are possible, for simplicity, this analysis follows the standard seakeeping assumptions and considers small amplitudes of motion in an ideal fluid. These normal forces and moments are then obtained by integrating the pressure over the body surface. First order terms proportional to unit body motion (displacement, velocity and acceleration) and incident wave amplitude are obtained. Forces proportional to unit body displacement are the hydrostatic forces **C**, forces proportional to unit body acceleration are called the added mass forces **A**, and forces proportional to unit body velocity are called damping forces **B**. The forces due to the incident wave are called wave exciting forces $\underline{F}(t)$. The matrix **M** is made up of the physical mass or inertias about the specific axis plus the inertial and coordinate coupling. The elements of the matrices A_{ij}, B_{ij}, and C_{ij} are the forces in the *i-th* mode of motion due to unit motion (displacement, velocity and acceleration) in the *j-th* direction. The mass matrix M_{ij} contains the appropriate mass, and inertial or coordinate coupling terms. The subscripts refer to the mode of motion and are: 1=surge, 2=sway, 3=heave, 4=roll, 5=pitch, 6=yaw. The appropriateness of either diffraction theory or Morrison inertial/viscous force representation is determined by the relative size of the cylinder and the wave length and height (see Figure 2).

In head seas, it may be possible to approximately study the pitch motion in isolation. This is so because there is no sway, roll or yaw excitation, there will in general not be any appreciable motion in these modes (see e.g., Falzarano, Holappa, and Taz Ul Mulk, 1993). In addition, since the longitudinal asymetry of the hull is small $xg \approx 0$. Moreover, one can either lower the coordinate system origin so that $zg \approx 0$, or restrain surge velocity and assume that the product term of pitch and heave velocity is small. Therefore, for the Poincaré map analysis, the pitch equation of motion is considered in isolation and is as follows,

$$(I_{55}+A_{55}(\omega))\ddot{\theta}+B_{55}(\omega)\dot{\theta}+\Delta GZ_{l}(\theta)=F_{5}\cos(\omega t+\gamma_{5}) \quad (2)$$

Details of the System

The resistance to capsizing of a typical mono-hull vessel is primarily a function of its roll restoring moment. However, the resistance to over-turning of a typical MODU must consider the vessel's tilt about an arbitrary orientation axis. A mono-hull over-turning about its pitch axis, i.e., "pitch-poling" is usually out of the question. However, a MODU's resistance to over-turning about its pitch axis may be almost as critical, as critical or even more critical than overtuning about its roll axis (i.e., capsizing). Typically the a typical MODU resistance to over-turning about an axis approximately half-way between the pitch and roll axis is most critical.

Although the typical MODU's resistance to over-turning an arbitrary tilt axis should eventually be studied, this work will concentrate on the vessels's dynamics about its pitch axis. We do this assuming that the vessel can be positioned so that it is headed into the predominate sea direction. We attempt to analyze the very large amplitude pitch motion in order to get a deeper understanding of these vessel's overall safety and to stress the fact that pitch over-turning may be as critical as roll for MODU's.

There are a number of characteristics of the MODU's pitch over-turning motion which are unique as contrasted to a mono-hull's (roll) capsizing and a mono-hull's pitch motion. First, MODU pitch is possibly asymmetric with regards to bow-down (+pitch) or bow-up (-pitch) rotations. This is due to the ship or torpedo shape of the MODU's lower hull. This is in contrast to the symmetric roll of ship's in the absence of an steady external wind moment. Second, the asymmetry of a MODU in pitch is far less important then that of a typical mono-hull. These characteristics are significant with regards to the critical pitch dynamics and the appropriateness of certain modelling approximations.

First, the asymmetry of pitch with regards to positive (bow down) or negative (bow up) pitch rotations suggests that MODU pitch motion behaves similarly to baised mono-hull roll motion (Figure 3). The bias may be due to a wind moment or cargo shift (Falzarano and Esparza, 1992). Obviously this asymmetry can be reduced or exaggerated if a steady wind is super-imposed on the MODU. Moreover, since this asymmetry is due to the ship shape of the lower hull it is more pronounced at lighter transit drafts and less pronounced in deeper survival drafts. The asymetry is somewhat important at typical operating drafts. Moreover, the presence of a lateral mooring system will also effect the stiffness of the system. As long as the lateral mooring system is intact its presence is beneficial. However, if a vessel is damged or its mooring system is dragged or damaged these will further contribute to the asymmetry of the system. The possibility that a damaged or dragged mooring line could have a detrimental effect is significant because often conservative analysis of MODU stability neglects he mooring system entirely.

Second, although the pitch righting arm of a MODU is asymmetric while roll is not, MODU pitch asymmetry is usually much less than the pitch for a typical mono-hull. This is due to the fact that only the ship-shape of the lower hull contributes to this asymmetry and the vertical columns are usually placed symmetrically with respect to the vessel's midships. This minimum amount of asymmetry will allow the pitch to be approximately decoupled from the heave for small angles. Moreover, the heave if considered will act as a passive coordinate (Falzarano and Zhang, 1993) and not effect the pitch qualitatively only quantatively. Again, the intact mooring system will increase the vessel's pitch stiffness but a damaged hull or mooring system will increase the asymmetry and therefore pitch heave coupling.

Uniqueness of Modeling Offshore Platform's Dynamics

Mobile Offshore Drilling Units (MODU's) differ from traditional mono-hull and ship shaped catamarans in that they are mostly made up of various sized submerged horizontal or water-piercing vertical cylindrical elements. Moreover these vessels usually operate in relatively shallow water although this is not always the case especially with the growing importance of deep-water exploration and production. The small size of many of these cylindrical elements either allows or requires alternatives to traditional strip theory ideal flow free-surface hydrodynamics modelling (Figure 2). The techniques used for slender members on floating platforms are generally adaptations of the Morrison type approaches used to predict forces on jacket-type fixed platforms. These methodologies usually depend upon empirically predicted inertial and viscous force coefficients. The viscous force usually depends upon a relative velocity squared. The relative velocity depends upon the wave induced orbital velocity and the vessel motion itself. The velocity squared term makes linearized techniques limited to very small motion amplitudes.

Moreover, the strip theory approximation which is quite effective for slender ships in head seas may be quite approximate or wholly inappropriate for many types of floating offshore platforms in certain modes of motion. Moreover, the interaction of the various column elements may be important and must be considered. In addition, offshore platforms often operate in relatively shallow water relative to long wave lengths of importance and this changes the hydrodynamic coefficients due to the presence of the bottom. Therefore, although some of the same techniques used to study mono-hull seakeeping can be applied to MODU's, they must be used cautiously and possibly modified or augmented to account for these unique features of the MODU and their operating environment.

Analysis Approach

Considering that MODU pitch motion is approximately as critical as its roll, in this paper, the critical pitch behavior is studied in some detail. As can be seen by the plot of tilting arm for various orientation angles, it is immediately evident that the tilt about approximately the quartering axis is minimum and therefore most critical and should ultimately be analyzed. However, again, assuming that the vessel could be pointed into or initially positioned into the direction of the most critical waves and in the interest of initially solving a more simple problem in order to ultimately gain insight into the more exact problem, we study pitch in isolation. In a future work, we plan to study large amplitude

MODU motion at a arbitrary heading and in random waves.

Two approaches are utilized in this paper to study the critical dynamics of typical MODU's. First, is to study the time behavior of the vessel and second, is to study the critical behavior boundaries. We select the regular wave frequency and amplitude based upon the maximum vessel linear response in a given sea spectra. Although, nonlinear resonance behavior is far more complicated than simple linear resonance, the linear superposition approach approximately (Taz Ul Mulk and Falzarano, 1994) will identify the peak response in the given seaway. The linear response will consider the peak wave exciting force per unit wave amplitude, the peak vessel response for a unit wave exciting force and the wave spectral peak. The first two are collectively refereed to as the vessel's (linear) Response Amplitude Operator (RAO) and when squared and multiplied by the given sea spectra point-by-point will yield the vessel's response spectra in that seaway.

Some Results for a Typical Floating Offshore Platform

In order to add realism to this study, we study a specific MODU which capsized in 1982 resulting in a multiple loss of life (USCG, 1982). By studying a specific platform we do not mean to suggest that this platform was either poorly designed or operated; on the contrary, it was probably one of the safest platforms at the time of its capsizing

Results

The first step in any sort of vessel stability analysis be it traditional or that proposed herein, is to calculate the vessel static righting arm curves. As mentioned previously, for a MODU the vessel's pitch restoring arm is roughly the same order of magnitude as the roll restoring moment arm. This can be seen in Figure 4. Moreover, what can also be seen in the figure is that the pitch restoring arm is asymmetric with respect to positive or negative pitch rotation. Also plotted in the same figure are the positive and negative tilt restoring arms about a 45 degree orientation axis. One can see that it is this orientation axis or one close to it where the vessel's restoring arm is minimum. Again, due to the vessel slight fore/aft asymmetry these curves are also slightly asymmetric. For comparison, the symmetric roll restoring moment is also included in the same figure.

The next step in this analysis is to determine the vessel dynamic response in a particular seaway (Figure 5) (USCG, 1982). As such, the pitch linear Response Amplitude Operator (RAO) are also determined. As mentioned previously, since MODU's operate in shallow water, the effect of water depth on the RAO's can significant. As such we include both an infinite depth RAO and one for the vessel operating in 300' of water (Figure 6). As can be seen in the figure, the RAO is not substantially effected by the presence of the bottom below a frequency of 0.5 rad/s which corresponds to a 700 foot wavelength. The 700 foot is close to a water depth equal to one half the wavelength which is a reasonable approximation to the diminishing effect of the bottom (Newman).

An alternative approach to linear frequency domain analysis which is probably more appropriate for large motion amplitudes, is time domain simulation. We have prepared a simple time domain integration program which utilizes the frequency domain hydrodynamics from above and a polynomial curve fit to the pitch restoring arm. By comparing the two time histories (Figure 7a and 7b), one can see harmonic response for low frequencies close to resonance and two frequency response above resonance. These two frequencies appear to be the resonance frequency and the forcing frequency. Finally, we include a plot of the vessel's pitch safe basin (Falzarano and Troesch, 1990) (Figure 8), for a wave forcing frequency of 0.3 rad/s and a wave amplitude of 1 foot. One can see from this result that due to the slight asymmetry of the pitch restoring moment curve, the safe basin is similar to a ship biased due to wind.

Conclusions

The results contained herein should be considered preliminary, and accurate in a qualitative sense at best. In this brief investigation, what we have identified are the regions of validity of the various modelling approximations and areas for refinement and future work. It turns out that due to the MODU's pitch resonance being low by design, far below the dominant wave frequencies, using ideal flow is questionable since the wave damping is negligible in this region and the viscous damping on the slender members may be significant.

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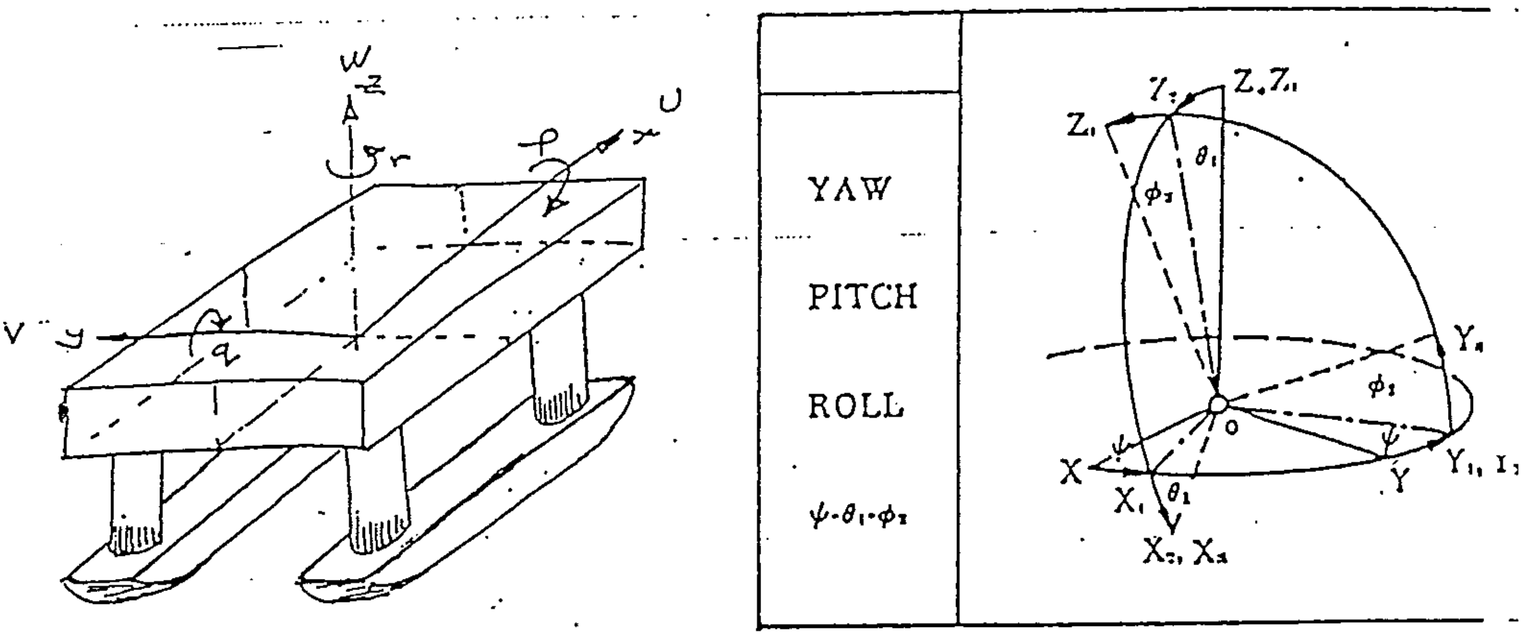

Figure 1 a) Platform Coordinate System, and b) Euler angle rotation sequence (Thelander, 1964)

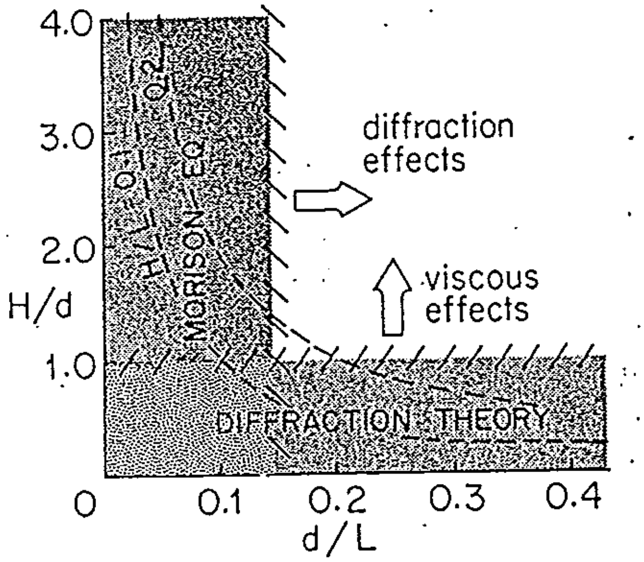

Figure 2 Regions of validity: wave interaction with a cylinder
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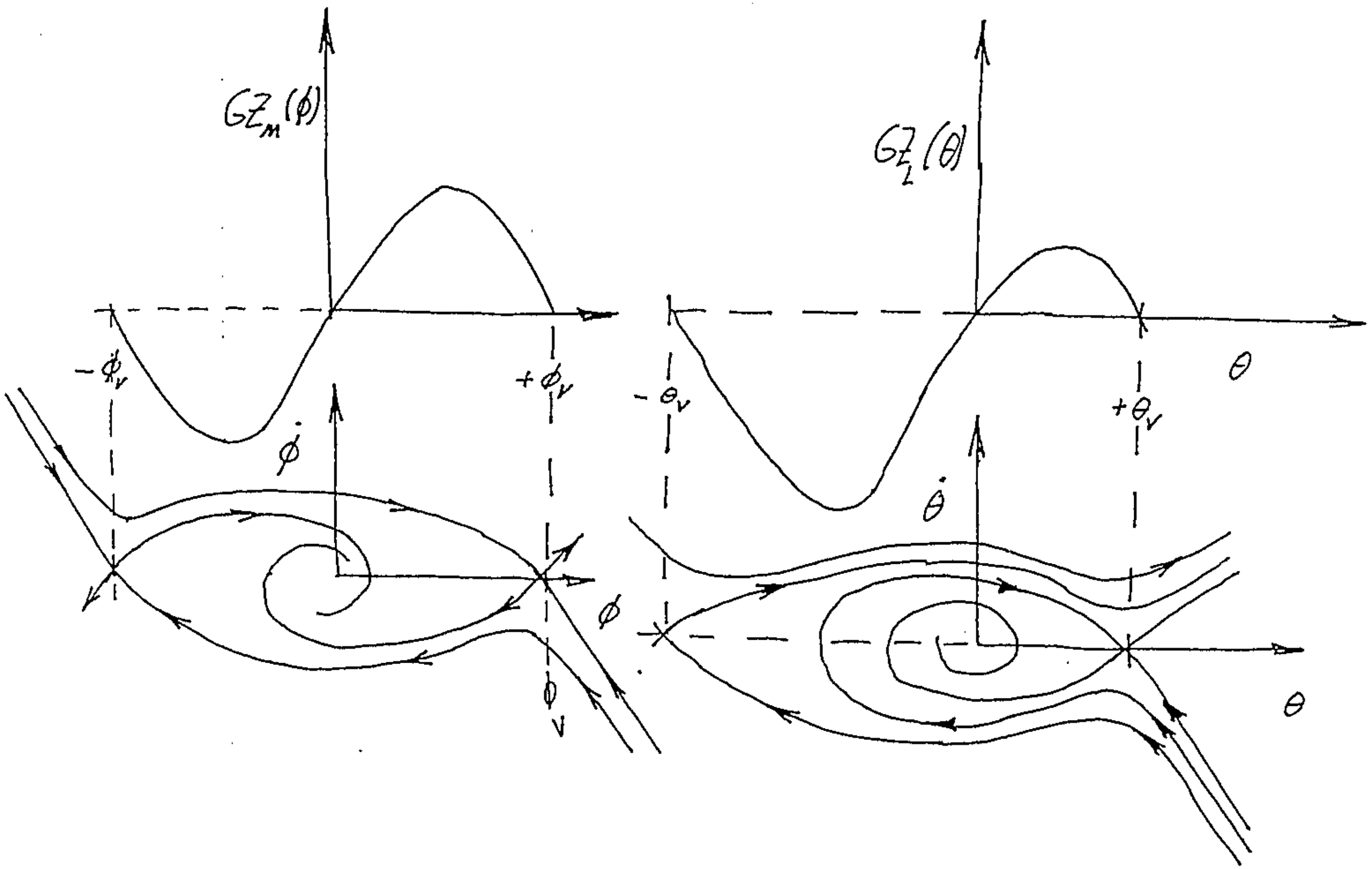
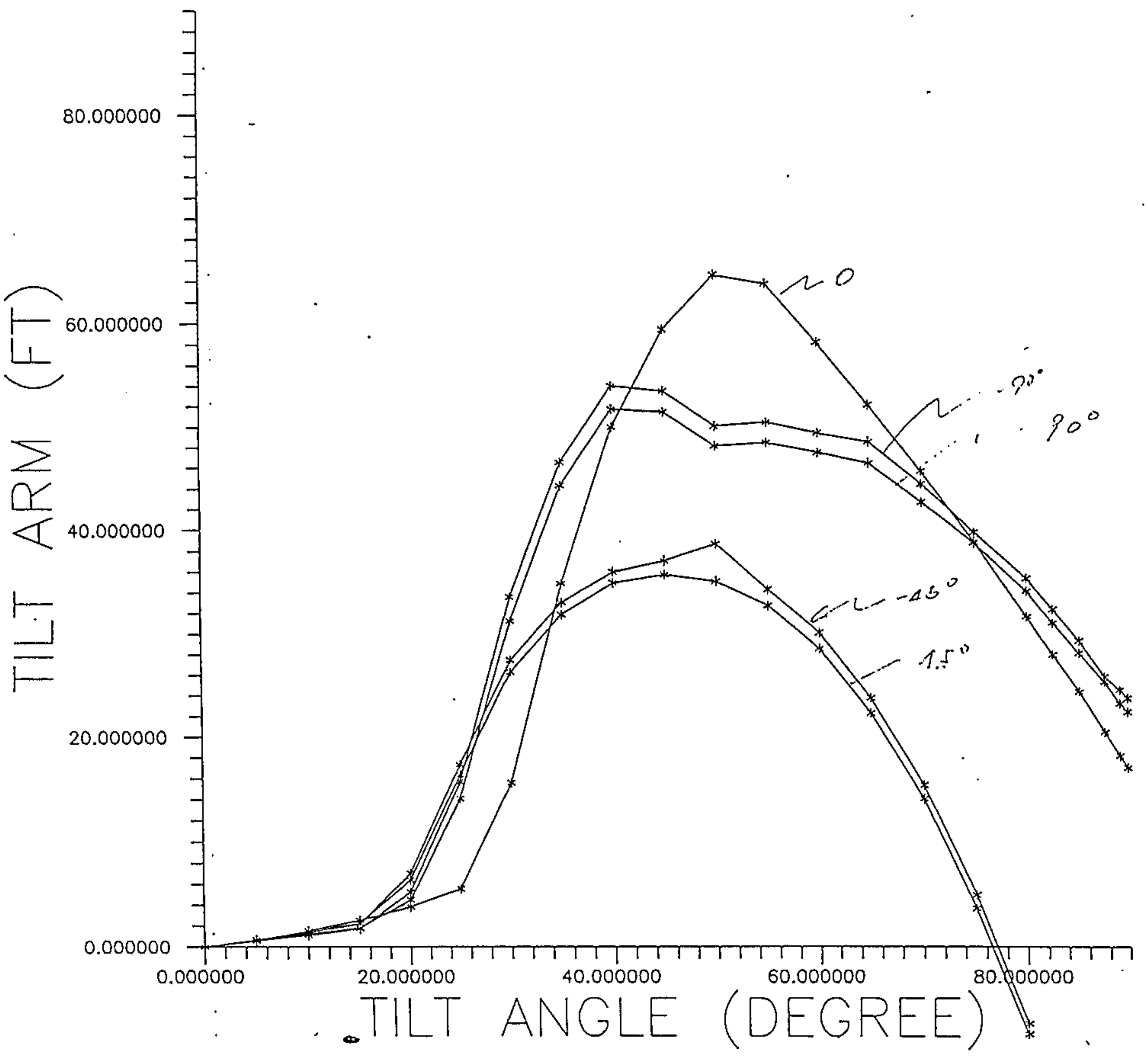


Figure 3 Righting arms and corresponding Poincaré maps for ship roll and MODU pitch



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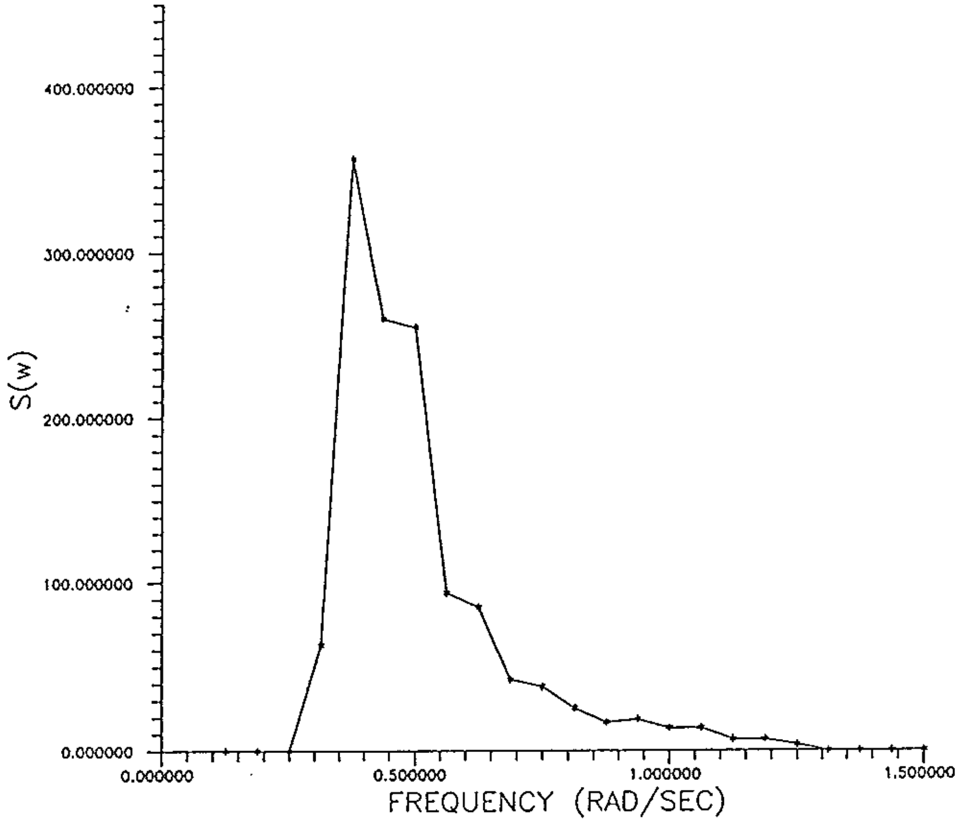

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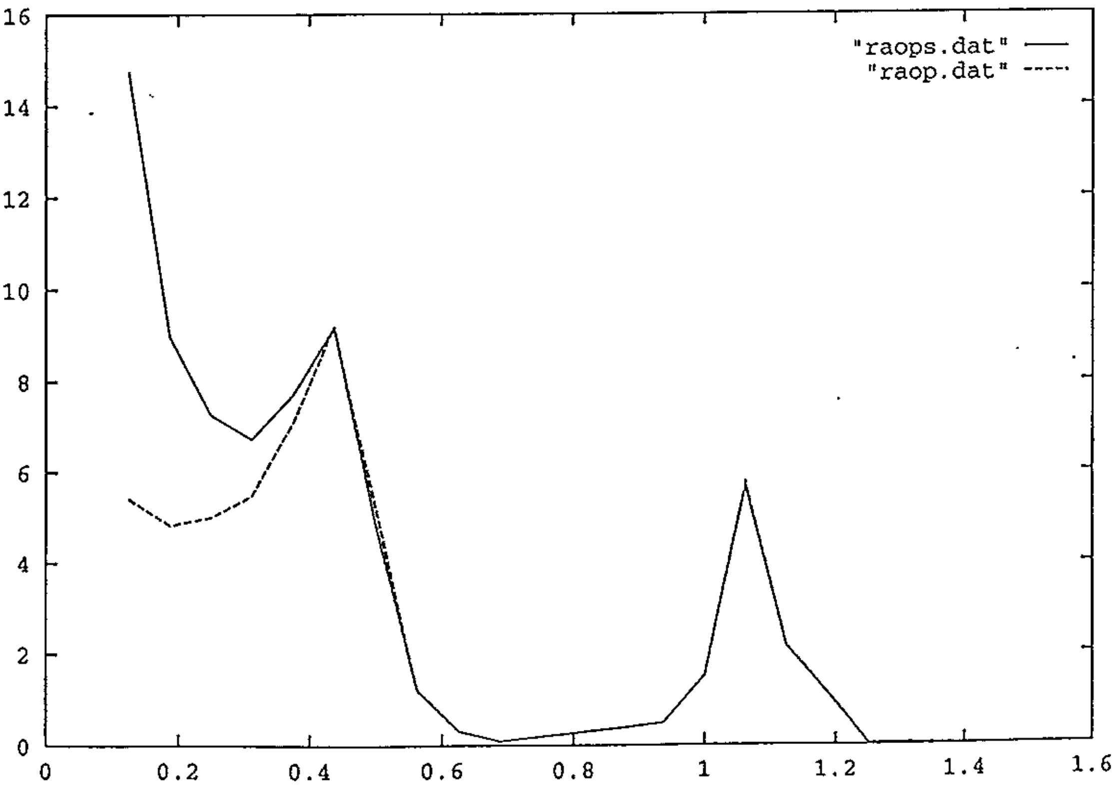

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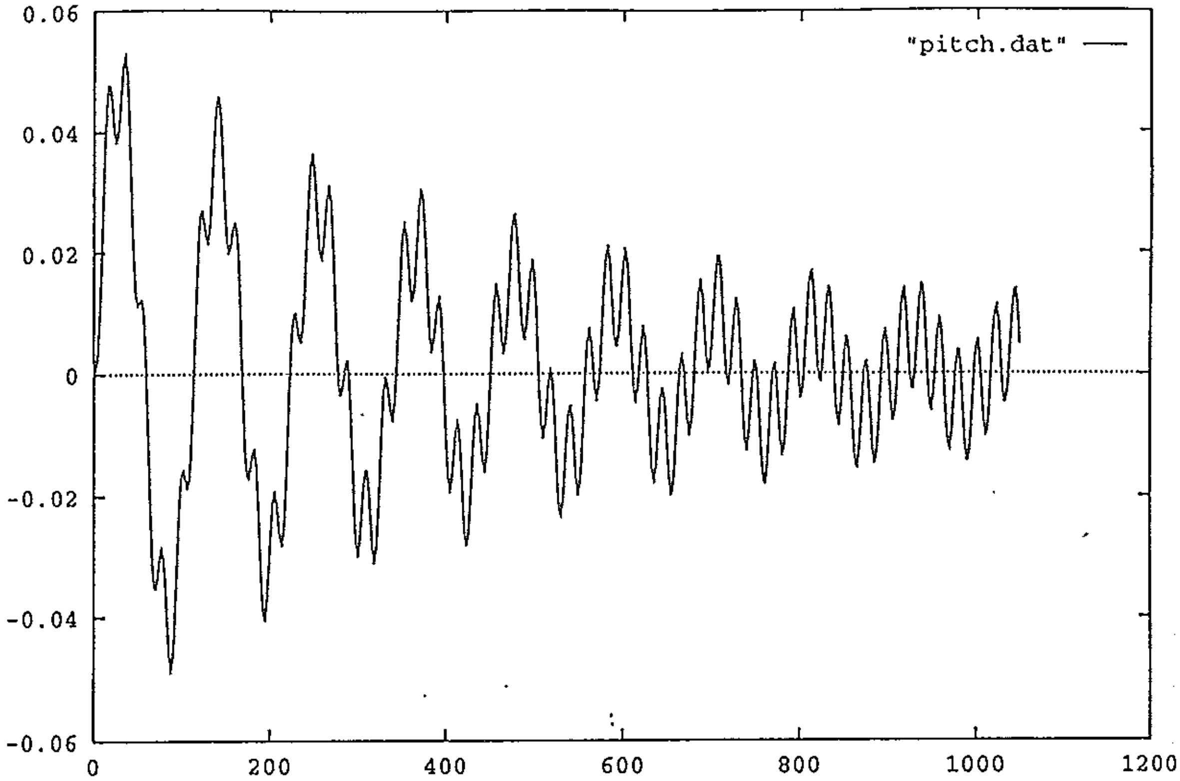

Figure 7b Time history of pitch motion in regular wave
ω = 0.1 rad/s, η = 0.1 ft

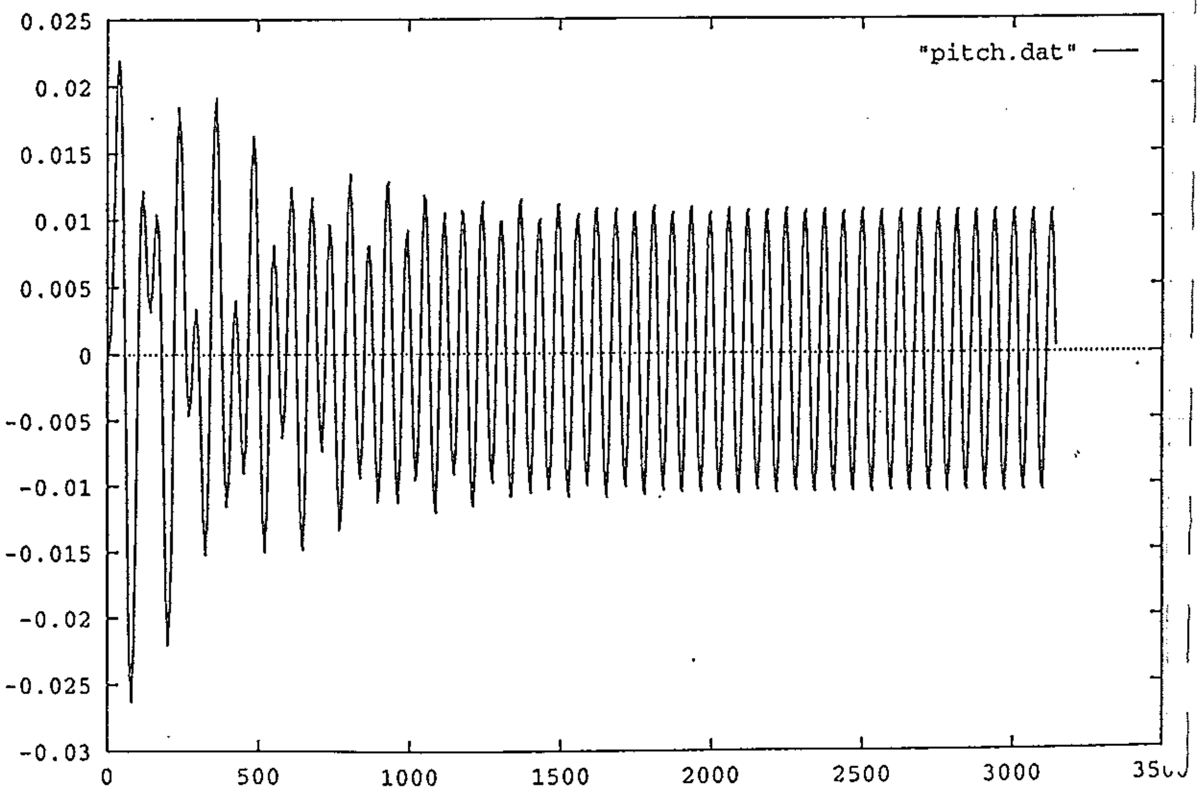

Figure 7a Time history of pitch motion in regular wave
ω = 0.3 rad/s, η = 0.1 ft

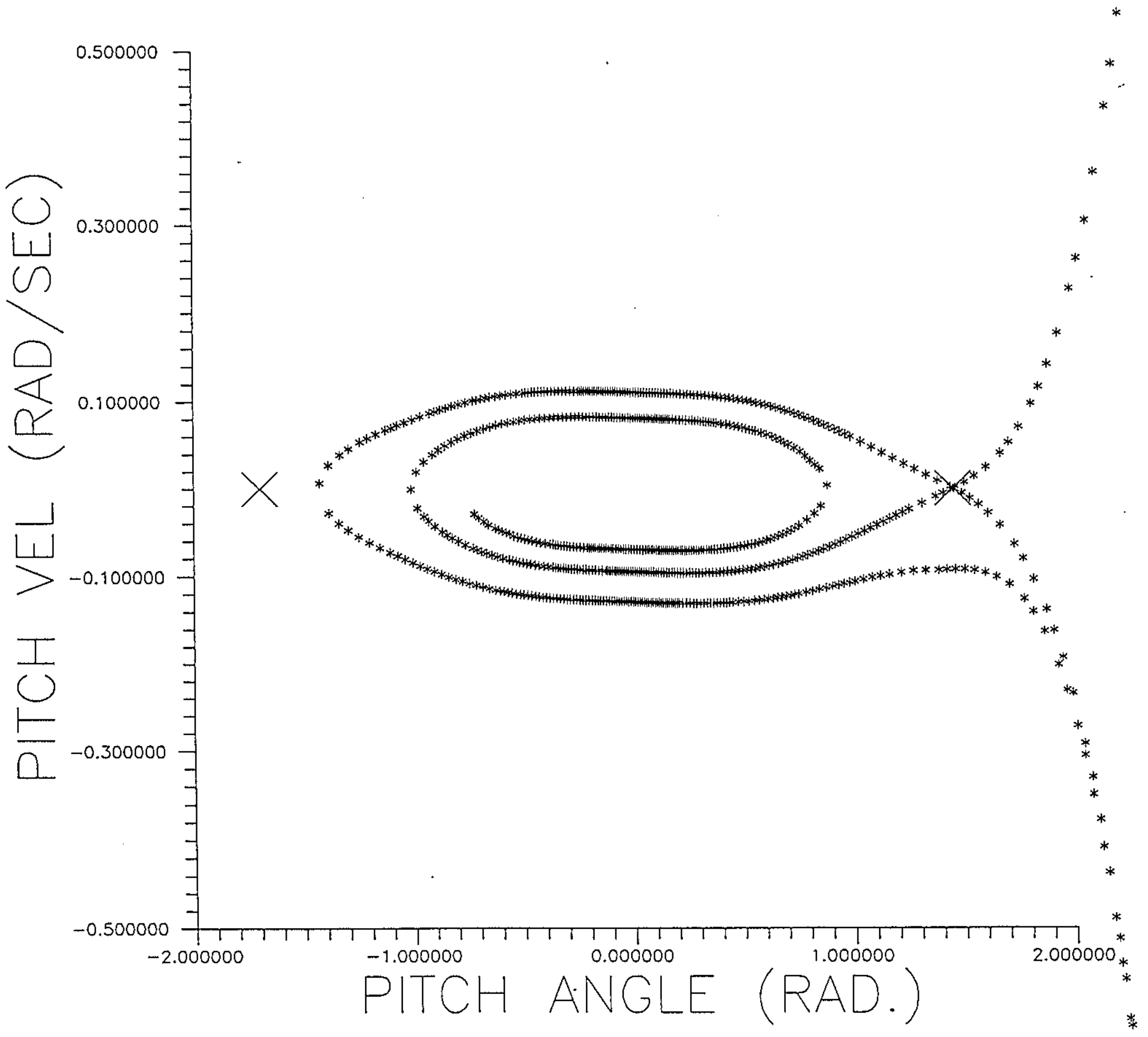

Figure 8 Pitch motion Poincaré map, ω = 0.3 rad/s, η = 1.0 ft

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Operational Stability in Following and Quartering Seas - A Proposed Guidance and Its Validation -

by
N.Umeda
National Research Institute of Fisheries Engineering, Japan

ABSTRACT

In this paper, a new practical guidance to the master for avoiding dangerous situations in following and quartering seas is described and validated. This guidance presents a desired course and velocity to avoid capsize due to pure loss of stability as well as due to broaching. For the pure loss of stability, the reduced righting arm on a wave profile and the spectrum of ocean waves are taken into account. For broaching, the critical velocity is identified for surf-riding, because surf-riding can be regarded as a prerequisite of broaching. The guidance requires the master to detect only the Beaufort scale, wave period and natural roll period. To examine this guidance, existing results from model experiments of fishing vessels and container ships are compared with the dangerous zones that are indicated by the guidance. As a result, it is confirmed that all capsize events in these experiments occurred in the dangerous zones specified in the guidance.

INTRODUCTION

To keep shipping and fisheries profitable, ship capsize should be prevented. Thus, a ship is designed to satisfy stability criteria. Stability criteria, such as the IMO weather criteria, require the ship to survive under heavy weather for a long duration without propulsive power and control. In this situation waves and wind makes the ship perpendicular to wave direction. The master of this ship is assumed to have nothing to do for escaping from this situation. This is the reason why the weather criteria consider beam wind and waves for a long duration.

These weather criteria had been thought to exclude not only capsizing in beam seas but also capsizing in following and head seas. However, it has been empirically or experimentally established that even a ship satisfying the weather criteria can capsize under some navigating conditions in following and quartering seas. In these cases, the ship master can avoid dangerous situations by changing ship speed or direction with propulsive power and a rudder. In other words, the master can avoid capsizing during navigating in following and quartering seas. At least, with the master stopping the engine, the ship satisfying the weather criteria shall survive in beam seas.

It seems to be not impossible to design a ship for preventing capsizing at every condition in following and quartering seas. However, a recent study (Umeda & Ikeda, 1994) shows that the existing proposal of stability criterion for container ships in following seas (Germany, 1990) cannot completely prevent capsizing in following and quartering seas. Nevertheless, the proposal is much more stringent than the weather criteria (Motora, 1990). Since capsizing in following and quartering seas is induced by waves whose length is similar to ship length, smaller ships can more easily meet dangerous waves and more easily capsize than larger ships. Thus, the "fail safe" stability criterion may spoil profitability of smaller ships with stringent limitation of loading. Therefore, it is more practical to provide a guidance from the ship designer to the master for avoiding dangerous situations in following and quartering seas than to establish the fail safe criterion for the designer. (Motora, 1990 and Japan, 1990) In other words, capsizing in following and quartering seas will be prevented with cooperation between a ship designer and master.

The following requirements can be pointed out to be satisfied with the guidance:

- It should be practical for ship operation.
- It should be used with ease by mariners, while it is provided by a ship designer with as much task as the IMO weather criteria.
- It should consider that a ship has a certain stability required by existing stability criteria.
- It should correspond to physical phenomena that are established by the current investigations.
- It should exclude capsizing observed in reliable model experiments.
- It should be applicable to all types of conventional ships without limitations in size.

Responding to Japanese proposal for providing a guidance to the master for avoiding dangerous situations in following and quartering seas (Japan, 1990), several versions of guidance have been presented by Japan (1991), Russia (1992) and Poland (1993). Nevertheless, these versions are found not to satisfy all the above requirements. (Japan, 1994A) Then, the author proposed a new draft guidance to satisfy all the above requirements. (Japan, 1994B) This paper describes details of the new draft guidance and presents results of its validation.

MODES OF CAPSIZING

It is well established that capsizing in following and quartering seas consists of the following modes (Paulling et al., 1975):

- pure loss of stability
- broaching
- low-cycle resonance

Here the capsizing due to pure loss of stability means that a ship suffers a large amplitude roll motion due to waves and then capsizes when a wave crest slowly passes through the centre of ship gravity. The capsizing due to broaching means that a ship surf-ridden loses her directional control and then capsizes with a violently increasing yaw motion. The capsizes due to low-cycle resonance means that a roll motion at one half of the encounter frequency increases in a cumulative manner up to capsize.

Experimental results for a container ship models satisfying the IMO stability criteria involve only capsizes due to pure loss of stability and broaching among 490 capsize events and no capsize due to low-cycle resonance. (Kan, 1993) Since low-cycle resonance requires low metacentric height and low ship speed, the IMO criteria may exclude capsize due to low-cycle resonance. Therefore, the new draft guidance considers two major modes of capsizing, namely, broaching and pure loss of stability. Low-cycle resonance will be added if future research will identify its importance for a ship satisfying the IMO stability criteria.

BROACHING

Since surf-riding is a prerequisite for broaching, the critical condition for surf-riding is provided to prevent capsizing due to broaching. The surf-riding condition was definitely identified by a phase plane analyses with an equation of longitudinal motion in following waves (USSR, 1970, Umeda, 1990 and Kan, 1990), and compares well with experiments of various ship models, as can be seen in Fig.1. These studies indicate that the longitudinal ship motion consists of three modes: *periodic surging motion*, *surf-riding under certain initial conditions* and *surf-riding under any initial conditions*. In regular waves, when the nominal velocity of a ship is small enough, the ship has a periodic surging motion. When the nominal velocity exceeds the critical velocity for the *surf-riding under any initial conditions*, the ship suffers surf-riding. In other words, a ship self-propelled with constant revolution in regular waves does not suffer the *surf-riding under certain initial conditions*.(Umeda, 1990) On the other hand, the ship cannot escape from the surf-riding, unless the nominal velocity decreases less than the critical velocity for the *surf-riding under certain initial conditions*, which is smaller than that for the *surf-riding under any initial conditions*. (Umeda, 1990 and Kan, 1990) Therefore, the critical velocity for the *surf-riding under any initial conditions* should be adopted in a guidance for following sea condition.

This critical velocity can be exactly determined by separatrices from equilibria of the phase plane, and mainly depends on the Froude-Krylov force. Kan(1992) confirmed that the critical velocity does not depend on a hull form very much because the Froude-Krylov force is not so sensitive to hull forms. As a result, it was experimentally and theoretically found that the Froude number F_n =0.3 is a good approximation as the critical velocity for surf-riding

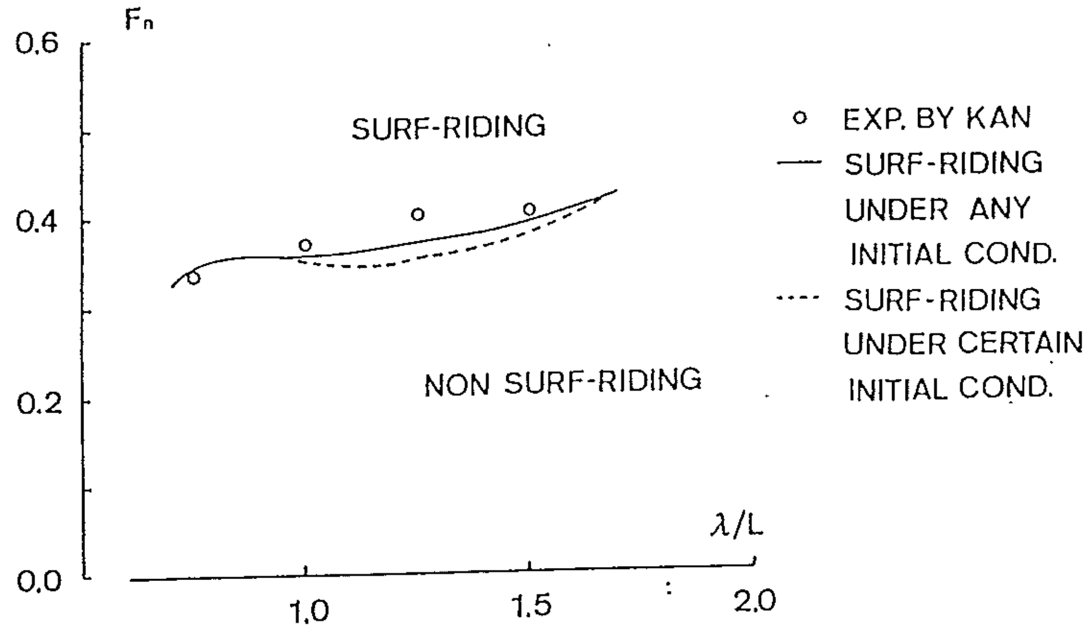

Fig. 1 Critical Froude number F_n for surf-riding of a stern trawler in a following waves whose steepness is 0.05 and wave length to ship length ratio is λ/L. (Umeda, 1990)

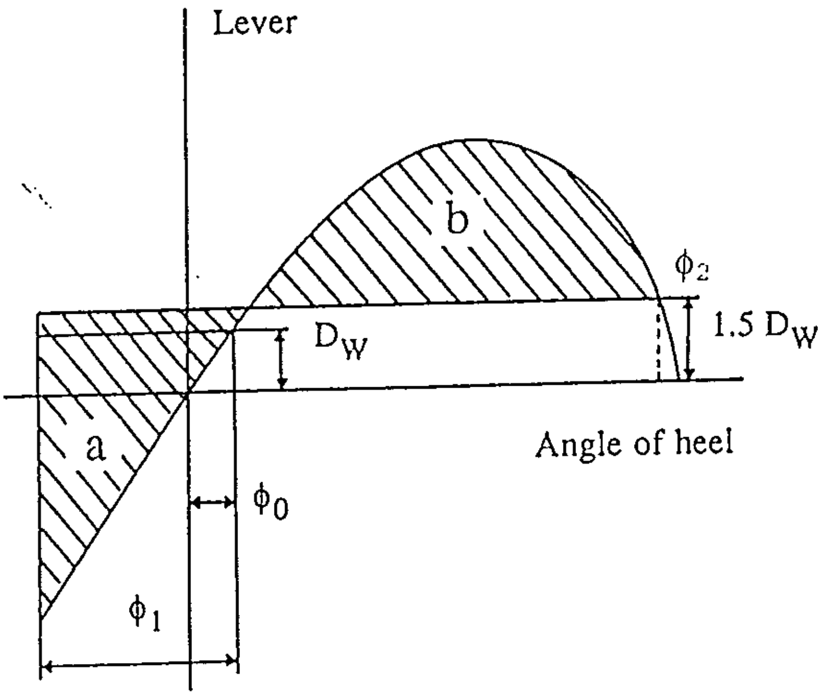

Fig. 2 Energy balance in the IMO weather criteria

of several type of conventional ships when the wave steepness is 1/10. Finally, the surf-riding condition for the guidance can be simply described as follows (Japan, 1991):

$$V(knot) > 1.8\sqrt{L(m)} \qquad (1)$$

where V and L mean ship speed and ship length, respectively. If the ship satisfies the condition obtained by Eq.(1), the master should reduce forward speed of the ship. However, during the surf-riding, reducing the propeller revolution is rather dangerous because the ship cannot always escape from the surf-riding but can lose her rudder effectiveness against broaching. (Umeda, 1990)

PURE LOSS OF STABILITY

In capsizing due to pure loss of stability, a GZ curve and roll motion play dominant roles. For a ship drifting in beam seas, the IMO weather criteria were rationally established with ship dynamics and environments being taken into account. Since a ship is assumed to be under lie-to, a severe roll motion due to resonance cannot be avoided when ocean waves develop. Thus a large rolling motion is considered while a GZ curve is not changed. On the other hand, for a ship running in following and quartering seas, a GZ curve can be drastically reduced by waves while the master can avoid severe rolling motions by changing her course or speed.

Since the guidance indicates the critical velocity for the surf-riding, a ship will not suffer surf-riding. As far as it does not, the ship situates on a wave crest for a longer duration and on a wave trough for a shorter duration because the ship has large variation of surging velocity. This tendency becomes significant when the ship nominal velocity exceeds the critical velocity for the *surf-riding under certain initial conditions*. If the ship nominal velocity tends to the critical velocity for the *surf-riding under any initial conditions*, the duration of the ship on a wave crest tends to even infinite. This phenomenon is sometimes labeled as "riding on crest" (Canada, 1993) or "marginal surf-riding" (Thomas & Renilson, 1991). Here we should note that the ship situates almost on a wave crest in this phenomenon. Thus, the ship is directionally stable but can be transversely unstable. On the contrary, if real surf-riding occurred, the ship is captured by a downhill or a wave trough, and directionally unstable. Although no significant difference seems to be observed from time records of experiments or numerical simulations, nonlinear dynamics definitely discern two phenomena, which have different importance for practical purpose.

Before the ship meets a wave crest, the ship suffers wave exciting roll moment at downhill or uphill of waves. If the ship satisfies resonant condition, the ship suffers large amplitude roll motions and then meets a wave crest, where it situates for a longer duration. If dynamic stability on the wave crest cannot absorb ship rolling energy due to the preceding wave, the ship will capsize.

Thus, we assume that a rolling ship remains on a wave crest for a longer duration. On the basis of this assumption, Umeda et al. (1990) formulated a method to assess stability, and validated with direct numerical simulation in time domain. (Umeda & Yamakoshi, 1992)

Therefore, the improved guidance for the pure loss of stability consists of two parts: a reducing stability condition and a severe rolling condition. If both of them are satisfied, the master should change her course or speed.

<u>Reduced stability condition</u>

Making use of the Beaufort scale B, the reducing stability condition is described as follows:

$$B > B_c \qquad (2).$$

Here the value of B_c corresponds to the Beaufort scale where heeling energy "a" equals to dynamic stability "b" in Fig. 2. The method to calculate "a" and "b" are based on the IMO weather criteria (IMO, 1985) but wind and wave direction for the roll motion is assumed to be 45 degrees from stern. In addition, the GZ curve is calculated for the ship in a longitudinal wave whose crest is at the centre of ship gravity and length is equal to ship length. The wave height is calculated by the Grim's effective wave concept (Grim, 1961) with a wave spectrum.

Details of the method calculating B_c for a certain ship is described as follows.

1. Wave height is assumed to be of a certain value.

2. GZ curve is calculated for the ship whose centre of gravity situates on a crest of longitudinal wave. Here wave length is equal to ship length between perpendiculars. The wave assumed here is identical to Grim's effective wave transformed from irregular waves. In addition, trim effect should be accounted but Smith's effect and other hydrodynamic effects are not. If a computer program for stability in longitudinal waves is not available, a simple empirical formula, such as Nechaev's formula (Nechaev, 1989), can be used as an alternative way.

3. The ship is subjected to a steady wind pressure which results in a steady wind heeling lever, l_{w1}. Here

$$l_{w1} = \frac{P \cdot A \cdot Z \cdot \sin 45^\circ}{1000 g \Delta} (m) \qquad (3)$$

where

P = 504.2 [N/m^2]

A = projected lateral area of the portion of ship and deck cargo above waterline (m^2)

Z = vertical distance from the centre of A to a point at one half the draught

Δ = displacement (t)

g = 9.81 m/sec^2.

4. From the resultant angle of equilibrium, ϕ_0, the ship is assumed to roll owing to wave action to an angle of roll, ϕ_1, to windward. Here

$$\phi_1 = 109 k X_1 X_2 \sqrt{rs} \text{ (degrees)} \qquad (4)$$

where

- X_1 = factor as shown in table 1 of the A.562 (IMO, 1985)
- X_2 = factor as shown in table 2 of the A.562 (IMO, 1985)
- k = factor as follows:
  - k = 1.0 for round-bilged ship having no bilge or barkeels
  - k = 0.7 for a ship having sharp bilges;
  - k = as shown in table 3 of the A.562 (IMO, 1985) for a ship having bilge keels, a bar keel or both

$$r = (0.73 \pm 0.6 OG/d)\sin 45^\circ \qquad (5)$$

with: OG = distance between the centre of gravity and the waterline (m) (+ if centre of gravity is above waterline, - if it is below)

d = mean moulded draught of the ship (m)

- s = factor as shown in table 4 of the A.562 (IMO, 1985).

5. The ship is then subjected to a gust wind pressure which results in a gust wind heeling lever, l_{w2}.

$$l_{w2} = 1.5 l_{w1} \text{ (m)} \qquad (6).$$

6. Repeating the above process, from 1. to 5., with different wave heights, the wave height, H_c, where area "b" is equal to area "a" is determined, with reference to the Fig.3. However, if, with a certain wave height value, the area "b" is greater than the area "a" and the sum of ϕ_0 and ϕ_1 is equal to ϕ_2, this wave height should be regarded as H_c.

7. On the other hand, the 3% largest heights of Grim's effective wave, $H_{3\%}^{eff}$, is calculated for each of the Beaufort numbers as follows (Grim, 1961):

$$H_{3\%}^{eff} = 5.58\sqrt{m_0} \qquad (7)$$

where

CONTAINER SHIP (G-type)
GM=0.764m

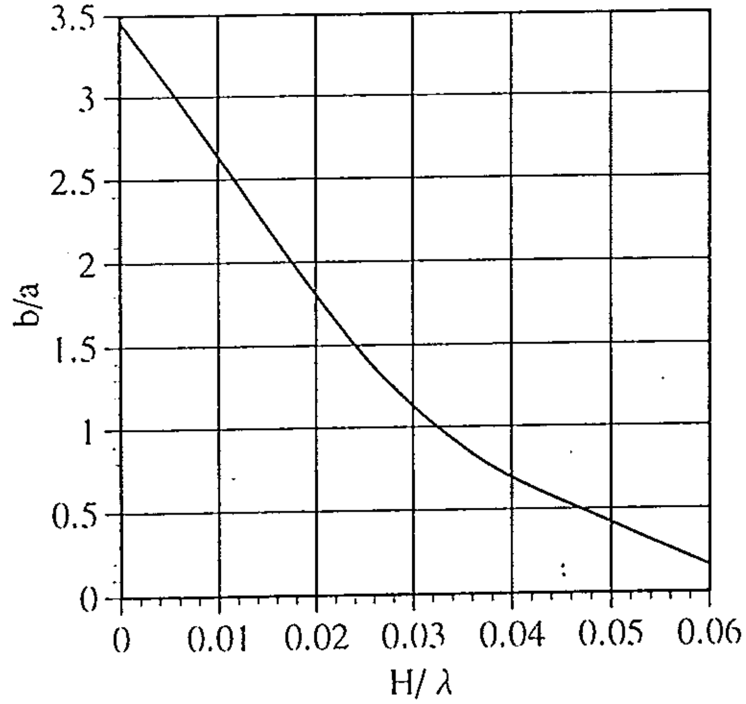

Fig. 3 Reduction of the b/a value due to a longitudinal wave for a container ship

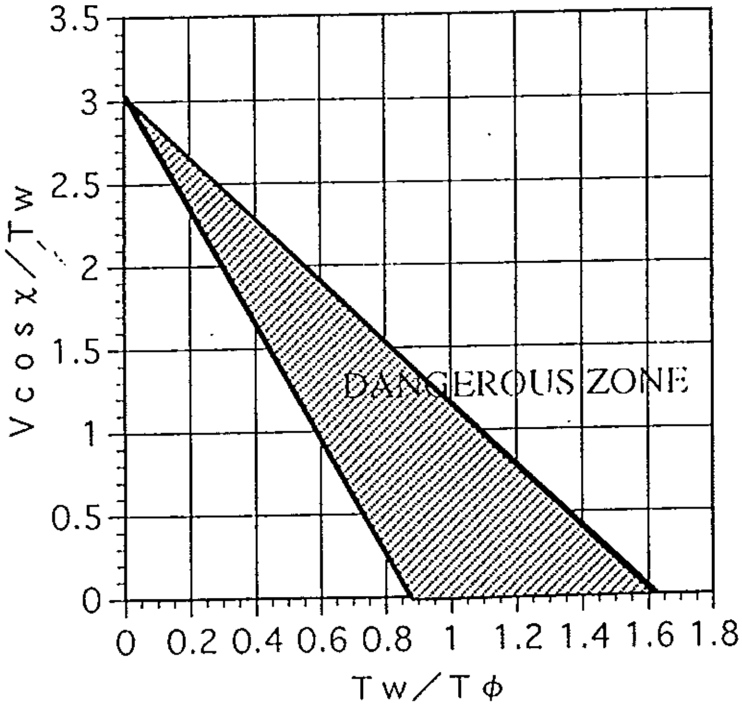

Fig. 4 Dangerous zone (severe rolling condition)

$$m_0 = \int_0^\infty \left\{ \frac{\frac{\omega^2 L}{g} \sin\left(\frac{\omega^2 L}{2g} \right)}{\pi^2 - \left(\frac{\omega^2 L}{2g} \right)^2} \right\} A\omega^{-5} \exp\left(-B\omega^{-4}\right) d\omega \qquad (8)$$

$$A = 173 H_{1/3}^2 T_{01}^{-4} \qquad (9)$$

$$B = 691 T_{01}^{-4} \qquad (10).$$

Here significant wave height $H_{1/3}$ and mean wave period T_{01} are determined by the Beaufort scale, as can be seen in Table 1. Since the integral of Eq.(8) is a function of only ship length and the Beaufort scale, it can be presented in tabular form. Once a table is provided, it is not necessary to calculate the integral.

8. The critical Beaufort number B_c is determined as the smallest number of the Beaufort scale satisfying the following condition:

$$H_{3\%}^{eff} > H_c \qquad (11).$$

Severe rolling condition

The severe rolling condition is obtained as follows:

$$3.03\left\{1-\left(T_w / T_\phi\right)/0.875\right\} < V\cos\chi / T_w < 3.03\left\{1-\left(T_w / T_\phi\right)/1.63\right\} \qquad (12).$$

where T_w, χ and T_ϕ indicate mean wave period, wave direction from stern and natural roll period with small amplitude, respectively. The dangerous zone determined by this formula is shown in Fig.4, which is common to all ships. The formula indicates the condition where an encounter period of the ship to waves is greater than 0.7 times natural roll period and smaller than 1.3 times natural roll period. Further, as a softening spring effect, the natural roll period in heavy seas is assumed to increase up to 1.25 times the natural roll period with small roll amplitude. The natural roll period with small roll amplitude, T_ϕ, can be measured by an inclining test or calculated by the following formula:

$$T_\phi = \frac{2C \cdot B}{\sqrt{GM}} \text{ (seconds)} \qquad (13)$$

where: $C = 0.373 + 0.023(B/d) - 0.043(L/100)$ (IMO, 1985). Although magnitude of softening spring effect depends on details of GZ curve to some extent, the value 1.25 can be used as presumable one for usual ships. (For example, Kan, 1991)

The guidance will be provided by a ship designer when a new ship is completed with documents to prove satisfaction of stability criteria. His main task for the guidance is to determine B_c and is almost similar to that for the IMO weather criteria.

The mariner can use this guidance with ease if he knows the Beaufort scale, mean wave period, wave direction, ship speed and natural roll period. In particular, for the mariner it is enough to usually pay rough attention to the Beaufort scale. Because, it is troublesome to always measure or estimate wave height, period and direction. If the Beaufort scale becomes severe enough, he should estimate wave period, ship speed and direction. These do not increase a task for the mariner very much.

Since this guidance considers relationships between ship size and wave spectrum with Grim's effective concept, it can be applicable to all types of conventional ships without any limitations in size. As a result, the guidance indicates effective advice for the master of smaller vessels, while it does not disturb the master of larger vessels, such as VLCC and large container ships, which are usually safe from dangerous phenomena: in following seas.

EXAMPLES AND VALIDATION

Examples of applying this guidance to two container ships and two fishing vessels are presented.

[Container ship F]
L=135m, B=23m, d=8.37m, C_b=0.589, GM=1.49m
b/a=1.28 (IMO Res.A.562 with wind velocity 26m/sec)
"The master should reduce ship speed to less than 20.9 knots in heavy following seas. When the Beaufort scale is more than No.7, he should select ship course and speed to avoid the dangerous zone shown in Fig. 4."

[Container ship G]
L=135m, B=24.3m, d=8.37m, C_b=0.570, GM=0.764m
b/a=1.96 (IMO Res.A.562 with wind velocity 26m/sec)
"The master should reduce ship speed to less than 20.9 knots in heavy following seas. When the Beaufort scale is more than No.8, he should select ship course and speed to avoid the dangerous zone shown in Fig. 4."

[Fishing Vessel A] - Purse seiner -
L=33m, B=7.51m, d=2.97m, C_b=0.673, GM=0.647m
b/a=0.072 (IMO Res. A.685 with wind velocity 26m/sec)
"The master should reduce ship speed to less than 10.3 knots in heavy following seas. When the Beaufort scale is more than No.1, he should select ship course and speed to avoid the dangerous zone shown in Fig. 4."

[Fishing Vessel E] - Small trawler -
L=14.4m, B=3.05m, d=0.875m, C_b=0.717, GM=0.152m
b/a=1.0 (IMO Res. A.562 with wind velocity 15m/sec)
"The master should reduce ship speed to less than 6.8 knots in heavy following seas. When the Beaufort scale is more

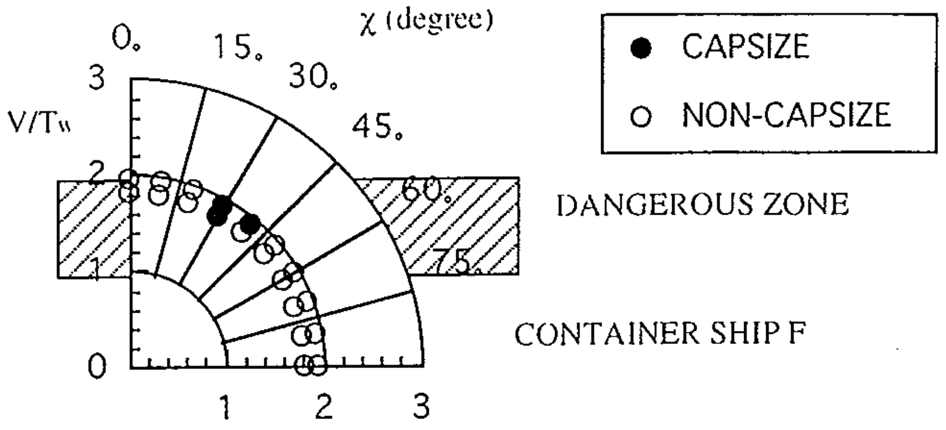

Fig. 5 Experimental results (under the Beaufort No.10-11) and dangerous zone indicated by the guidance

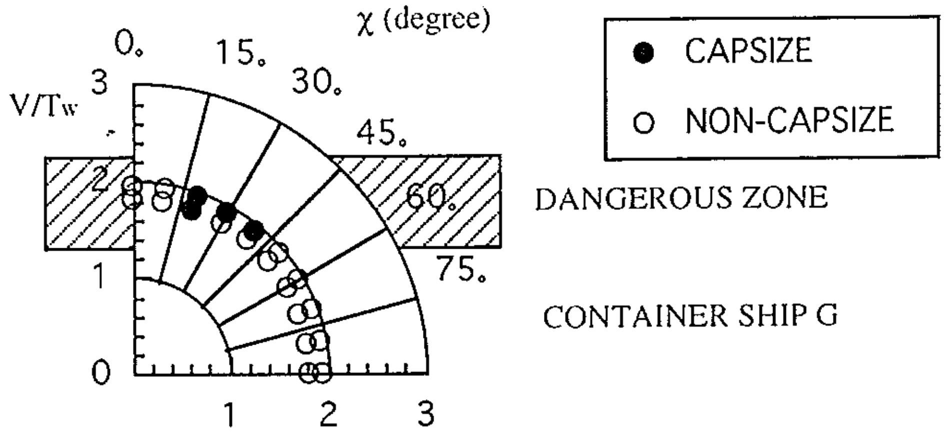

Fig. 6 Experimental results (under the Beaufort No.10-11) and dangerous zone indicated by the guidance

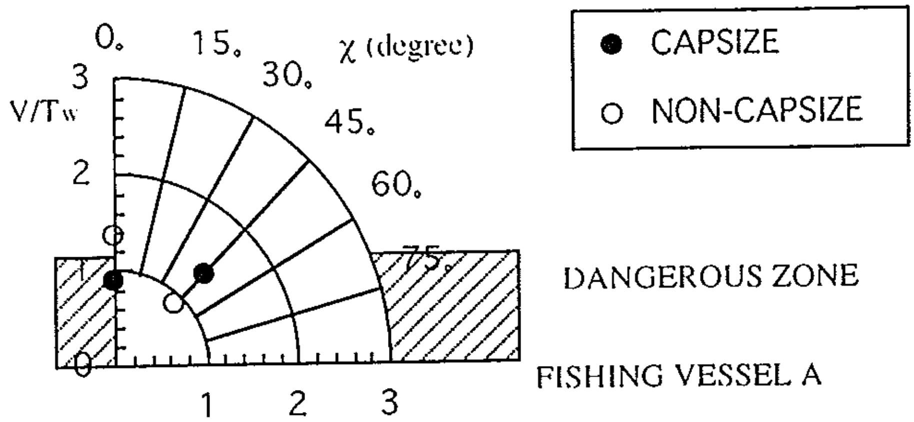

Fig. 7 Experimental results (under the Beaufort No.7) and dangerous zone indicated by the guidance

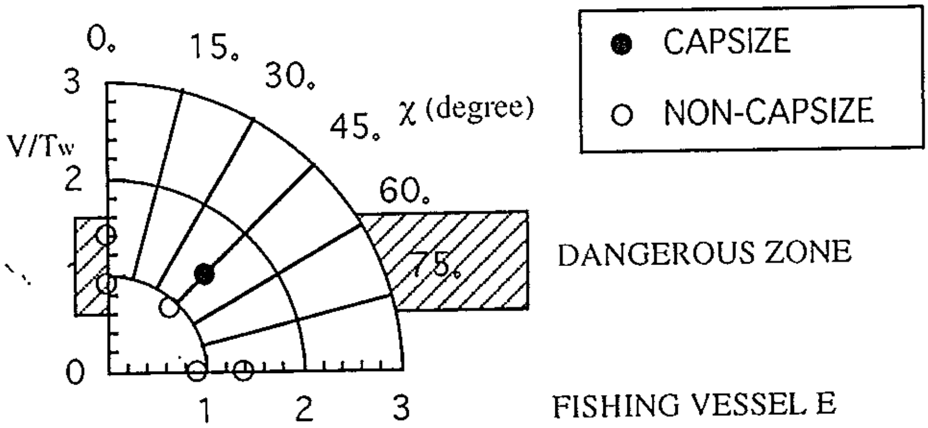

Fig. 8 Experimental results (under the Beaufort No.5) and dangerous zone indicated by guidance

than No.3, he should select ship course and speed to avoid the dangerous zone shown in Fig. 4."

To validate the guidance, experiments of free running models in irregular waves (Kan,1993 and Yamakoshi, 1982) are compared with the newly-developed guidance. The results are shown in Fig. 5 to Fig. 8. These comparisons show that all capsize events occurred within the dangerous zone indicated by the guidance and no capsize was observed outside the dangerous zone. In addition, no broaching was observed outside the surf-riding condition. Therefore, the newly-proposed guidance was validated by model experiments for small ships as well as large ships. The new guidance shows better agreement with model experiments than the other existing proposals. (Japan, 1994A)

Although the dangerous zone obtained by Eq.(12) seems to be rather large, the use of the dangerous zone is limited by the Beaufort scale. Thus, the guidance cannot require excessive limitations for practical operation of ships with acceptable stability. On the other hand, for a ship with poor stability, the guidance requires stringent limitations as the case of the fishing vessel A. The guidance can fairly evaluate ship stability for operation without contradiction with the IMO weather criteria because the guidance and criteria were based on a common methodology, that is, energy balance method.

CONCLUSION

On the basis of this paper, it is found that the newly-developed guidance satisfies all of seven requirements for a suitable operational guidance to the master for avoiding dangerous situations in following and quartering seas as follows:

- It is practical for ship operation.
- It can be used with ease by mariners, while it can be provided by a ship designer with as much task as the IMO weather criteria.
- It considers that a ship has a certain stability required by existing stability criteria.
- It corresponds to physical phenomena that are established by the current investigations.
- It excludes capsizing observed in reliable model experiments.
- It is applicable to all types of conventional ships without limitations in size.

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Table 1 Beaufort scale*

Beaufort No.	$H_{1/3}$[m]	T_{01}[sec]
1	0.1	1.22
2	0.2	1.73
3	0.6	2.99
4	1.0	3.86
5	2.0	5.46
6	3.0	6.69
7	4.0	7.72
8	5.5	9.05
9	7.0	10.21
10	9.0	11.58
11	11.5	13.09
12	14.0	14.44

* This table is obtained with the World Meteorological Organization Code 1100 and Pierson-Moskowitz's formula.

MATHEMATICAL MODELLING OF MOTIONS AND DAMAGED STABILITY OF RO-RO SHIPS IN THE INTERMEDIATE STAGES OF FLOODING

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Abstract

It is generally recognized that the phenomenon of a 'rapid capsize' during the intermediate stages of flooding may represent a potentially dangerous situation to a ship. This is particularly true for a roll-on/roll-off ferry sustaining a 'high energy' collision due to the lack of subdivision on the cardeck.
In order to analyse the associated safety problems a mathematical model has been developed with the aim to describe the actual motion behaviour and the associated residual stability of a ship in the time domain after sustaining a collision damage.
This prediction instrument has been validated to a great extent by model experiments for a pontoon with crossduct connected wingtanks and a typical ferry with a few different damage orifices and in addition various compartment configurations.
An outline of the mathematical model and the corresponding computer program is presented discussing in particular the dynamics involved during the ingress of water with due emphasis on the importance of the instantaneous roll angle and the effect of subdivision arrangements on the distribution of flood water.
Further the results of a study on the parameters involved, such as the configuration of the damage orifice, and systematic calculations carried out for a number of existing ro-ro passenger ships yielded a number of conclusions relevant to the inherent safety of such vessels after a high energy collision.
These conclusions address the validity of calculations for regulatory purposes traditionally based on quasi-static considerations, the actual occurring angle of heel in relation to possible shifting of cargo and the effectiveness of several subdivision alternatives.

1. Introduction

Due to a great number of capsize accidents involving ro-ro ships there was a growing concern about the safety of this type of ship. This development has led to an intensified effort in the field of scientific research and safety legislation.

It was generally recognized that the phenomenon of rapid capsizing during the intermediate stages of flooding represents a potentially dangerous situation to a ro-ro ship after a collision damage. This view is also reflected in previous studies and reference is made to e.g. [1], [2] and [3] for a more comprehensive analysis of the different aspects under consideration.
Because of this reason a research project was initiated with the aim to develop an accurate numerical prediction model of the phenomenon.
The present paper gives an account of the development and validation of this prediction instrument with the capability to identify dangerous situations.

2. Mathematical model

The basic assumptions, inherent to the computer program, are that the ship is floating with zero speed in still water and that the damage opening is in no way obstructed by the ramming vessel. This means that the ship motions due to wave excitation are ignored and likewise sloshing of flood water has been disregarded. In this way the flooding mechanism is more explicitly reflected.
For the initial condition at t=0 it has been arbitrarily assumed that the damaged ship is motionless.

There are three main areas which are relevant in the technical sense to be distinguished for the structure of the mathematical model describing the ship behaviour due to sudden water ingress:

- hydrostatic particulars of the ship
- flow characteristics of flood water
- motion response of the ship in the time domain

In order to give an outline of the mathematical model the above indicated elements will be discussed in more detail in the following sections.

2.1 <u>The motion response of the ship in the time domain</u>

The ship motions are calculated using the equations of motion for a mass-spring system. The coupled motions roll, sway and yaw have been used whereas the dynamics of the other motions have been ignored.

For a harmonic excitation of a linear system the equations of motion may be written as follows (using a generalized vector notation):

$$\sum_{j=2,4,6} (A_{kj}+a_{kj})\ddot{\eta}_j+b_{kj}\dot{\eta}_j+c_{kj}\eta_j = X_k \qquad (k = 2,4,6)$$

where: A_{kj} = mass or moment of inertia of ship

a_{kj} = hydrodynamic mass or hydrodynamic moment of inertia
b_{kj} = hydrodynamic damping coefficient
c_{kj} = spring coefficient
η_j = displacement of harmonic oscillation in direction j
X_k = harmonic exciting force or moment in direction k

The linear hydrodynamic coefficients in the frequency domain have been calculated with a program based on potential flow theory using the so-called 'strip theory' method. For the determination of the hydrodynamic deep water two-dimensional coefficients of ship-like cross sections, these sections are conformally mapped to the unit circle by the so-called two-parameter (i.e. sectional half breadth to draught ratio and sectional area coefficient) Lewis transformation.
However the computer program offers a choice of other conformal mapping methods in association with an alternative potential theory to be used for any desired water depth.

In order to carry out accurate ship motion response calculations in the time domain non-linearities and memory functions have to be included in the equations of motion as follows:

- non-linear contributions to be included in the right-hand side of the equation of motion additional to the external force/moment terms

- non-linear viscous roll damping to be calculated according to the empirical method of Ikeda et al [4]

- convolution integrals have to be applied in order to use hydrodynamic data, which are available in the frequency domain only, in transient calculations

For this purpose the memory functions can be represented by the 'Cummins' equations [5], [6] resulting in $a_{kj}(t) = a_{kj}(\omega)$ for $\omega = \infty$ and a damping term written in the form of a convolution integral consisting of the factors:

j-th velocity component at time t $\dot{\eta}_j(\tau)$ and a retardation function

$K_{kj}(\tau)$ which reads as:

$$K_{kj}(\tau) = \frac{2}{\pi}\int_0^\infty b_{kj}(\omega)\cos(\omega\tau)\,d\omega$$

2.2 The flow characteristics of flood water

The flow characteristics of flood water are completely described by the application of two fundamental theorems:

- Bernoulli's theorem
- Boyle-Gay Lussac's law

Application of Bernoulli's theorem and taking into account the linear distribution of the hydrostatic water pressure over the height of the orifice leads to the following expression for the flux ΔQ of flood water through any horizontal strip:

$$\Delta Q = \Delta A\{(2\Delta P)/(\rho C_D)\}^{1/2}$$

where
ΔA = sectional area of flow strip considered
ΔP = pressure difference across the strip
ρ = density of sea water
C_D = pressure loss ('drag') coefficient

The value of the pressure loss coefficient C_D for cross flooding openings may be found in literature and reference is made to e.g. [7], [8] and [9].
In this regard it should be noted that the speed reduction factor F as defined in [9] is related to C_D as $F = 1/(1+C_D)^{1/2}$.

In a similar way the application of Boyle-Gay Lussac's law in addition to Bernoulli's theorem leads to an expression for the flow rate of air.

2.3 The hydrostatic particulars of the ship

The restoring moment is determined by the following expression:

$$c_{\varphi\varphi}\varphi = \Delta(KN_\varphi - KG)\sin\varphi$$

where
φ = angle of heel
Δ = ship's displacement
KN_φ = metacentric height above keel
KG = distance of ship's vertical centre of gravity above keel

It should be noted that the other spring terms for sway and yaw are non-existent (i.e. zero).

The heeling moment M_φ is determined by a summation of the inclining moments caused by the weight of flood water in each compartment:

$$M_\varphi = \sum_i \gamma v_i(-y_i\cos\varphi + z_i\sin\varphi - KG\sin\varphi)$$

where
γ = specific gravity of sea water
v_i = volume of flood water in compartment i
y_i = transverse position of centre of gravity of v_i from centre line
z_i = vertical distance of centre of gravity of v_i above keel

Use has been made of a conventional 'ship hydrostatics' software package modified in such a way to fit this particular application. On the basis of the defined geometry of hull form and compartmentation arrangement the following information is available:

- the intact stability characteristics by means of so-called cross curves ($KN_\varphi \sin\varphi$) as function of the instantaneous ship draught, trim and angle of heel

- the actual location of the centre of gravity (x_i, y_i and z_i) of the instantaneous volume of flood water v_i in each compartment i considered in the flooding process

3. Computer program

The applied calculation method has been laid down in a computer program called DYNING, which is the acronym of the expression 'DYNamic INGress' referring to the dynamic response of a ship due to a sudden ingress of water.
An interface has been established between the main program with two pre-processing programs generating the following information:

- hydrodynamic coefficients of the equations of motion as described in section 2.1 of the paper; all algorithms are described in detail in reference [10]

- hydrostatic data and tank data as described in section 2.3 of the paper

The 2nd order differential equations of motion are solved by a Runge Kutta time integration.

The output of the computer program yields information, presented as a function of time, on the following parameters:

- amount of sea water in all compartments liable to flooding in the damage case under consideration

- draught, trim and roll angle

- residual righting lever curve $GZ_\varphi = (KN_\varphi - KG)\sin\varphi - M_\varphi/\Delta$ at any arbitrary time instant during the intermediate stages of flooding assuming the amount of flood water in each compartment fixed to the amount present at that time instant

The effect of permeability, vent openings, tween decks and major obstructions, such as a propulsion unit in the engine room, are accounted for in the mathematical model in order to reflect the actual flooding process as closely as possible.

4. Experimental validation

The calculation model has been validated by model experiments on a rectangular pontoon with crossduct connected wingtanks. A comprehensive discussion of the above results has been presented in reference [11].

Further model experiments have been carried out on a representative ferry, which are to be reported in reference [12]. For this purpose several subdivision arrangements and areas of damage orifices were taken into consideration.

5. Test calculations and parameter study

In order to gain experience with the practical application of the computer program as an instrument a parameter study and extensive test calculations on several existing ro-ro passenger ships have been carried out.

5.1 Parameter study

The main purpose of the parameter study is a sensitivity analysis of the most critical parameters in the mathematical model. The following parameters have been subject of a systematic investigation:

- damage orifice area: variation of width of hole
- initial mass moment of inertia of ship: variation of +/-25%
- drag coefficient for internal flow: variation of +/-50%
- area of air vent: variation of +/-50%
- presence of bilge keel: variation of roll damping

It turns out that the results, although not unexpected, are to a great extent affected by variations of damage orifice area as demonstrated in figure 1 in terms of a time simulation of the roll angle.
The other parameters, within the range indicated above, have however only a very minor effect on the maximum roll angle and the conclusion seems justified that the mathematical model is not very dependent on a high accuracy of these parameters.

In addition to this parameter study an investigation was carried out into the SOLAS-criteria in the final stage of flooding. For the damage scenario under consideration (non)-submergence of the margin line was by far the overriding criterion due to the fact that a high degree of asymmetry is present in this particular (non-SOLAS) one-compartment damage case. A decrease of KG_{max} or draught (see figure 2) shows respectively a substantial reduction (due to an excessive high GM-value) or a slight increase (due to a decrease in displacement) of the maximum roll angle which is defined as the first roll amplitude in time.

It should be noted that the residual stability in the intermediate stages of flooding does not present a safety problem because the car deck is not submerged during the intermediate stages of flooding.

5.2 Test calculations

For the purpose of providing a basis of comparison one damage orifice configuration, as given in figure 3, has consistently been used throughout all test calculations.

The results of a two-compartment damage of crossduct connected wingtanks of a pre-SOLAS 90 ro-ro passenger ship are presented in figure 4. The ship does survive this particular damage case although shipping of water occurs at the first roll amplitude. This minor amount of water remains trapped on the car deck due to the trim developed. Due to a relative high time constant for the flow through the crossduct it may be concluded that the equalizing arrangement does not significantly affect the first overshoot of the roll angle.

The result (in terms of a time history of the roll angle) of a two-compartment damage of wing compartments of a one-compartment standard SOLAS 90 ro-ro passenger ship (intended for the carriage of lorry drivers) is presented in figure 5. Likewise the ship does survive the assumed damage scenario although a greater angle of heel occurs in the final stage of flooding due to the absence of a cross-flooding arrangement.

The results of a two-compartment damage of engine rooms of a pre-SOLAS 90 ro-ro passenger ship are presented in figure 6. In this particular damage case the effects of a tweendeck and the main propulsion unit have been included in the model. It is obvious from figure 6 that the ship does not survive this particular damage case because shipping of water occurs at the first roll amplitude and the heeling moment due to flood water in the engine room's sub-compartments is so large that flooding of the car deck continues subsequently developing an ever-increasing list. It should be noted that the ship survives a one-compartment damage to only one engine room.

The result (in terms of a time history of the roll angle) of a two-compartment damage of engine rooms of a SOLAS 90 ro-ro passenger ship (one-compartment standard) for various effects of modelling is presented in figure 7. Likewise the ship does not survive the assumed damage scenario although flooding of the car deck occurs at the second roll amplitude. This means that the distribution of flood water in the engine room and the associated heeling moment is one major effect inducing a rapid capsize. In this respect it should be observed from figure 7 that the presence of a tweendeck is a deteriorating factor whereas the presence of a major obstruction is beneficial to some extent.

6. General discussion and conclusions

A mathematical model has been developed to describe the actual motion behaviour and the associated residual stability of a ship in the time domain after sustaining a collision damage. An outline of the mathematical model is presented in section 2 of the paper. The calculation method has been made effective by means of a computer program as described in section 3 of the paper. Validation experiments, as indicated in section 4 of the paper, have been carried out showing that the accuracy of the calculation method is such that the computer program may well be used as a prediction instrument.
The main conclusion of the study is that calculations carried out in accordance with existing international legislation do not represent a realistic picture of the actual flooding process of a damaged ro-ro ship after a high energy collision.

It has been demonstrated that the sudden ingress of water in the transient stages initiates and increases the heeling moment generated by flood water. This is particularly true for ro-ro ships which usually have a centre of gravity above keel in the range 1.5-2.0 times full load draught d.
The lever of the heeling moment at t=0 reads as follows:

$$(KG_{max}-d+z_A)z_A A_t/\nabla$$

where ∇ = volume of displacement
z_A = distance from centre of hydrostatic pressure of A_t to waterline
A_t = submerged area of the damage orifice which by nature is time-dependent and increases in a progressive manner during the first quarter of the first roll cycle assuming that the damage extends above the car deck

In this very elementary analytical expression the value of KG_{max}/d is very critical for e.g. car deck submergence at the first roll amplitude.

The test calculations, reported in section 5.2 of the paper, have shown that the application of a longitudinal subdivision may be very beneficial to avoid the rapid capsize phenomenon.
The compartment configuration which is unfavourable from this point of view is characterized by voluminous compartments without a longitudinal subdivision to bound the heeling moment of flood water. The effect becomes even more pronounced by the existence of tweendecks below the waterline, a high ship's centre of gravity relative to the waterline and obviously the occurrence of a relative low value of freeboard and/or GM in the intact condition.

Although not accounted for in the mathematical model it may be expected that a rapid capsize is preceded by shifting of cargo due to the occurrence of a substantal list for some time. The rapid

capsize process as such is a fatality and for this reason it is desirable to have pertinent knowledge on the probability of occurrence of a high energy collision.
For the purpose of this study a realistic damage orifice with a rather large area has been used, however in this respect statistical information for the assessment of the size of the damage orifice, either deterministic or in the probabilistic sense, is indispensable.

Acknowledgement

The authors would like to express their appreciation to H.J. Koelman of SARC BV who provided the software on ship hydrostatics for the generation of input data.

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Figures

<u>Figure 1</u> Time simulation of rolling angle for various damage orifice areas

	width (m)		height (m)
(1)	6.0	x	6.0
(2)	4.5	x	6.0
(3)	3.0	x	6.0
(4)	1.5	x	6.0
(5)	0.16	x	6.0

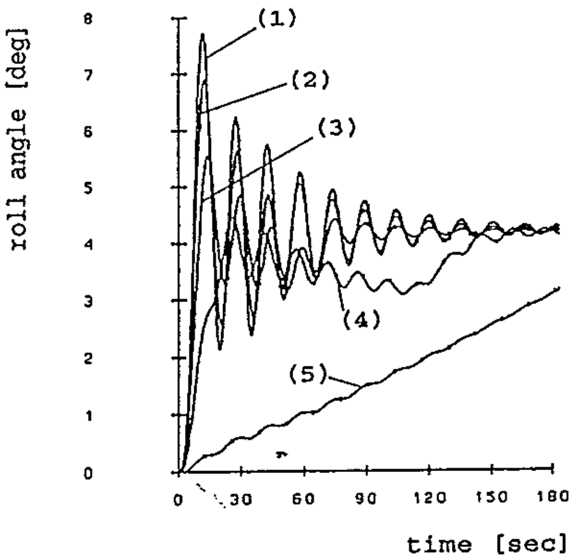


Figure 2 Maximum allowable KG as function of displacement based on the criterion of margin line submersion

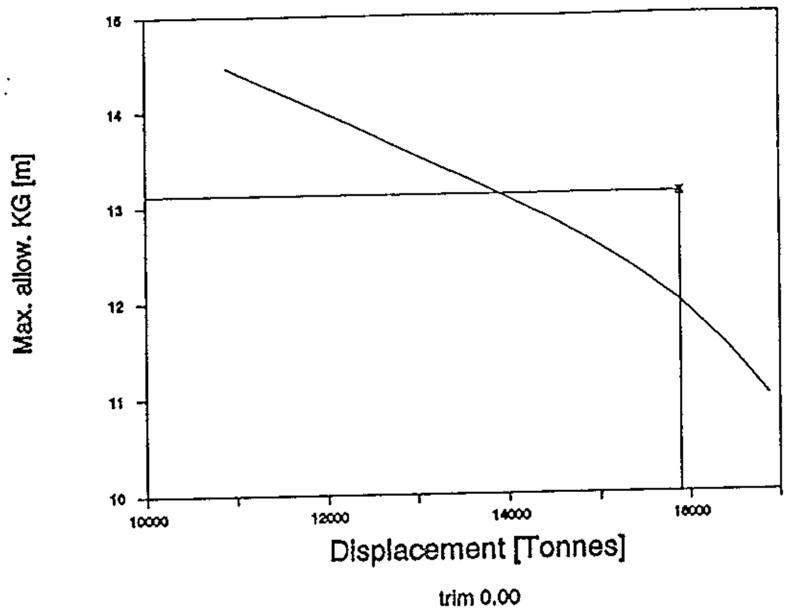


Figure 3: Configuration of damage orifice for the test calculations

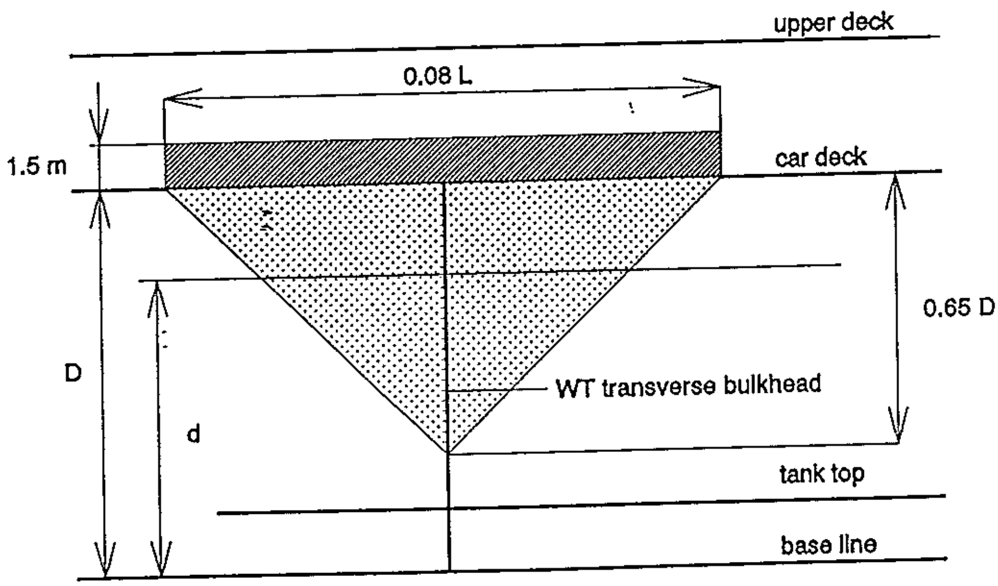


Figure 4: Time simulation of the behaviour of a pre-SOLAS 90 ro-ro passenger ship following a two-compartment damage of crossduct connected wingtanks

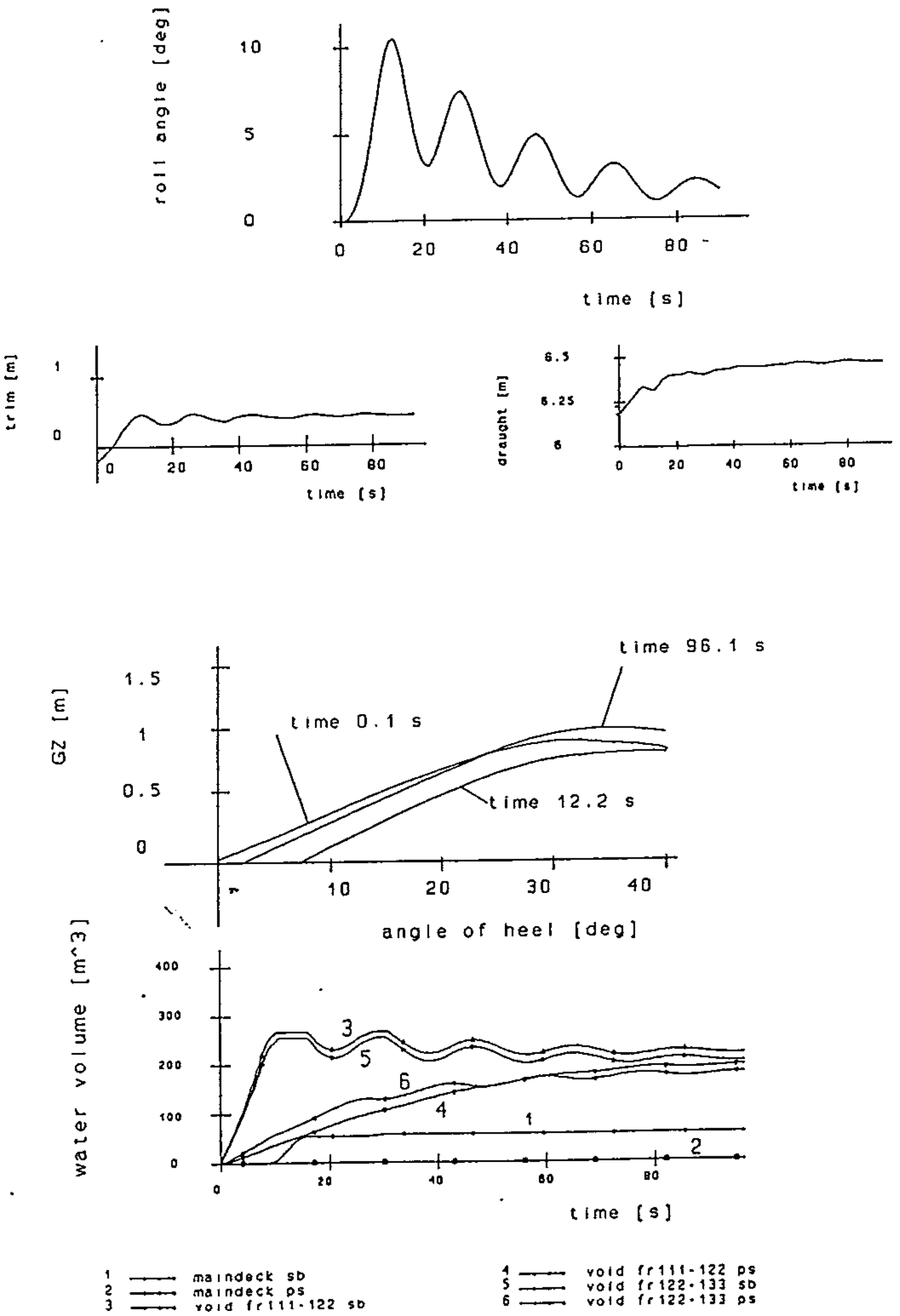


Figure 5: Time simulation of roll angle of a SOLAS 90 ro-ro passenger ship (one-compartment standard) following a two-compartment damage of void spaces in the side

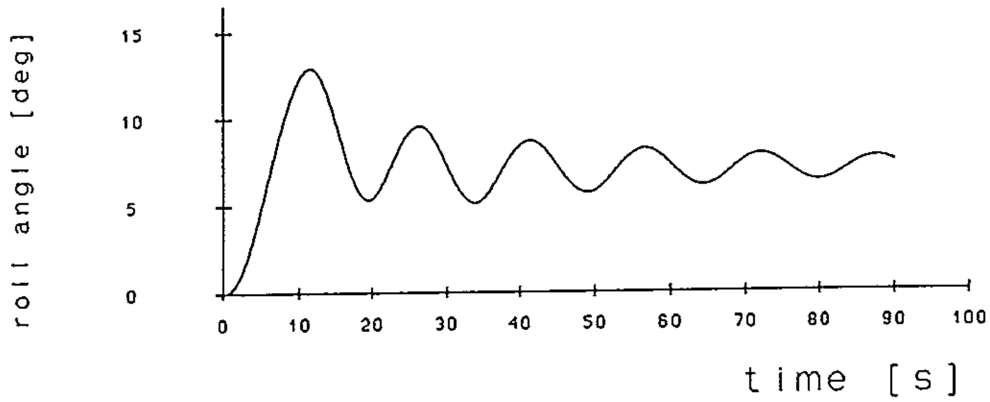


Figure 6: Time simulation of the behaviour of a pre-SOLAS 90 ro-ro passenger ship following a two-compartment damage of engine rooms

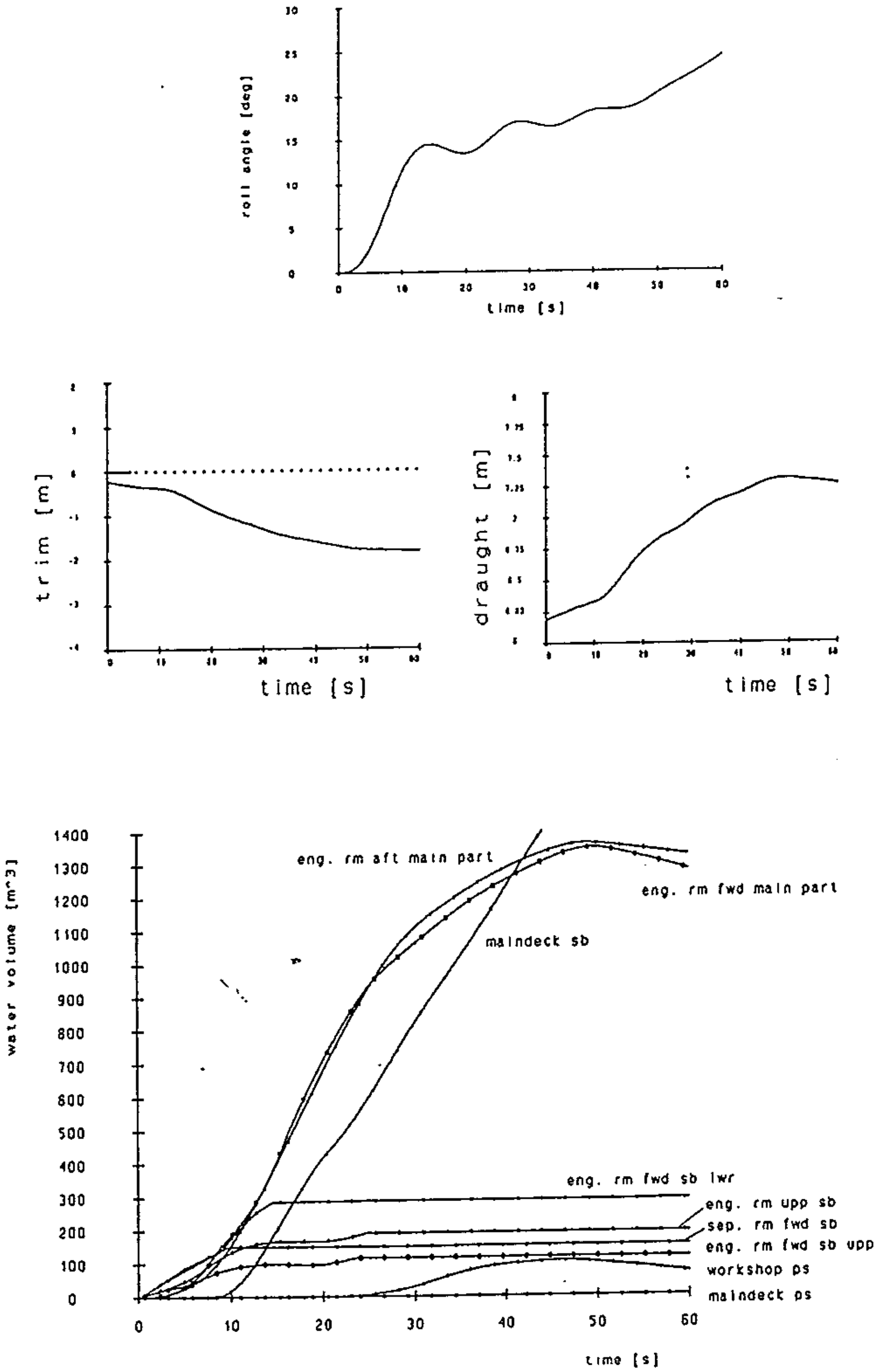


Figure 6: Time simulation of the behaviour of a pre-SOLAS 90 (contnd.) ro-ro passenger ship following a two-compartment damage of engine rooms

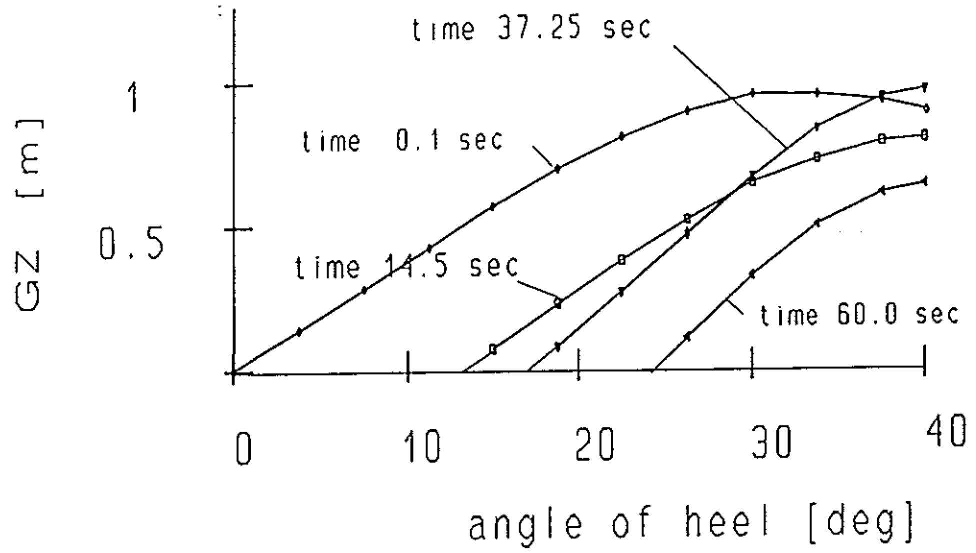
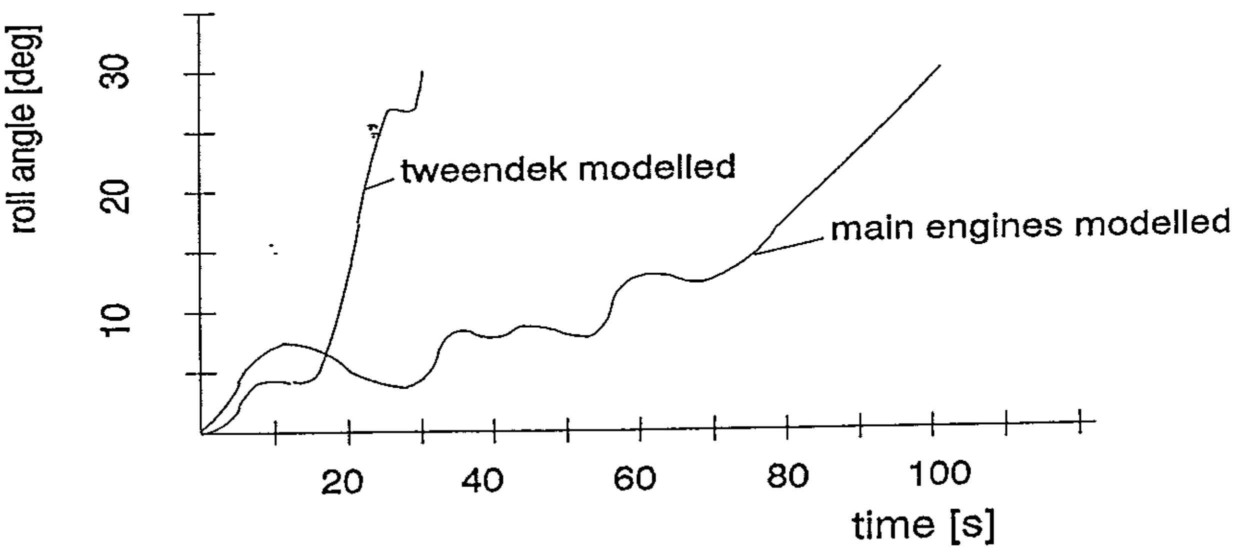


Figure 7: Time simulation of roll angle of a SOLAS 90 ro-ro passenger ship (one-compartment standard) following a two-compartment damage of engine rooms for various effects of modelling



PREDICTION OF MOTION OF SHIPS WITH FLOODED COMPARTMENTS IN A SEAWAY

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ABSTRACT

It is shown that in the presence of free surface the flooded water in compartments have an influence upon all dynamic characteristics of a damaged ship, i.e. inertia, motions damping, transverse stability and exciting forces. These circumstances are proposed to be taken into account by means of series of corrections for the relevant coefficients of dynamic equations. In order to calculate the corrections it is sufficient to have the information concerning the static corrections for metacentric height, response amplitude operators (RAO) for liquid sloshing relative to every compartment and the corresponding phase values. It is found that the satisfactory results can be obtained within the linear calculation scheme if dynamics of only fundamental forms of liquid volume oscillations is taken into consideration while damping of these oscillations is related to the energy dissipated in generation higher order waves on liquid surface and in hydraulic losses if any. Then the possibility of considering the compartment permeability factors is easily realized.

The method of corrections, apart from clear physical interpretation of influence of a flooded compartment on ship motions, is very convenient due to possibilities offered for application of one and the same algorithm for predicting the motions of a damaged ship both with the flooded compartments and without them.

1. INTRODUCTION

Most of the problems from the field of subdivision designs and of the development of measures for the restoration of stability and for equalization of a damaged ship are solved on the basis of a static approach. Corresponding information systems and documents supply a shipmaster with data on a static list and trim and stability of a ship in calm water under the various flooding combinations, but do not provide any information on these data changing during the ship motions in waves or on risk level for losing a damaged ship. An answer for that kind of quastions could be only obtained provided that a damaged ship response for external adverse effects in specific sea conditions is known. In relation with what has been said, theoretical and experimental research for such a ship dynamics and developing the calculation techniques for determination of ship behaviour and safety characteristics at given sea state gain great practical significance and vitality.

To the number of main features of a ship which keeps up buoyancy and positive stability being in a damaged condition, the following may be first of all ascribed: (a) presence of considerable water masses inside the hull; (b) an asymmetric distribution of weight load; (c) the asymmetry of a ship wetted surface due to a static list and trim appearance.

At present, the questions of ship dynamics under asymmetric weight load and form of a ship submerged volume are most completely analysed. Among the works of this direction the studies [3], [4], [5], [9], [20], [21] should be noted. These studies make it possible to calculate the damaged ship motions in a common case with a wide range of encounter frequencies taking into consideration an initial list and trim under an arbitrary heading of incident waves. In the studies the water in flooded compartments is considered as an additional solid mass which influences upon the ship weight and its distribution. In other words, only the alteration of ship's displacement and transverse stability is examined without the effect of the liquid free surface being taken into consideration.

In reality the effect of flooded water on ship motions is rather more complicated. Up to present day the questions of the dynamics of a ship with liquid cargoes have been studied in particular formulations mainly for the cases when the ship compartments do not link to the seawater [13], [15], [16], [23], [24].

One should also take note of the whole number of special experimental studies for separate questions of damaged ship models dynamics in waves. These studies were carried out at different times beginning from 1960 in Krylov Shipbuilding Research Institute [17], in Davidson Laboratory [11], in Kaliningrad Technical Institute of Fisherries [10], in National Phisycal Laboratory [2], in Hamburg [25]. A great number of original and interesting results were found but the generalization and explanation of these were intricate due to absence of a reliable mathematical model of motions of a ship with flooded compartments in waves.

The aim of the present study is the dicussion of peculiarities of the analytical calculation technique for predicting the forced motions of a ship with flooded compartments in regular waves. All cases of principal for flooded compartments are taken into cosideration. Among them there are the compartments of the 1-st category that are fuly flooded with water without a free surface left, and the compartments that are partially flooded with water which has a free surface. In the 1-st case two kinds of compartments are considered, namely: the compartment of the 2-nd category which have no link to the seawater, and the compartment of the 3-nd category which have bottom or side damages.

The discussed techniques are used for calculation of damaged ship motions in beam incident waves. In addition it is assumed that neutral heel and trim angles are zero as much as the influence of the latter upon ship motions was examined in details [9]. The technique is based on the assumption of a linear character of incident waves and the motions of both the ship and the flooded water in her compartments caused by those waves. It is believed, the linear analisis makes it possible to distinguish the main features of flooded water effect upon damaged ship dynamics. If necessary the most essential nonlinear factors could be taken into consideration in future.

2. EQUATIONS OF MOTION FOR A SHIP WITH FLOODED COMPARTMENT IN BEAM REGULAR WAVES AT NO FORWARD SPEED

To derive equations of motion, a ship is considered as a solid body with compartment flooded partially or fully by the seawater. For the purposes of simplification the permeability of compartment spaces shall be taken as 100%. Absolute liquid motion in a compartment is supposed as potential as well as mass forces which affect the particles of this inner liquid. The potential field of the mass forces is created due to sway and heave linear accelerations and angular rolling accelerations. A ship supposed to be drifting beam to the seas with no wave yawing. It is assumed that ship motions and liquid sloshing in flooded compartment are linear.

Finally, wave-excited motion equations of a damaged ship in transverse vertical plane are derived by means of the Lagrange equations of the 2-nd kind. In such a case the kinetic energy relationship for the ship includes kinetic energy of liquid sloshing which is described by means of generalized independant coordinates in the form having been suggested by Moiseev [12].

The frame of reference for body motion is a right handed Cartesian coordinate system originating at the ship center of gravity in its mean equilibrium condition, OZ - axis is directed vertically downward and OY - axis is to starboard. For dynamic consideration of liquid sloshing in a flooded compartment it is used additional coordinate system $O\eta\zeta$ originating at the center of symmetry of liquid free surface in calm condition S_0 (see Fig.1). The ship center of gravity coordinates (G-coordinates) are denoted in this system as η_g, ζ_g.

Under these assumptions the equations of ship motions in the transverse plane discussed in [16] can be easily adapted to the case of the 3-nd category flooded compartment and expressed as follows:

$$(M + A_{22})\ddot{\xi}_2 + B_{22}\dot{\xi}_2 + A_{24}\ddot{\xi}_4 + B_{24}\dot{\xi}_4 - \gamma\sum_{m=1}^{\infty}\beta_m \frac{\omega^2}{g} q_m(t) = F_2^c\cos(\omega t) + F_2^s\sin(\omega t) \tag{1}$$

$$(M + A_{33})\ddot{\xi}_3 + B_{33}\dot{\xi}_3 + C_{33}\xi_3 = F_3^c\cos(\omega t) + F_3^s\sin(\omega t) \tag{2}$$

$$A_{42}\ddot{\xi}_2 + B_{42}\dot{\xi}_2 + (I_x + A_{44})\ddot{\xi}_4 + C_{44}\xi_4 - \gamma S_0\eta_g q_{0r}(t) + \gamma\sum_{m=1}^{\infty}\beta_m\left(1 + \frac{\delta_m}{\beta_m}\cdot\frac{\omega^2}{g}\right)q_m(t) = \tag{3}$$

$$= F_4^c\cos(\omega t) + F_4^s\sin(\omega t)$$

where, ξ_i - the displacement of the ship from her mean position in the i-th mode of motion, i.e. sway (i=2), heave (i=3) and roll (i=4). Coefficients A_{ik} are described by added mass matrix, B_{ik} - by the damping matrix, C_{ik} - by the restoring matrix, F_k^c and F_k^s - the amplitudes of the cosine and sine wave-exciting force or moment components in the k-th mode of motion, ω - the wave-encounter frequency. All the above coefficients are defined in accord with the recommendations of linear theory of ship motions either for an intact ship [19] for the case of symmetrically flooded ships or for a damaged ship [9] in the case of asymmetrical flooding. The coefficients M and I_x denote mass of a ship and her central inertia mass moment which shall take into account the mass of the flooded water and its motion for the case of assumed absence of the free surface [22]. The dots indicate the time derivatives of the variables.

The equations (1)-(3) differ from the equations of motions of the ship with a flooded compartment of the 2-nd category by the presence of an additional term

$$\Delta L = \gamma S_0 q_{0r}(t)\eta_g \tag{4}$$

Here $q_{0r}(t)$ denotes one of the principal generalized coordinate which characterizes the periodical fluctuation of a mean flooded water level in a compartment if the latter is linked to the seawater body. The formula (4) defines the heeling moment due to the weight force of an additional flooded water mass flowing in and out of the compartment trough its damage as a result of ship motions and waves.

The time-dependent functions $q_m(t)$ in (1)-(3) are the other principle coordinates of the boundary value problem which characterizes the relative motion of water in a flooded compartment. The coordinate $q_m(t)$ defines the m-th form of the liquid free surface oscillations. Theoretically the number of these forms is infinite.

3. FORMULATION OF THE LINEAR PROBLEM OF FORCED LIQUID SLOSHING IN A SHIP COMPARTMENT

3.1. General

To solve the equations (1)-(3) it is necessary to define relative motion of a ship and a liquid mass in the ship compartment. The velocity potential of a such relative motion φ must, on one hand, satisfy Laplace's equation

$$\nabla^2\varphi = 0 \tag{5}$$

in liquid volume τ and, on the other hand, meet the conditions at the liquid boundaries, i.e. at the wetted surface Σ of a compartment

$$\frac{\partial\varphi}{\partial n} = 0$$

and at the free surface S_0 the condition of pressure equality which follows from the general form of the Lagrange-Cauchy integral. In linear formulation this condition can be written as follows:

$$\frac{\partial\varphi}{\partial t} + \mu\varphi + E = 0 \tag{6}$$

where the function E represents the potential of mass forces affecting the volume τ and the parametr μ is a coefficient in linear damping law of liquid sloshing.

If a ship compartment is heeled at an angle ξ_4 and fixed, the potential E shall be expressed as

$$E = -g\zeta - g\eta\xi_4$$

If a compartment is oscillating due to ship motions, the liquid volume τ shall be affected, besides the gravity force field, by the fields of linear and angular accelerations. As a result the potential E shall be of more complicated form and can be given by

$$E = -g\zeta - g\eta\xi_4(t) + \varphi_\omega\ddot{\xi}_4(t) + \ddot{\xi}_2(t)\eta + \ddot{\xi}_3(t)\zeta \tag{7}$$

where φ_ω is the Zhoukowsky function which describes liquid motion due to compartment rotation for the case of no free surface. This function depends on compartment form, i.e. on variables η and ζ in the limits of space τ considered [22].

3.2. Liquid sloshing in a 2-nd category compartment

Substituting (7) to (6) and assuming that $\ddot{\xi}_3 \ll g$ the linear boundary-value problem for liquid sloshing in a compartment caused by the ship motions in a transvers plane can be formulated as follows:

$$\nabla^2\varphi = 0 \qquad \text{within space } \tau \tag{8}$$

$$\frac{\partial\varphi}{\partial t} + \mu\varphi - g\tilde{\zeta}(t) = -\eta\ddot{\xi}_2(t) + g\eta\xi_4(t) - \varphi_\omega\ddot{\xi}_4(t) \qquad \text{on } \zeta=0 \tag{9}$$

$$\frac{\partial\varphi}{\partial\zeta} = \frac{\partial\tilde{\zeta}}{\partial t} \qquad \text{on } \zeta=0 \tag{10}$$

$$\frac{\partial\varphi}{\partial n} = 0 \qquad \text{on the surface } \Sigma \tag{11}$$

This problem appears to be a generalization of the problem solved by Moiseev [12], [13] in a way of taking into account the liquid sloshing damping and the sway motions of a ship in waves.

The solution of the problem is obtained using the form which have been suggested in [12]:

$$\varphi = \sum_{n=0}^{\infty} \dot{q}(t)\,\frac{\varphi_n(\eta,\zeta)}{\lambda_n} \tag{12}$$

$$\tilde{\zeta} = \sum_{n=0}^{\infty} q_n(t)\varphi_n(\eta,0) \tag{13}$$

where $\varphi_m(\eta,\zeta)$ and λ_m are the eigenfunctions and the eigenvalues of the volume τ correspondingly. As a result, the problem passes over to an infinite number of differential equations for the time-dependent Fourie coefficients $q_m(t)$ in the expansion (13) of the free surface elevation $\tilde{\zeta}(\eta,t)$. These equations are solved together with ship motion equations (1)-(3). In this case for even forms of liquid sloshing motion (m=0, 2, 4, ...) the equations are given as

$$\ddot{q}_m + 2b_m\dot{q}_m - g\lambda_m q_m = 0 \tag{14}$$

and for odd forms (m=1, 3, 5, ...) as

$$\ddot{q}_m + 2b_m\dot{q}_m - g\lambda_m q_m = -\beta_m\lambda_m\ddot{\xi}_2 + g\beta_m\lambda_m\xi_4 - \delta_m\lambda_m\ddot{\xi}_4 \tag{15}$$

where β_m and δ_m are the Fourier coefficients in the expansions of the coordinate η and the function φ_ω on the undisturbed free surface $\zeta=0$, so that

$$\eta = \sum_m \beta_m\varphi_m(\eta,0) \quad , \quad (m=1,\ 3,\ 5,\ \ldots) \tag{16}$$

$$\varphi_\omega = \sum \delta_m\varphi_m(\eta,0) \quad , \quad (m=1,\ 3,\ 5,\ \ldots) \tag{17}$$

Thus, equations (15) characterize the main forms of a free surface sloshing motion and show that the water in a flooded compartment of the 2-nd category represents an oscillatory system of an infinite number of freedom degrees. Each of the latter possesses of its natural frequency

$$\sigma_m^2 = -g\lambda_m \tag{18}$$

and damping

$$2b_m = \frac{\mu}{g} \tag{19}$$

For a prismatic compartment of rectangular form, it is known

$$\sigma_m^2 = \frac{g}{2d}\, m\pi \cdot \mathrm{th}\left(m\pi\,\frac{f}{2d}\right) \tag{20}$$

where 2d is the compartment width and f is the compartment flooding depth. The choice of a compartment form of that kind is very convenient for practical use.

From the formulas (14)-(17) given above, it follows that the ship motions excite in a flooded compartment of the 2-nd category only odd forms of liquid volume oscillations. The periodic shift of a gravity centre of this volume in the plane of its oscillations is characteristic for the latter. An integral influence of the flooded water on a whole vessel in the case will be mainly determined by the first odd form of sloshing motion $q_1(t)$, which characterizes a plane rotation of the free liquid surface relative to the compartment. This is confirmed by the expansion of (16), the first coefficient β_1 of which within the precision limits of about 2% determines the intrinsic moment of inertia of the free surface S_0 [16]:

$$i_\xi = \sum_{n=0}^{\infty}\beta_{2n+1}^2 \cong \beta_1^2 \tag{21}$$

What have been said is of great importance since it makes possible to limit practical calculations to the consideration of the first harmonic of a liquid cargo sloshing only.

3.3. Liquid sloshing in a 3-rd category compartment

This problem can be represented as generalization of the previons one, if we assume the existence of a damage with an area S_r in the surface Σ through which the seawater may enter and leave the compartment. The field of pressures within the compartment in this case will alter on a difference of:

$$\Delta E = -g\tilde{\zeta}_R + (\eta - \eta_g)\xi_4 \quad (22)$$

where $\tilde{\zeta}_R$ is a relative heaving motion. The normal velocity $\partial\varphi/\partial n$ will not be zero within the damage area S_r

$$\frac{\partial\varphi}{\partial n} = V_{or}(\eta, \zeta) \cdot \overline{V_r(t)} \quad (23)$$

Here $\overline{V_r(t)}$ is the velocity of the seawater flowing into and out of a compartment and averaged in the limits of the damage area S_r.

As a result, the boundary-value problem for a 3-rd category compartment is formulated as follows:

$$\nabla^2\varphi = 0 \quad , \quad \text{in } \tau \text{ domain;} \quad (24)$$

$$\frac{\partial\varphi}{\partial t} + \mu\varphi - g\tilde{\zeta}(t) = -\eta\ddot{\xi}_4(t) + g\eta\xi_4(t) - \varphi_m\ddot{\xi}_4(t) + g\tilde{\zeta}_R(t) - (\eta - \eta_g)\xi_4(t)$$

on the free surface $\zeta=0$ (25)

$$\frac{\partial\varphi}{\partial\zeta} = \frac{\partial\tilde{\zeta}}{\partial t} \quad \text{on the free surface } \zeta=0 \quad (26)$$

$$\frac{\partial\varphi}{\partial n} = 0 \quad \text{on the surface } \Sigma \quad (27)$$

$$\frac{\partial\varphi}{\partial n} = V_{or}(\eta, \zeta)\ddot{V}_r(t) \quad \text{on the surface } S_r \quad (28)$$

The solution of the problem is achieved by means of the principal coordinates' method in a form similar to what is proposed in [12]:

$$\varphi = \dot{q}_{or}(t)\,\frac{\varphi_r(\eta, \zeta)}{\lambda_{or}} + \sum_{m=1}^{\infty} \dot{q}_m(t)\,\frac{\varphi_m(\eta, \zeta)}{\lambda_m} \quad (29)$$

$$\tilde{\zeta} = q_{or}(t)\varphi(\eta, 0) + \sum_{m=1}^{\infty} q_m(t)\varphi_m(\eta, 0) \quad (30)$$

Here the coefficients $q_i(t)$ appear to be generalized coordinates of the problem (24)-(28) and are defined by an infinite number of independent equations.

- For the coordinate $q_{or}(t)$ which represents an average water level change in a compartment:

$$\ddot{q}_{or} + 2b_{or}\dot{q}_{or} - g\lambda_{or} = -g\lambda_{or}\tilde{\zeta}_R - g\lambda_{or}\eta_g\xi_4 - \lambda_{or}\beta_{or}\dot{\bar{V}}_r \quad (31)$$

- for the $q_m(t)$ with even numbers (m=2, 4, ...):

$$\ddot{q}_m + 2b_m\dot{q}_m - g\lambda_m q_m = -\lambda_m\alpha_m\dot{\bar{V}}_r \tag{32}$$

Where α_m are the coefficients of the function $V_{or}(\eta,\zeta)$ expansion on the surface $S_0(\zeta=0)$ into Fourier series on the basis of the problem eigenfunctions

$$V_{or} = \sum_{m=0}^{\infty}\alpha_m\varphi_m(\eta,0) \tag{33}$$

- for the $q_m(t)$ with odd numbers (m=1, 3, 5, ...):

$$\ddot{q}_m + 2b_m\dot{q}_m - g\lambda_m q_m = -\beta_m\lambda_m\ddot{\xi}_2 + \beta_m g\lambda_m\xi_4 - \delta_m^*\lambda_m\ddot{\xi}_4 \tag{34}$$

where β_m and λ_m are already known coefficients from the problem (8)-(11) and the coefficient

$$\delta_m^* = \delta_m - \beta_m l_{or} \tag{35}$$

where l_{or} value is explained further on. It is necessary to note that the coefficient δ_m characterizes the mutual vertical position of the ship rotation axis Gx and mobile masses of liquid [6]:

$$\delta_m = \beta_m(\zeta_g - \zeta_m) \tag{36}$$

As one can see in the general case, the seawater that has entered into the 3-rd category compartment, if there is the free surface,, manifests itself as the oscillatory system similar to a such one from the problem (8)-(11). An essential feature of this system is the presence of even oscillatory forms $q_{2m}(t)$ apart from already established odd ones $q_{2m+1}(t)$. The main principal even form (31) reflects the fact of water flowing into and out of a compartment through the damage S_r.

The natural frequency of this oscillatory motion depends on the effective length l_{or} of the involved-into-motions liquid volume τ_{or} which takes into account the added mass of the seawater in a volume to be estimated in accord with a shape of the damage area

$$\sigma_{or}^2 = -g\lambda_{or} = \frac{g}{l_{or}} \tag{37}$$

As one may see from [8] in the case of liquid mass oscillations considered, the damping of the latter may be linked with the wave generation at the external free surface. The engineering estimates of the damping characteristics and the natural frequency σ_{or} one may find in [18] where the case of the 3-rd category flooding through the bottom damage is examined.

4. DYNAMIC QUALITIES OF THE SHIP WITH A FLOODED COMPARTMENT

Dynamics of a damaged ship drifting beam to regular waves is described in accord with the assumptions mentioned earlier by means of three linear equations of ship motions (1)-(3) and by infinite system of linear equations of the kind (31)-(34) which shall define the generalized principal coordinates $q_{or}(t)$ and $q_m(t)$ from the boundary value problem (24)-(28).

The mentioned set of the equations is equitable to the most common case of flooding when the flooded compartment is linked to the seawater and classified as the 3-rd category ($S_r \neq 0$). If the coordinate $q_{or}(t)$ is supposed to be zero in these equations one will meet the case of the 2-nd category flooding ($S_r = 0$). And at last, if it is assumed that both kinds of coordinates $q_{or}(t)$ and $q_m(t)$ are identically equal to zero, i.e. there is no free surface in the flooded compartment, the latter will be of the 1-st category. The liquid motion in such a case has been examined by Zhoukowsky [22] who has shown that the inertia ellipsoid of a solid body with an inner cavity filled in with liquid lies inside of the analogous ellipsoid for the same solid body but including the iced liquid cavity restricted by the same surface Σ. On the strength of that it can be written

$$I_x = I_{x0} + \int_{\Sigma} \varphi_\omega \frac{\partial \varphi_\omega}{\partial n} dS = I_{x0} + i - \Delta i_\omega \tag{38}$$

where I_{x0} and i are the moments of mass inertia for a damaged ship without the flooded water and for the flooded water mass only relative to the general central longitudinal axis Gx, and Δi_ω is the negative correction taking into account the liquid mobility [16]:

$$\Delta i_\omega = 4M_r d^2 \Gamma\left(\frac{f}{d}\right) \tag{39}$$

Here M_r is the flooded water mass, 2d the characteristic transverse size of a compartment, f the characteristic liquid depth. The function $\Gamma(f/d)$ is defined by the compartment form [22].

In the given formulation heaving motion (2) does not depend on liquid mass dynamics. Rolling motion (3) and sway (1) become additionally coupled due to vertical $q_{0r}(t)$ and crossover $q_{2m+1}(t)$ motions of liquid mass in a compartment.

Excluding heave equation as uncoupled with the other modes of ship motions in the case considered, the equations (1)-(3) and (31)-(34) may be put in the following canonical form:

$$\begin{aligned}&\left(M + A_{22} - \tilde{\Delta}M\right)\ddot{\xi}_2 + \left(B_{22} - \tilde{\Delta}B_{22}\right)\dot{\xi}_2 + A_{24}\ddot{\xi}_4 + B_{24}\dot{\xi}_4 = \\ &= F_2(t) + \tilde{\Delta}F_2(t)\end{aligned} \tag{40}$$

$$\begin{aligned}&A_{42}\ddot{\xi}_2 + B_{42}\dot{\xi}_2 + \left(I_x + A_{44} - \tilde{\Delta}I_x\right)\ddot{\xi}_4 + \left(B_{44} - \tilde{\Delta}B_{44}\right)\dot{\xi}_4 + \gamma\left[V \cdot GM - \tilde{\Delta}(V \cdot GM)\right]\xi_4 = \\ &= F_4(t) + \tilde{\Delta}F_4(t)\end{aligned} \tag{41}$$

where symbol $\tilde{\Delta}$ denotes the correction to an appropriate coefficient.

It is seen that principally a flooded compartment affects not only ship's statics but also the most of dynamic characteristics of a damaged ship governing her motions. Additional terms in the right-hand side of equations (40) and (41) describe the dynamic effects of the liquid mass which reflect the coupling of ship motions due to the creation of an internal free surface. These effects are classified as additional exciting forces.

The relations are given below as an illustration reffering to ship rolling for the most general case of the 3-rd category flooding when the centre of the compartment space does not coincide with the ship centreplane ($\eta_r \neq 0$). These relations characterize :

a) Correction of stability coefficient

$$\begin{aligned}\tilde{\Delta}(V \cdot GM) = {} & V\tilde{\Delta}GM_0\left[\left(1 + \frac{\omega^2 l_{0r}}{g}\right)K_{0r}(\omega)\right] \cdot \cos\left(\varepsilon_{0r}\right) + \\ & + V\tilde{\Delta}GM_1\left[\left(1 + \frac{\omega^2}{g} \cdot \frac{\delta_1^*}{\beta_1}\right)K_1(\omega)\right] \cdot \cos\left(\varepsilon_{q_1}\right)\end{aligned} \tag{42}$$

b) Correction of roll damping coefficient

$$\tilde{\Delta}B_{44} = \gamma V\tilde{\Delta}GM_0\left[\left(1 + \frac{\omega^2 l_{or}}{g}\right)K_{or}(\omega)\right]\frac{\sin(\varepsilon_{or})}{\omega} + \\ +\gamma V\tilde{\Delta}GM_1\left[\left(1 + \frac{\omega^2}{g}\cdot\frac{\delta_1}{\beta_1}\right)\left(1 + \frac{\omega^2}{g}\cdot\frac{\delta_1^*}{\beta_1}\right)\right]\frac{\sin(\varepsilon_{q_1})}{\omega} \tag{43}$$

The formulas make allowance for the fact that the intrinsic moment of inertia of the free water surface S_0 in a flooded compartment within the precision limits 2% is defined by the relation (21). The value $\tilde{\Delta}GM_0$ represents a correction to the metacentric height due to the transient moment of inertia of the area S_0 i.e. $\tilde{\Delta}GM_0 = S_0\eta_s^2/V$. The meaning of the correction $\tilde{\Delta}GM_1$ is obvious (21) $\Delta GM_1 = i_t / V$. The symbol V denotes the volume displacement of a damaged ship including the volume of a flooded compartment.

In formulas (42)-(43) dynamic behaviour of the liquid mass is reflected in the response amplitude operators:

$$K_{or}(\omega) = \frac{\sigma_{or}^2}{\sqrt{(\sigma_{or}^2 - \omega^2)^2 + 4b_{or}^2\omega^2}} \tag{44}$$

$$K_1(\omega) = \frac{\sigma_1^2}{\sqrt{(\sigma_1^2 - \omega^2)^2 + 4b_1^2\omega^2}} \tag{45}$$

and in the phase-frequency responses:

$$tg(\varepsilon_{or}) = -\frac{2b_{or}\omega^2}{\sigma_{or}^2 - \omega^2} \quad ; \quad tg(\varepsilon_{q_1}) = -\frac{2b_1\omega}{\sigma_1^2 - \omega^2} \tag{46}$$

5. LIQUID MOTION DAMPING IN A FLOODED COMPARTMENT FOR THE MAIN OSCILLATORY FORMS OF ITS FLUID DOMAIN

On the whole the question of evaluation of liquid motion damping in a flooded compartment is very complicated. At present time there is no general solution of the latter, where all physical factors affecting such a motion energy dissipation are taken into consideration with equal completeness. However, from general considerations, one could assume the existance of at least three liquid motion damping components for an arbitrary case of the 3-rd category compartment. The compartment damping may be explained by: (a) the boundary surface friction, (b) the hydraulic losses and (c) wave generation on inner and external free surfaces.

A few studies and an everyday experience show that the friction damping for light liquids and water is relatively small. The hydraulic damping must depend upon a flow compression factor $n=S_7/S_0$ and an angle α between the seawater flow directions in the damage area S_7 and in the undisturbed free surface area S_0.

For the case of a damage located in the ship bottom the damping of liquid average level oscillatory change of (31) in the compartment will be determined by energy losses due to wave generation on the external seawater free surface [8] if one assumes the areas S_7 and S_0 to be equal and takes into account that an angle $\alpha=0$. The relative damping coefficient in this case may be estimated according to formula [18]:

$$\nu_{or} = \frac{b_{or}}{\sigma_{or}} = \frac{1}{4} \cdot \frac{\varepsilon}{K_1} \cdot \frac{1}{1 + \frac{\pi}{4} \cdot \frac{\varepsilon}{K_1}} \tag{47}$$

where $K_1=f/2d$ and $\varepsilon=2$. As it follows from the results [8], [18] at such damping the average level change $q_{or}(t)$ in the limits of the resonance frequencies will exceed its static displacement (44).

For the case of a damage below the free surface, but in a side, the compartment damping increases substancially due to hydraulic components ($\alpha \neq 0$ and $n < 1$). The experimental estimates of the total compartment damping on the basis of measuring the phase difference between the oscillatory level change q_{or} and model rolling in waves [7] give the meanings of the linearized damping coefficient of the following order:

$$\nu_{or} = \frac{b_{or}}{\sigma_{or}} \cong \frac{1}{2} \tag{48}$$

under the conditions that $\alpha=90°$ and $n=0, 45$. In the case of $n<0.4$ the compartment damping increases significantly and the inner water level change q_{or} in all frequency range will be less than the static displacement (44).

In the 2-nd category flooded compartment the hydraulic damping in liquid sloshing is excluded. The compartment damping in this case will be practically determined by wave phenomena. For the first main oscillation form q_1 the damping will be related to energy dissipation due to forming secondary waves (subharmonics) which depend on amplitudes of compartment motions and make a free surface be curved and breaking down. As it follows from the results of [15], the relative damping coefficient ν_1 in the case considered lies within the limits of:

$0.1<\nu_1< 0.33$

and depends on the ratio of frequencies ω and σ_1 and ship motions amplitude.

6. THE PERMEABILITY COEFFICIENTS EFFECT ON DYNAMICS OF FLOODED WATER

As it has been demonstrated, for the estimation of a damaged ship dynamics it is important to know the natural frequencies of oscillations of the water and the compartment damping of these oscillations. These data are significant for the determination of phase shifts between ship motions and liquid sloshing (46). The flooded compartment permeability has a visible effect on both dynamic parameters of the flooded water. One should in this case note the difference between the volume permeability μ_v and the surface one μ_s.

If $\mu_s=1$ one can easily find that

$$\sigma_{1\mu}^2 = \mu^2\sigma_1^2 \tag{49}$$

$$b_{1\mu} = \mu^{-2}b_1 \tag{50}$$

If $\mu_s=\mu_v=\mu$, then the relationship (50) stays as it is but the frequency will not be dependent on a permeability coefficient any more, i.e. $\sigma_{1\mu}=\sigma_1$.

7. CALCULATION RESULTS AND DISCUSSIONS

A prismatic model of a rectangular form and with curved bilge-angles, which was chosen for calculations, had been tested in one of the towing tanks of the Krylov Institute in beam regular waves [17]. The inner model space was divided into three parts, two of which contained three watertight compartments each and one contained two of the latter (Fig.2). In the course of the tests the middle compartments were filled in with water in various combinations reflecting the cases of the 1-st and the 2-nd category of flooding in relation to the water depth f. The side tank was filled once in the case when one of the middle compartments was tested with a side damage (Fig.2b). The principal characteristics of the model are given in Table 1.

Table 1

Principal characteristics of the model without flooded water

Length L	2,56 m
Breadth B	0,576 m
Draught	0,165 m
Volume displacement V	0,231 m
Block coefficient C_B	0,95
Centre of buoyancy KB	0,82 m
Centre of gravity KG	0,165 m
Metacentric height GM	0,10 m
Transverse radius of gyration	0,36 B

Calculations were made for all three categories of flooding in order to illustrate the possible effect of the flooded water on the ship stability and rolling motion. The model displacement and location of her centre of gravity KG in this case stayed unvariable.

Fig.3 demonstrates the 1-st category flooded compartments' effect upon the roll response amplitude operator (RAO) of the model $K_\phi(x)$, which determines the ratio of roll amplitude ϕ_0 to the amplitude of an effective wave slope angle [1] as a function of relative encounter frequency $x=\omega/\sigma_\phi$ where σ_ϕ denotes the roll natural frequency of the model without flooded water. As one can see (Fig.3), for the considered case of flooding the increase of the roll natural frequency

$$\sigma_\varepsilon^2 = \gamma \frac{V(GM - \tilde{\Delta}GM)}{I_x + A_{44}} \tag{51}$$

and of the relative damping coefficient

$$2\nu_\varepsilon = \frac{B_{44} - \tilde{\Delta}B_{44}}{I_x + A_{44}} \cdot \frac{1}{\sigma_\varepsilon} \tag{52}$$

takes place due to the efficient inertia moment (38) decrease. It is obvious that in this case $\tilde{\Delta}GM = \tilde{\Delta}B_{44} = 0$. The coefficient B_{44} for the unflooded model has been taken from the experimental data.

The calculation results for the 2-nd category flooded compartments are illustrated in Fig.4. In this case the corrections must differ from zero

$$\tilde{\Delta}GM \neq 0 \quad ; \quad \tilde{\Delta}B_{44} \neq 0$$

and consequently variations of the roll natural frequency σ_ε and the roll damping coefficient are dictated not only by decrease of the inertia moment I_x but also by corrections (42) and (43). As it is seen the correlation of computed results and experimental data from [17] is quite acceptable.

In Fig.5 an attempt is made to compare the computed results with experimental data for the model with a damaged side. The comparison of results does not give full correlation, although the RAO $K_\phi(x)$ maximum location at the frequency axis is evaluated correctly. The case requires a subsequent study. It is quite possible that the additional roll damping components shall be considered if the next step in analysis of ship motions is made taking into account the 3-rd category compartments with damaged side.

In Fig.6 the calculation results for the 3-rd category compartment with a damaged bottom are shown for the case when $S_f/S_0=1$ and the compartment centre of symmetry does not coincide with a ship centre-plane ($\eta_f \neq 0$). The value η_f has a direct effect on the correction $\tilde{\Delta GM_0}$ and its increase leads to roll natural frequency decrease (42), (51), from one side, and to relative roll damping increase (43), (52), on the other side. All this is reflected in Fig.6.

In the last Fig.7 the maximum value of roll RAO $K_{\phi max}$ is presented against the relative compartment damping coefficient ν_{or} for the case of the ratio $2\eta_r$ over B to be equal 0.4. The curve under discussion has a minimum value near

$$\nu_{or} \cong \frac{1}{\sqrt{2}}$$

when the average water level change in a compartment becomes less than static displacements in all frequency range. In the same Fig.7 the scale of relative bottom damage area S_r/S_0, taken from data of [7], is shown additionally. The correlation of the latter with the coefficient ν_{or} seems to be obvious, but the possibility of generalization of the data [7] on the case of a damage of arbitrary form and sizes require further research.

8. CONCLUDING REMARKS

From the present study, the following findings can be noted:

1. The presence of considerable liquid masses in ship compartments may produce an essential effect on her dynamical characteristics. This effect manifests itself in decrease of mass inertia and stability of a ship, in increase of ship motions damping and in exciting forces change.

2. The proposed method of corrections allows to estimate analytically the wave-excited motions of a ship with compartments of any category of their flooding. In the case of partially flooded compartments, it is necessary to note that roll and sway motions of a damaged ship become additionally coupled because of free surface dynamic effect.

3. To calculate dynamic characteristics of a damaged ship, such as: effective mass inertia, stability, motions damping and exciting forces, it is sufficient in practice to take into consideration only two first forms of liquid volume motions (the even and the odd ones) in the flooded compartment of the 3-rd category, and only the first odd form of sloshing motion in the flooded compartment of the 2-nd category.

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Figures

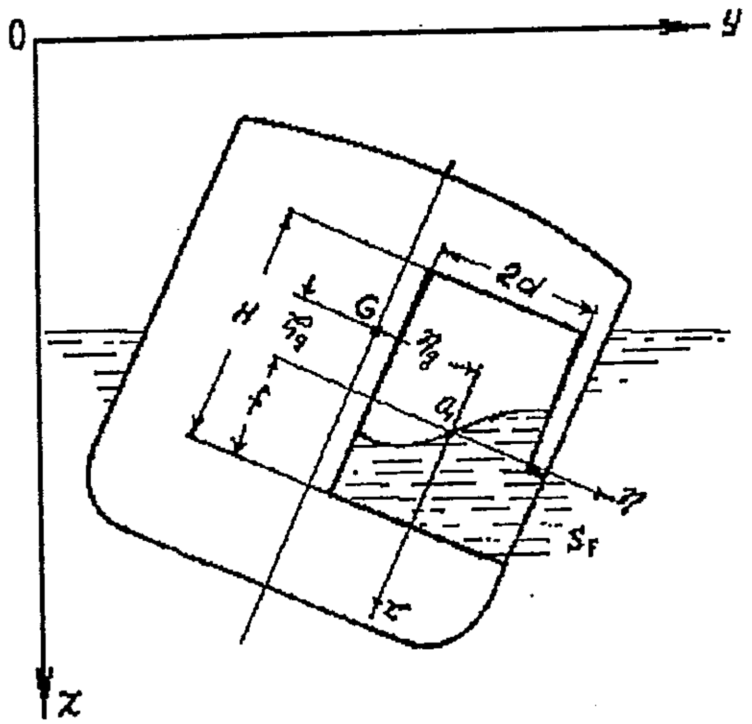
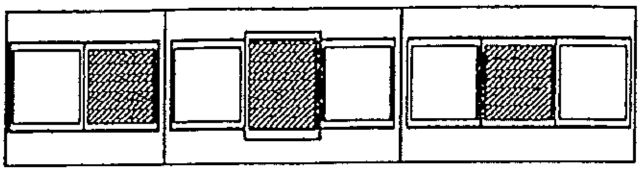
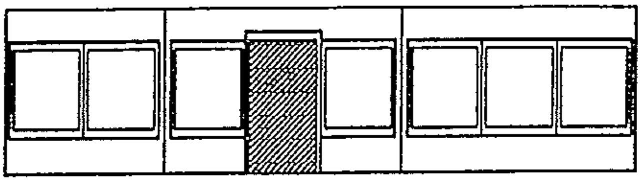


Fig.1. Coordinate Systems



a)



b)

Fig.2. Scheme of model compartment location and flooding: (a) 2-nd category compartments, (b) the 3-rd category compartment

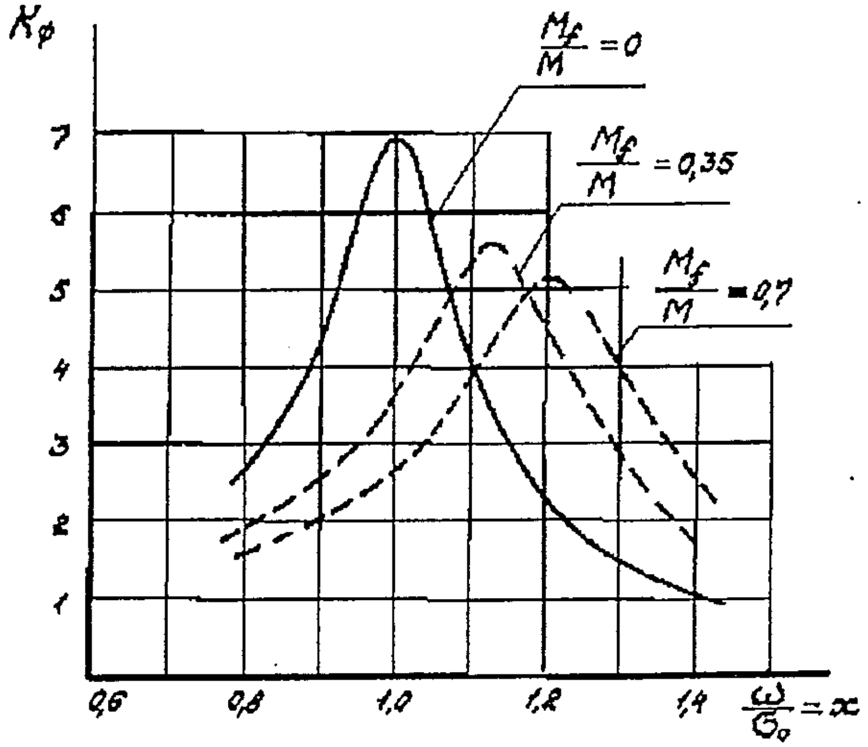
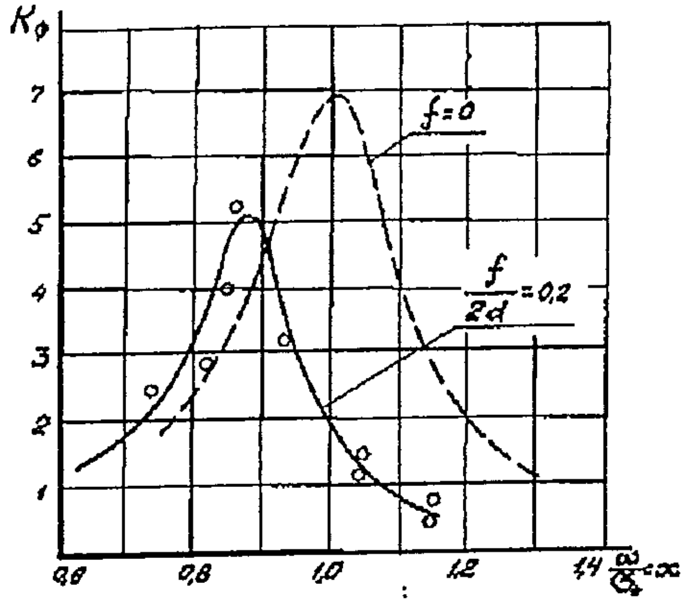


Fig.3. Roll response amplitude operators of the model with flooded compartments of the 1-st category: M_f denotes the flooded water mass, M is the entire model mass



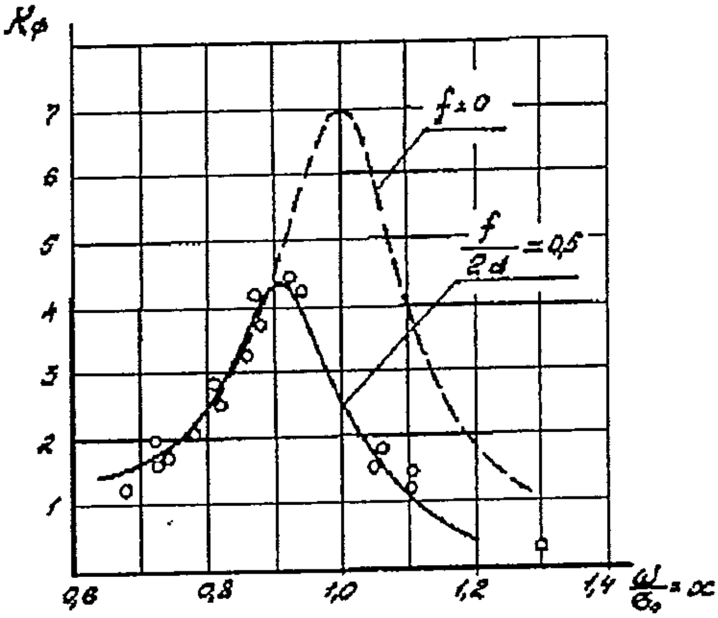


Fig.4. Roll response amplitude operators of the model with the 2-nd category compartments of a different flooding depth

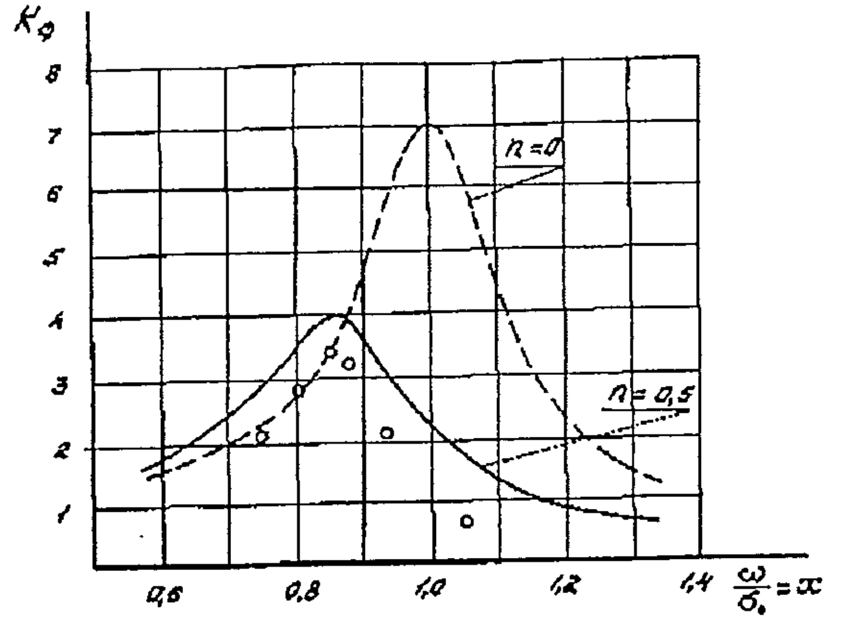


Fig.5. Roll response amplitude operator of the model with a compartment flooded through the side damage

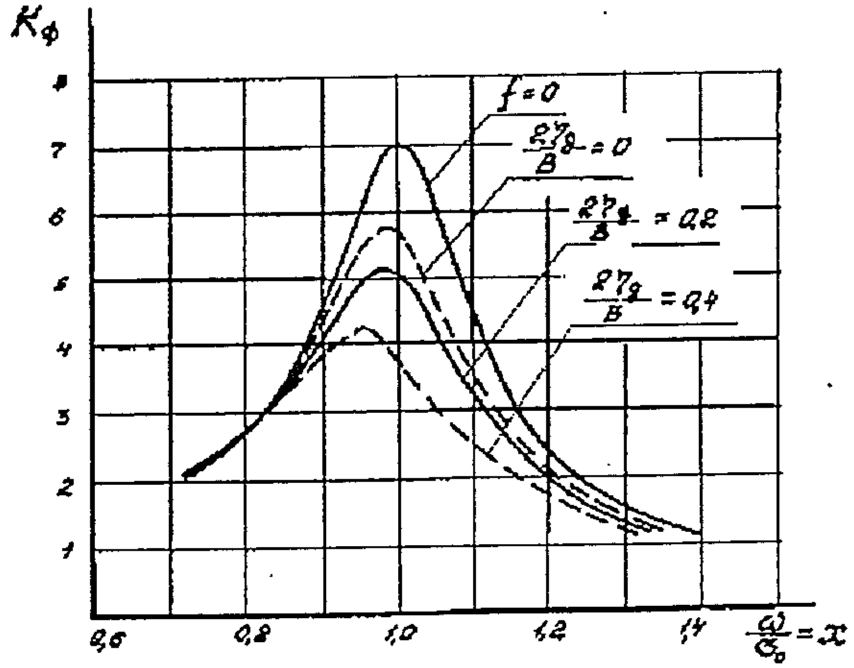


Fig.6. Roll response amplitude operators of the model with a compartment flooded through the bottom damage having the same area as the flooded water free surface

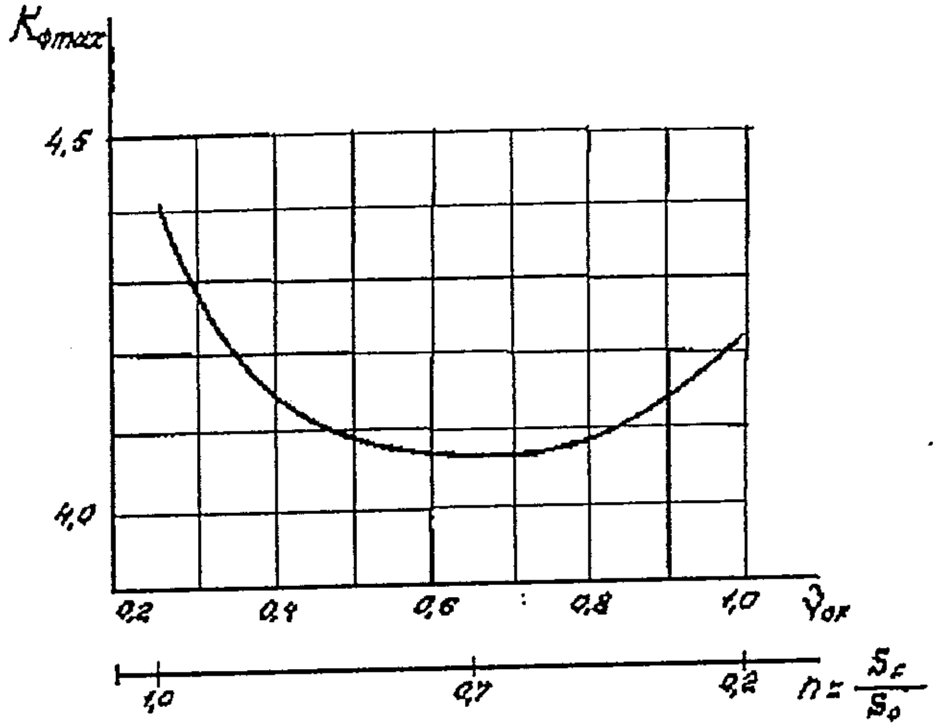


Fig.7. RAO maximum values of model roll motion against the relative damping coefficient of liquid volume oscillations in a compartment flooded through the bottom damage.

NUMERICAL ANALYSIS OF A VESSEL'S DYNAMIC RESPONSES WITH WATER TRAPPED ON DECK

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ABSTRACT

When analyzed statically, the presence of water on deck reduces intact stability. In a dynamic analysis, the sloshing of water on deck affects the vessel's responses to waves and may evanesce the ship to capsize. The focus of this research is to determine the vessel's dynamic response when water is present on the deck. A numerical program developed by combining two existing computer programs was used to calculate the motion response of a vessel with water on deck. Three degree-of-freedom (sway, heave, and roll) vessel motions were calculated. These calculations indicate that when the vessel is in a stable condition, the determination of the "low frequency" response due to water on deck may be the critically important in predicting vessel capsize. When the vessel is in a marginally stable condition, the water on deck may result in a quasistatic heel angle.

1. INTRODUCTION

The effect of water on deck on the stability of a fishing vessel has been studied over the past 40 years. The water trapped on a fishing vessel's deck runs to one side of the vessel, and exerts an additional moment on the vessel. The water on deck may produce a large overturning moment. The effect of water on deck is divided into two parts -statics and dynamics. Statically, the effect of water on deck, which is well known as the free surface effect, is studied extensively. Dynamically, the effect of sloshing on the vessel's response and the stability of the vessel has not been investigated adequately.

In 1982, Caglayan and Storch [1] reviewed the early attempts to study the effect of water on deck. Most of dynamical analyses on the effect of water on deck were based upon empirical or experimental investigations. Only one hydrodynamic approach to study the effect of water on deck was found [2]. Recently, a limited number of analytical and computational studies have been performed on the dynamics of water sloshing and its effect on a vessel's response [3,4,5,6]. In these studies, Dillingham [2] found that the motion of the water sloshing on the deck sometimes tends to reduce the ship's roll motion. Caglayan [3] conducted experiments to verify the phenomena of roll motion reduction in three degree-of-freedom motions. The results of roll motion reduction is contradictive to the expectation of statical analysis of the effect of water on deck. Another important phenomena verified in Caglayan's experiment is called pseudostatic angle of heel. This pseudostatic angle of heel is defined as a stable static equilibrium heel angle for a vessel with water on deck and with negative metacentric height (the metacentric height includes the effect of water on deck).

In 1991, Falzarano[7] proposed a ship's dynamical system with water on deck. In his approach the water on deck was transferred to a fixed weight which could modify the restoring moment curve. If a negative metacentric height is induced by the water on deck, three types of motion response were found. These three types of motion are harmonic, subharmonic, and aperiodic motions. Experiments verification of the importance of these response motions is needed.

The purpose of this study is to investigate the interactions between the sloshing of water on deck and the vessel motions for three degrees of freedom in order to discover the effects of water on deck on the vessel responses. Two contributions should be noted. First, the results from the numerical computations provide information about the effect of water on deck which may help to interpret the mechanism by which the water on deck serves as a motion absorber. This explains the contradiction between the results of roll motion analyses of Caglayan and Dillingham[2,3] and the expectation from the statical analysis of the effect of water on deck.

Second, a more realistically numerical simulation reaffirms the three types of motion responses and demonstrates a route to an aperiodic motion.

Based upon the present work, extension to the study of capsizing due to water on deck and the determination of the dynamic stability boundary can be made in the future. Such that the designer may take the steps necessary to mitigate the effects of water on deck. .

2. THEORETICAL ANALYSIS OF THE WATER ON DECK PROBLEM

In this section, a mathematical formulation of the equations of motion for a vessel with water on deck is presented. Some of the important assumptions required in this formulation are postulated as follows:

1. All ship responses are nonlinear to some extent, but in many cases when nonlinearities are small, a linear theory will yield good ship motion predictions.
2. To calculate the hydrodynamic and exciting forces acting on the ship, potential theory is employed to obtain the velocity potential of the fluid flow around the ship's hull.
3. On the deck, a shallow water wave equation is employed to simulate the sloshing of water on deck.

Under these assumptions, the equations of three degree-of-freedom motions (sway, heave, and roll) are formulated for a geometrical model shown in Figure 1. These equations of motion employ a combination of linear theory for the ship motions and nonlinear shallow water wave theory for the water sloshing on deck. A yz coordinate system as shown in Figure 1 is adopted. Here, z is directed vertically upward, and y is directed horizontally. In the upright position of the ship without water on deck, the center of the yz coordinate system is located on the center of gravity.

2.1 External Force Method

The appearance of water on deck leads to confusion when considering the physical and hydrostatic properties of the vessel. For instance, where is the center of gravity located? Because the water is moving on the deck, the center of gravity for the combined vessel and water on deck is also moving. How are the other properties of the vessel (e.g. metacentric height, mass moment of inertia etc.) computed?

There are two ways to approach the formulation of the equations of motion for a vessel with water on deck. In the first method, the water on deck and the vessel are combined into one system. In the second method, the water on deck is treated as an external force.

When the water on deck and the ship are combined into a single system, the problem of computing the time dependent physical properties is extremely difficult to solve. Thus, this method is not used in the present research to simulate the vessel's motions with water on deck.

To illustrate the external force method, the equation of heave motion is represented. From Newton's second law, the translational equation is in the form

$$\mathbf{F} = m\,\dot{\mathbf{v}} \qquad (1)$$

where **F** is the external force acting on the ship body, $\mathbf{v}$ is the absolute velocity of the center of mass, and m is the mass of the ship. Then, the equation of heave motion expressed in the z body fixed axis is

$$m(\ddot{z}+\dot{\alpha}\,\dot{y}) = BF_3 \qquad (2)$$

where $\ddot{z}$ = instantaneous absolute acceleration of the origin of the yz body fixed system, $\dot{y}$ = instantaneous absolute velocity of the origin of the yz body fixed system, $\dot{\alpha}$ = instantaneous absolute angular velocity of the origin of the yz body fixed system, m = mass of the vessel without water on deck, and BF_3 = total force in the z direction.

On the right hand side of equation, the term, BF_3, is decomposed as follows:

BF_3 = z-directional pressure forces on hull + z-directional gravity force on vessel + z-directional pressure forces on deck and bulwark (3)

The pressure forces may be separated into static and dynamic components and using linear superposition, the pressure forces on the hull may be further separated.

BF_3 = oscillating body induced z-directional forces + z-directional external wave forces + z-directional static hull forces + z-directional gravity force on vessel + z-directional force of water on deck (4)

Here, the forces are

oscillating body induced forces = added mass + damping forces + restoring forces, (5)

external wave forces = incident and refracted wave forces, (6)

static hull forces = buoyancy of the vessel with water on deck, (7)

gravity force on vessel = – weight of the vessel without water on deck (8) z-directional force of water on deck = $\vec{k} \cdot \int P\, \vec{n}\, dS$ (S= deck + bulwark), (9)

where $\vec{k}$ is the unit vector in the direction of the z axis, and $\vec{n}$ is the unit inward normal vector on the bulwark and deck.

When the forces are added together, the equation of heave motion becomes

$m(\ddot{z} + \dot{\alpha}\, \dot{y})$ = oscillating body induced z-directional forces + z-directional external wave forces + z-directional static hull forces + z-directional gravity force on vessel + z-directional force of water on deck (10)

Therefore, using the external force method, the center of gravity is located at the center of gravity of the vessel without water on deck. The hydrostatic and hydrodynamic coefficients as well as exciting wave force are computed at the water line of the vessel with water on deck. A more detailed description for the external force method is shown by Lee [6].

2.2 Equation of Motion with Water on Deck

Using a linear theory, the three coupled differential equations of motion for a vessel with water on deck are written

$$(m+A_{22})\ddot{\eta}_2+B_{22}\dot{\eta}_2+A_{24}\ddot{\eta}_4+B_{24}\dot{\eta}_4=F_{2w}+F_{2wod} \tag{11}$$

$$(m+A_{33})\ddot{\eta}_3+B_{33}\dot{\eta}_3+C_{33}\eta_3=F_{3w}+F_{3wod} \tag{12}$$

$$A_{42}\ddot{\eta}_2+B_{42}\dot{\eta}_2+(A_{44}+I_{44})\ddot{\eta}_4+B_{44}\dot{\eta}_4+B'_{44}\dot{\eta}_4^2+C_{44}\eta_4=M_{4w}+M_{4wod} \tag{13}$$

Here, η_2, η_3, and η_4 are the displacements in sway, heave, and roll motions, respectively, m is the mass of the ship without water on deck, and I_{44} is the moment of inertia about the roll axis. Here the inertia terms are computed with respect to the yz body fixed system shown in Figure 1. The hydrodynamic coefficients (A and B) and exciting wave force and moment (F_w and M_w) are obtained numerically, using a potential theory at the waterline of the vessel with water on deck. The hydrostatic coefficients (C_{44}) are determined by the external force method shown by Lee [6]. The nonlinear force and moment due to the water on deck (F_{wod} and M_{wod}) are determined by solving the equations for shallow water waves on deck. The symbol of B'_{44} represents the nonlinear damping coefficient in roll [3].

2.3 Sloshing Motion on Deck

Since the water on deck normally flows out through freeing ports (openings in the bulwarks), it is difficult to retain a very large amount of water on deck. However, a significant amount of water may remain on the deck, if the freeing ports are small. In order to simplify the mathematical formulation of the problem of water on deck, the water is assumed to be inviscid, incompressible, and irrotational. The bulwark is assumed to be high and without scuppers to maintain a constant amount of water on deck.

Since the depth of water on deck is small compared to the beam, the shallow water wave equations suggested by Stoker [8] are used. In two-dimensions the equations of shallow water waves are

$$v_t + v v_y = -g\zeta_y \tag{14}$$

$$[v(\zeta + h)]_y = -\zeta_t \tag{15}$$

where v is velocity in y direction, h is water depth, ζ is free-surface elevation, and g is acceleration of gravity.

Because the water particle's accelerations are strongly affected by the vessel motions, it is necessary to introduce the vessel's movement into the equations of shallow water waves. So, these equations must be transformed into a moving coordinate system attached to the vessel. A complete derivation is given by Pantazopoulos [4]. Only the results are shown in present paper. The coordinate systems are shown in Figure 2. Here, the $\tilde{Y}\tilde{Z}$ system is the coordinate system fixed in space, the yz system is the coordinate system fixed in the ship, and v and w are the fluid particle velocities with respect to the yz body fixed system. The shallow water wave equation in the moving coordinate frame yz is

$$v_t + v v_y = -a_{(z)}\zeta_y + f_{2y} \tag{16}$$

$$[v(\zeta + h)]_y = -\zeta_t \tag{17}$$

where

$$a_{(z)} = -\ddot{\eta}_2 \sin\eta_4 + \ddot{\eta}_3 \cos\eta_4 + 2w_1 v + \dot{w}_1 \bar{y} + g\cos\eta_4 \tag{18}$$

$$f_{2y} = -\ddot{\eta}_2 \cos\eta_4 - \ddot{\eta}_3 \sin\eta_4 + w_1^2 \bar{y} - g\sin\eta_4 \tag{19}$$

Here, w_1 is the roll angular velocity. The y component distance between the water particle and the origin of ship coordinate system is labeled as $\bar{y}$.

Using the random choice method (RCM) on a series of dam-breaking problems, the motion of the water on deck is computed. The velocities and water levels are obtained at every time step. Then, the force and moment due to water on deck are calculated with respect to yz body fixed system.

3. NUMERICAL SOLUTION OF THE WATER ON DECK PROBLEM

The three degree-of-freedom motions of the vessel with water on deck are solved using a numerical scheme including of five steps. These steps are

1. the creation of a numerical geometry file from the vessel's offsets,

2. a numerical method to find the hydrostatic and hydrodynamic coefficients and the wave forces,

3. a numerical simulation of the nonlinear sloshing of water on deck,

4. the combination of the results obtained from the second and third steps to complete the equations of motion for the vessel with water on deck, and

5. a Runge-Kutta method to solve the equations of motion numerically.

3.1 Geometrical Model

A model was selected and used to study the effect of water on deck experimentally and numerically. This model has a cross-section similar to the midship section of a crabbing boat. It was constructed with a scale ratio of 24 to one when compared with standard sized fishing vessels. The dimensions of the model are shown in Figure 3. The draft measured from model's keel and design water line (DWL) is 6.38 inches. The internal ballast is a solid weight which may be moved to alter the location of the vertical center of gravity. After construction, the location of the vertical center of gravity, and the mass moment of inertia about the roll axis were determined experimentally. Three ballast locations were established. By moving the solid weight vertically inside the model, the center of gravity was located at 3.5 inches above the keel (GM/Beam = 0.199). Here GM is the metacentric height and Beam is the beam the model. The mass moment of inertia of the model around its roll axis was 0.251 slug-feet squared. A detailed description of physical properties of the model in GM/Beam = 0.199 condition are given in the reference [6]. The physical properties of the model in other stability conditions (GM/Beam=0.157, 0.241) were also calculated and included in the reference [6].

3.2 Numerical Programs for Ship Motions without Water on Deck

A computer program (BRK2D) was available for the present paper. The BRK2D program developed by Adee, Richey, and Christensen [9] was intended for assessing the performance of floating breakwater and operates under some restrictions. The first restriction is that the floating breakwater is two-dimensional. Under this restriction the motions of the breakwater are sway, heave, and roll. The breakwater also assumes small incident wave amplitude and small motion response of the breakwater allowing linear theory to be applied. Based upon potential theory, the hydrodynamic coefficients and forcing function are computed. Here, the velocity potential is found by distributing constant strength pulsating source singularities on the surface at the body formed by connecting the offset points with straight line segments. This method for solving the boundary value problem is called the Frank Close Fit method.

Some of the important parameters which must be specified as input for the BRK2D program, include:

1. Parameters to specify the cross-sectional area of the immersed body, beam, location of the origin of coordinate system, fluid density, incident wave amplitude, and acceleration of gravity,

2. Incident deep-water wave frequency expressed as Beam/Wavelength, and

3. Offset data to describe the cross-sectional shape of the breakwater.

From the output of the BRK2D program, the added mass and damping coefficients for sway-sway, heave-heave, roll-roll, and sway-roll are obtained for the specified frequency range. The exciting force and moment are also obtained for three degree-of-freedom motions.

3.3 Numerical Simulation of Sloshing Water on Deck

In 1983 Caglayan solved the shallow water wave equation to simulate the sloshing motion of water on deck in a two-dimensional case. The shallow water wave equations (16) and (17) are solved efficiently using a finite difference approximation. The shallow water waves are treated as a series of dam-breaking problems. This treatment is based on dividing the deck into a grid of cells and treating adjacent cells individually. The solution proceeds by assuming that an imaginary dam on the cell boundary suddenly collapses. Depending on the velocity and the depth of water on the two sides of the dam, the Riemann problems, or dam-breaking problems, are solved by the method of characteristics given by Stoker [8]. Then using the random choice method, the new water levels and velocities are found for the next time step. For the dam-breaking problem and the random choice method (RCM), a step by step example is shown in reference [4].

3.4 Numerical Solution for the Vessel Motions with Water on Deck

There are several numerical methods which could be used to solve the equations of motion for a vessel with water on deck. Because of the nonlinear force and moment due to the water on deck, these numerical methods operate in the time domain. In Dillingham's approach [2], the convolution integral technique was the method used to solve the equations of motion. Caglayan [3] used the Laplace transform technique suggested by Livingston [10] to solve the equations of motion. For the Runge-Kutta integral method, the equations are solved directly, even if the nonlinear terms are included in the differential equations. Because the Runge-Kutta method is simple and effective, the present research employs this powerful tool to solve the equations of motion for the vessel with water on deck.

For the sway, heave, and roll motions, analysis using the Runge-Kutta method begins by defining new variables as follows:

$$\dot{\eta}_2 = \eta_8 \quad (20)$$

$$\dot{\eta}_3 = \eta_9 \quad (21)$$

$$\dot{\eta}_4 = \eta_{10} \quad (22)$$

Then, the equations of motion are rewritten:

$$(m+A_{22})\dot{\eta}_8 + B_{22}\eta_8 + A_{24}\dot{\eta}_{10} + B_{24}\eta_{10} = F_{2w} + F_{2wod} \quad (23)$$

$$(m+A_{33})\dot{\eta}_9 + B_{33}\eta_9 + C_{33}\eta_3 = F_{3w} + F_{3wod} \quad (24)$$

$$A_{42}\dot{\eta}_8 + B_{42}\eta_8 + (A_{44}+I_{44})\dot{\eta}_{10} + B_{44}\eta_{10} + B'_{44}\eta_{10}^2 + C_{44}\eta_4 = M_{4w} + M_{4wod} \quad (25)$$

After the second order differential equations are transformed into first order differential equations, the equations of motion are expressed

$$[A]\,[U] = [F] \quad (26)$$

where

$$[A] = \begin{bmatrix} 1 & 0 & 0 & 0 & 0 & 0 \\ 0 & 1 & 0 & 0 & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 \\ 0 & 0 & 0 & m+A_{22} & 0 & A_{24} \\ 0 & 0 & 0 & 0 & m+A_{33} & 0 \\ 0 & 0 & 0 & A_{42} & 0 & I_{44}+A_{44} \end{bmatrix} \quad (27)$$

$$[U] = \begin{bmatrix} \dot{\eta}_2 \\ \dot{\eta}_3 \\ \dot{\eta}_4 \\ \dot{\eta}_8 \\ \dot{\eta}_9 \\ \dot{\eta}_{10} \end{bmatrix} \quad (28)$$

$$[F] = \begin{bmatrix} \eta_8 \\ \eta_9 \\ \eta_{10} \\ F_{2w} + F_{2wod} - B_{22}\eta_8 - B_{24}\eta_{10} \\ F_{3w} + F_{3wod} - B_{33}\eta_9 - C_{33}\eta_3 \\ M_{4w} + M_{4wod} - B_{42}\eta_8 - B_{44}\eta_{10} + B'_{44}\eta_{10}^2 - C_{44}\eta_4 \end{bmatrix} \quad (29)$$

When both sides are multiplied by the inverse of A,

$$[U] = [A]^{-1}\,[F] \quad (30)$$

The required first order equations for the Runge-Kutta method are obtained. After determining the hydrostatic and hydrodynamic coefficients as well as exciting forces from linear potential theory and obtaining the nonlinear force from the numerical simulation of the sloshing of water on deck, the equations are integrated using the Runge-Kutta method. The motions (i.e. displacement, velocity, and acceleration) of the vessel with water on deck are calculated in the time domain. From the results in the time domain, the frequency response curves are obtained.

Preliminary numerical tests indicated a transient response in the first five seconds of the vessel's responses. This is due to the non-zero exciting force and moment at the initial time. The sloshing motion on deck may react violently due to the non-zero exciting force and moment at the initial time which cause the vessel's motion to be irregular for up to five seconds before reaching a steady-state condition. Therefore, it is suggested that the wave exciting force and moment are set to zero at the initial time and then increased linearly until the time equals five seconds. At five seconds, the amplitude of exciting force and moment reach the values obtained from the output of the BRK2D programs. The transient response is reduced dramatically by this method.

4. RESULTS OF CALCULATIONS

In order to study the dynamical effect of water on deck, three stability conditions (GM/Beam = 0.241, 0.199, and 0.157) were examined.

For GM/Beam = 0.241 the model was in a highly stable condition. The water on deck acts as a motion absorber. The numerical simulations were performed with three water levels on deck ($m_{wod} / m = 0$, 9, and 18 %), where m_{wod} is the mass of water on deck and m is the mass of the vessel without water on deck. Two incident wave slopes (one and two degrees) were used for water level on deck at m_{wod} / m = 18%. The nondimensional frequency range was from $f \times \sqrt{L/g} = 0.02$ to $f \times \sqrt{L/g} = 0.051$, where f is the frequency in Hertz, L is the longitudinal length of the model, and g is the acceleration of gravity.

For GM/Beam = 0.199, the predicted numerical results are verified by Caglayan's experimental results [3]. Tests were conducted with two water levels ($m_{wod} / m = 0$ and 9%), one wave height (one inch), and a nondimensional frequency range from $f \times \sqrt{L/g} = 0.275$ to $f \times \sqrt{L/g} = 0.475$.

For GM/Beam = 0.157, the model was in a marginally stable condition.. One water level on deck ($m_{wod} / m = 9\%$) and one incident wave frequency (f = 0.5 Hertz) were chosen. Four wave amplitudes (0.15, 0.25, 0.6, and 1.5 feet) were adopted to demonstrate a route to an aperiodic motion. Three types of motion responses were shown.

4.1 Comparison of the Experimental Results

In order to test the validity of the analytical approach to predict the vessel's response with water on deck, experimentally measured data are needed for comparison. Unfortunately, only a very limited number of experiments have been performed for the problem of a vessel in waves with water on deck. In the present section, comparisons of numerical predictions and Caglayan's experimental measurements [3] of a model's motion with water on deck are performed for three degree-of-freedom motions. The model's stability condition was chosen to be GM/Beam = 0.199.

The first comparison between predicted and measured motions is performed for the model without water on deck. The calculated model responses were obtained for several regular waves at different frequencies. Then, the frequency response curves in Figure 4 were constructed for sway, heave, and roll motions. The curves from the current numerical program (labeled 2D Program) are compared to Caglayan's predictions (labeled Caglayan) and the experimental measurements. There were two sets of experimental data which were acquired for regular wave amplitudes from 0.4 to 0.6 inch (those are represented by the symbol •) and amplitudes from 0.8 to 1.2 inches (those are represented by the symbol ×). Although some discrepancies between the experiments and the predictions are observed, the results are still closely correlated with peak locations. However, a discrepancy between predicted and measured nondimensional response amplitude is found. The amplitude of nondimensional roll resonant responses from the predictions has a higher value than the one from the experiments (20% – 50% difference for the nondimensional amplitudes of roll resonance) This result may be due to the underestimated roll damping coefficient in the numerical programs or additional damping in the physical experiments due to the motion measuring apparatus.

The second comparison is between the experimental measurements and numerical predictions for the model with water on deck. The frequency response curves for the model with a water level ($m_{wod} / m = 9\%$) on deck are shown in Figure 5. In general, the comparison between the numerical predictions and the experiments is favorable. The individual comparisons for each mode of motion are stated as follows:

- In the predictions of heave and roll responses from the current numerical program (2D Program) and Caglayan's predictions (Caglayan), the results are in close agreement with the experimental measurements (5% – 25% difference for the nondimensional amplitudes near resonance)
- In the predictions of nondimensional sway amplitude from the current numerical program (2D Program) and Caglayan's predictions (Caglayan), the predicted amplitudes are larger than the experimental measurements. (10% – 100% difference for the nondimensional sway amplitudes) A possible reason for the difference was pointed out by Caglayan [3] "the low sensitivity in sway measurements, such that the whole length of travel in sway could be covered, small oscillations of the model are hard to distinguish."

In order to investigate the effect of water on deck, a comparison is between the responses for the model without water on deck (Figure 4) and the responses for the model with water on deck (Figures 5). For the sway and heave responses, there is no significant difference for the model with and without water on deck. However, there is a remarkable change in the roll response of the model with water on deck compared to the response of the model without water on deck. With water on deck, the amplitude of roll motion is small for the entire frequency range. The water on deck also causes the roll natural frequency to shift to a higher frequency.

4.2 Highly Stable Condition

In order to further understand the dynamic effect of water on deck, it is necessary to examine the motion response in the frequency range below the lowest frequency examined by Caglayan. GM/Beam is equal to 0.241. Calculations were performed with three water levels on deck (m_{wod} / m = 0, 9, and 18%). Two incident wave slopes were performed with water level on deck at m_{wod} / m = 18%. The resulting frequency response curves are shown in Figure 6. For all cases there is no significant peak for the sway mode and a resonance for the heave mode. The resonant frequency of heave is around one Hertz ($f \times \sqrt{L/g} = 0.33$) and the nondimensional heave amplitude at resonance is close to 2.0. The most noticeable and important motion is roll. When the model has no water trapped on deck, only one resonance is found and the resonant frequency is near 1.30 Hertz ($f \times \sqrt{L/g} = 0.43$). For the model with water on deck, two resonances, as expected, were observed. For the model with one inch of water depth on deck (m_{wod} / m = 18%), the resonant frequencies are 0.26 Hertz ($f \times \sqrt{L/g} = 0.09$) and 1.54 Hertz ($f \times \sqrt{L/g} = 0.51$) and the null frequency is 0.33 Hertz ($f \times \sqrt{L/g} = 0.11$). For the model with 0.5 inches of water depth on deck ($m_{wod} / m = 9\%$), the resonant frequencies are 0.21 Hertz ($f \times \sqrt{L/g} = 0.07$) and 1.33 Hertz ($f \times \sqrt{L/g} = 0.44$) and the null frequency is 0.25 Hertz ($f \times \sqrt{L/g} = 0.08$). Comparing the nondimensional roll resonant amplitudes of the model with and without 0.5 inch of water depth on deck ($m_{wod} / m = 9\%$), the resonant responses for the model with water on deck in both the low and high frequency regions are smaller than the maximum roll response for the model without water on deck. The small amount of water on deck tends to decrease roll motion. Comparing the nondimensional roll resonant amplitudes of the model with and without one inch of water depth on deck (m_{wod} / m = 18%), the resonant response for the model with water on deck in the low frequency region is larger than the resonant response in the high frequency region and smaller than the maximum roll response for the model without water on deck. This resonant response in the low frequency region, which could not be investigated in the experiment, may indicate another critical situation for a rolling ship with water on deck.

4.3 Marginally Stable Condition

The roll response for the model with GM/Beam = 0.241 and water on deck has no pseudostatic heel angle. In order to study the pseudostatic heel angle, the value of GM/Beam must be reduced. In this section the value of GM/Beam is adjusted to 0.157 which represents a marginally stable condition for the model. Calculations were made for four incident wave amplitudes. These waves have a two second period. Predicted roll responses in the time domain are shown in Figure 7. Here, the three axes are time, incident wave amplitude, and roll response. For an incident wave amplitude of 1.5 feet, the time history of the roll response reveals essentially a harmonic motion. As the wave amplitude is reduced to 0.6 feet, the roll response includes subharmonic motion. With a continuing reduction of wave amplitude to 0.26 feet, an aperiodic motion appears. Finally, as the wave amplitude decreases to 0.15 feet, a harmonic motion appears again, but with the oscillation about a quasistatic heel angle. Here the quasistatic heel angle is defined as a steady heel angle about which the vessel rolls in waves. This quasistatic heel angle may be the first indication that before the vessel is in danger of capsizing.

5. Conclusions

The presence of water on deck has the following effects:

1. For an initially stable vessel with a small amount of water on deck, there is no noticeable quasistatic heel angle. The primary effect of water on the deck is to reduce the roll motion response of vessel.
2. For an initially stable vessel with a large amount of water on deck, the determination of the ''low frequency'' response due to water on deck may be the critically important in predicting vessel capsize.
3. For an initially marginally stable vessel with water on deck, the effect of water on deck results in the presence of quasistatic heel angle and three types (harmonic, subharmonic, and aperiodic motions) of roll responses.

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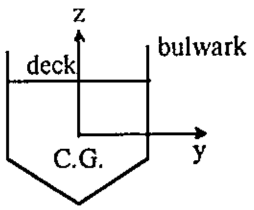

Figure 1 Ship coordinate system

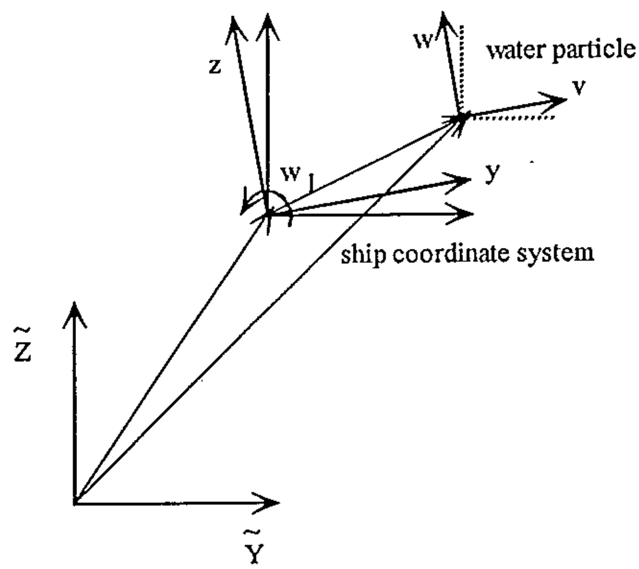

Figure 2 Moving coordinate systems

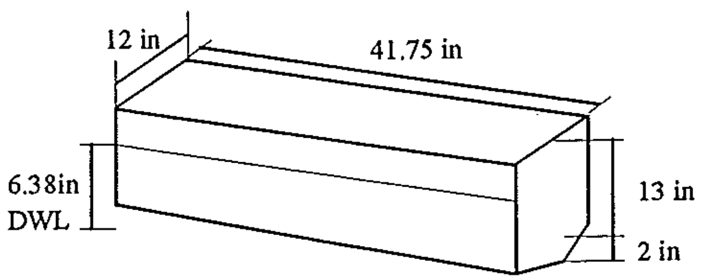

Figure 3 Dimensions of model

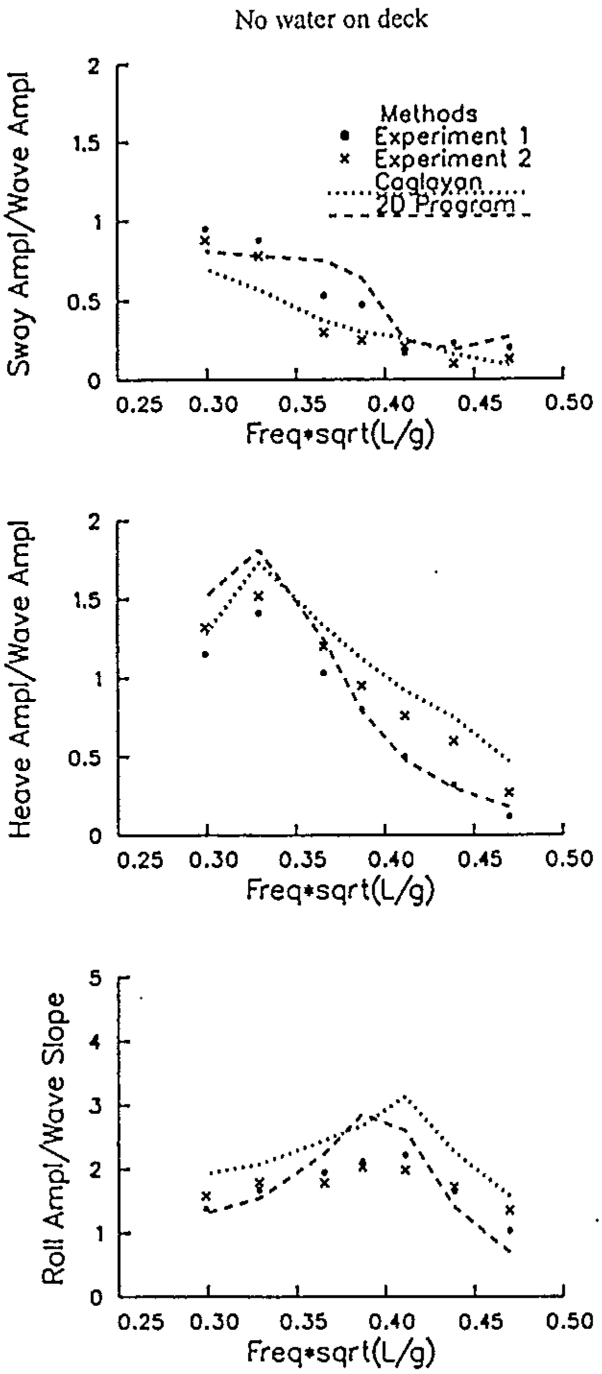

Figure 4 Frequency response curves for the model without water on deck

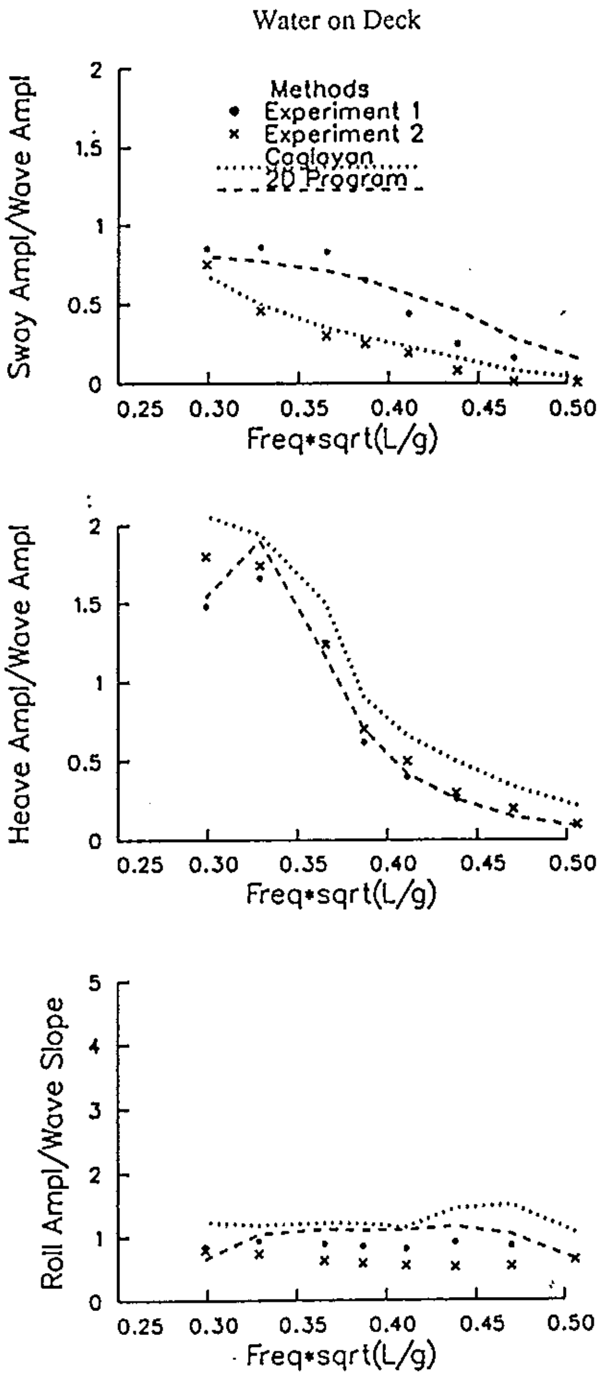

Figure 5 Frequency response curves for the model with water on deck

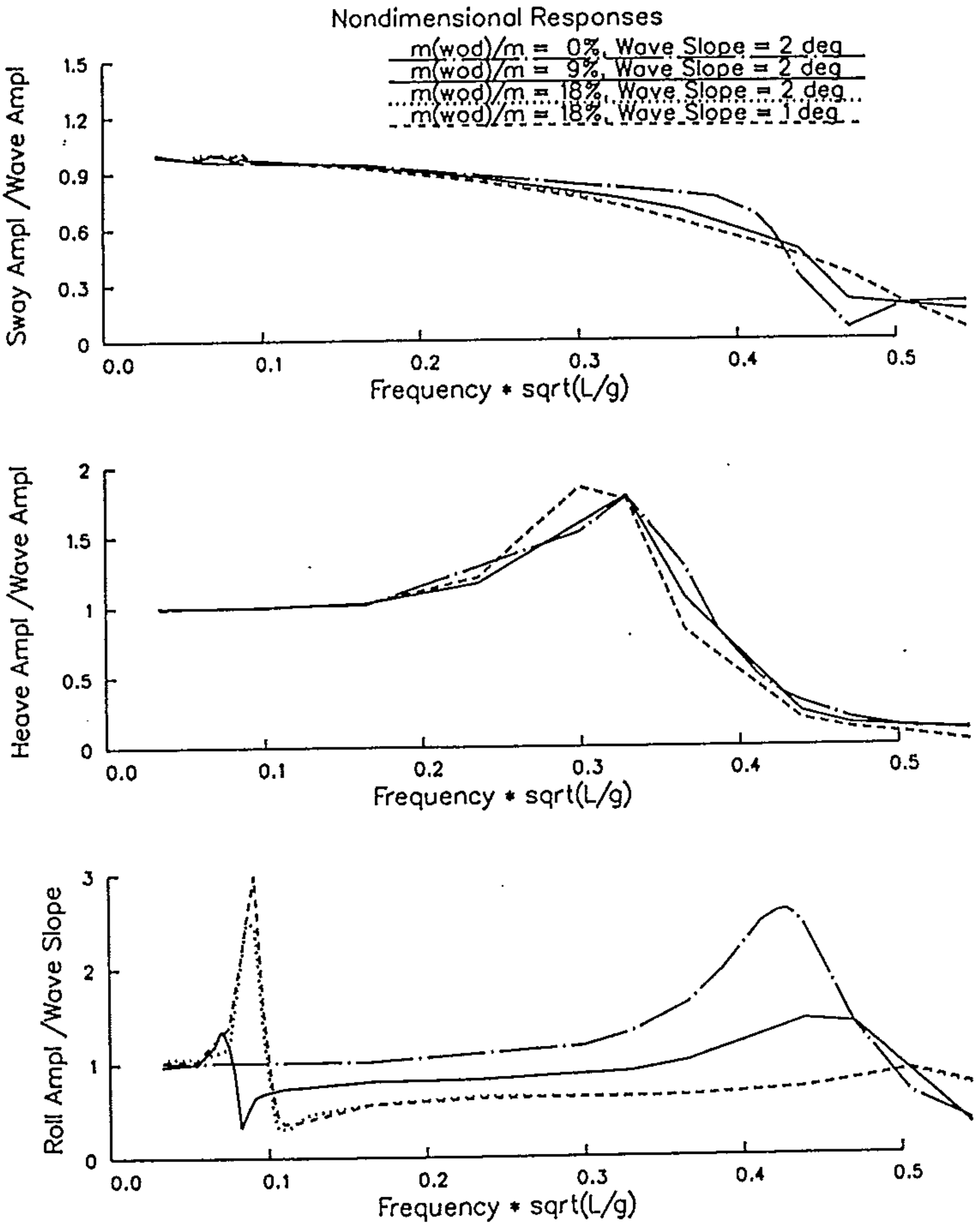

Figure 6 Frequency response curves for the model in a highly stable condition

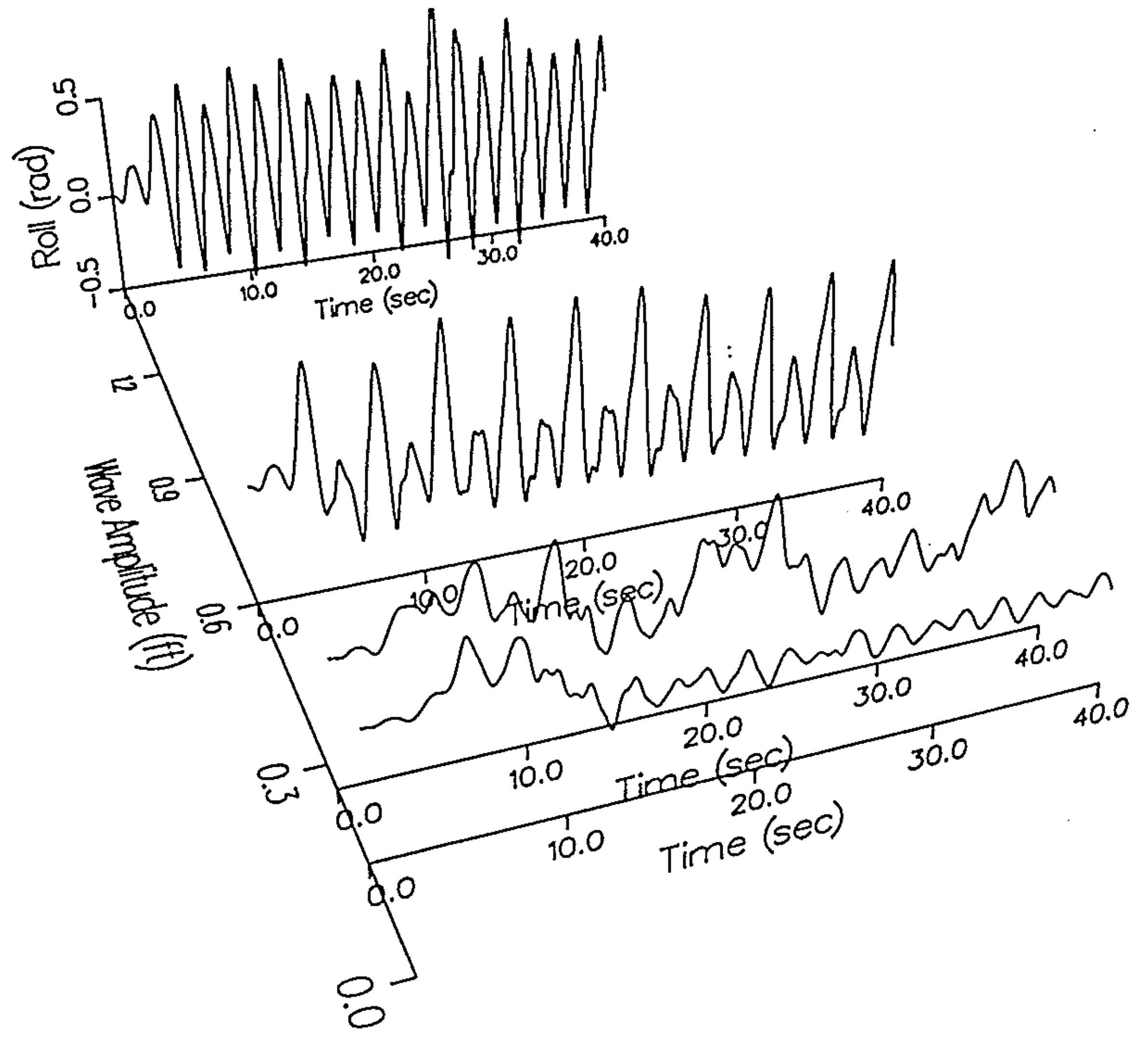

Figure 7 Roll responses for four levels of incident wave amplitudes

Determination of the Capability of Roll Damping Devices using Capsizing Probability

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Abstract

This paper presents a design method for bilge keels of conventional ships by using capsizing probability. First, a calculation method for capsizing probability is well validated with capsizing model experiments of conventional cargo ships. Secondly, it is confirmed by an equivalent linearized method that effects of nonlinear resoring moment are not so significant on capsizing probability. Then, effects of bilge keels on capsizing probability are discussed as well as that of metacentric height. The results demonstrate that we can rationally design bilge keels of cargo ships by the calculation method for capsizing probability with prediction methods for roll damping coefficient and wave exciting moment.

1 INTRODUCTION

To evaluate ship stability in a seaway, the ultimate measure is capsizing probability. Because, wind and waves, that cause capsizing of a ship, have stochastic nature. To do so, Umeda et al. proposed a calculation method for capsizing probabilty of a ship drifting in beam seas, and used it to examine stability criteria for hard-chine craft as well as conventional ships [1] [2].

The capsizing probability will be used as an useful measure not only for establishing stability criteria but also for a ship design. It is an important problem for a designer how size and shape of bilge keels are determined. Since a quantitative measure for stability has not been available so far, bilge keels have been designed with experience. Although a prediction method for hydrodynamic relationship between bilge keels and roll damping coefficient was rigorously established [3], the method has not yet completely used for designing bilge keels.

In this paper, therefore, a design method for bilge keels is presented in the light of capsizing probability obtained by the above method with hydrodynamic predictions of roll damping and wave excitation. This method will be easily extended to a design for other roll damping devices, such as anti-rolling tank, fin stabilizer and so on.

Before using the calculation method of capsizing probability for design purpose, it is desirable to validate the calculation method throughout. Although the authors have already validated the method for capsizing of a hard-chine craft with capsizing model experiments [2], the method should be validated for conventional displacement-type ships. Thus, model experiments were conducted for conventional cargo ships to validate the calculation method. Moreover, the effect of nonlinear restoring moment is examined by both experiments and calculations. Because, the calculation method used in the previous studies by the authors [1][2] only consider nonlinear damping moment but dose not consider nonlinear restoring moment.

2 CALCULATION METHOD OF CAPSIZING PROBABILITY

In this section, the calculation method of capsizing probability is summerized[1][2][4]. The method proposed by Umeda et.al[1] assumed that a ship without advanced speed makes severe roll motion in drifting in beam irregular wind and waves. This condition is the same one as used in Japanese stability standard. The wind velocity is assumed to change with time around an average velocity U_T according to the Davenport spectrum. It is also assumed that the wind generates long crested irregular waves on sea surface with a significant wave height $H_{1/3}$. Since the spectral density of fully developed waves is greater than that of developing waves, the ITTC(1978) spectrum for fully developed waves is adopted to be used.

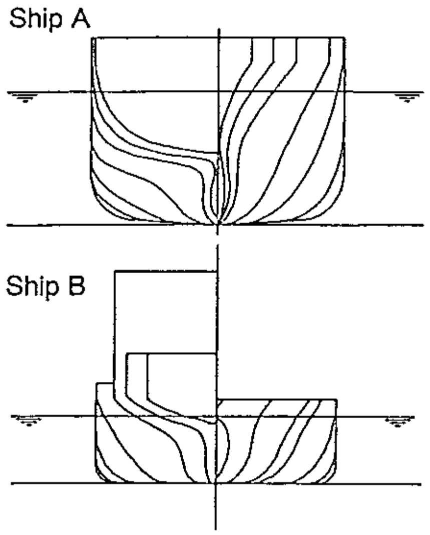
Although, the ship motion in beam sea is usually analyzed by the equations for coupled motions of six-degree of freedom, the one-degree of freedom roll motion equation proposed by Watanabe[5] as follows is used because of the simplicity.

$$(I_{44} + A_{44})\ddot{\phi} + B(\phi, \dot{\phi}) + WGM'\phi = WGM'\gamma\Theta(t) \tag{1}$$

where I_{44} is the moment of inertia of the ship in roll, A_{44} the added moment of inertia in roll, B the roll damping moment, GM' the slope of righting arm at the point of balance by steady wind force, which is obtained by the GZ curve, γ is an effective wave slope coefficient and $\Theta(t)$ the instantaneous wave slope. It has been pointed out that a coupling effect from sway into roll is most important. However, when the wave length is much larger than the ship breadth, Tasai[6] theoretically proved that the roll equation of one-degree of freedom, Equation (1), approximately represents the roll motion including the coupling effect from sway if only Froude-Krylov component is used as the wave exciting moment. This is because that the diffraction moment in roll cancels the radiation moment due to sway into roll. The effect of heave motion on roll motion can be neglected since Shin[7] showed it to be small for heeled ships. In the present calculation the right-hand term of the Equation (1), that is Froude-Krylov component, is obtained by a strip method using Lewis form assumption. The roll damping in the present analysis is represented in nonlinear form as follows:

$$B(\phi, \dot{\phi}) = (I_{44} + A_{44})(2\alpha\dot{\phi} + \beta|\dot{\phi}|\dot{\phi}) \tag{2}$$

This roll damping coefficient can be estimated by the Ikeda's method[3] for conventional ships. In the method, the components of the roll damping, namely frictional, wave making,



	Model	Full scale
L_{pp}(m)	1.12	105.0
B (m)	0.19	18.0
D (m)	0.14	12.9
d (m)	0.10	9.21
W (t)	0.013	11057.
GM (m)	0.009	0.84
T (sec)	1.28	12.4

	Model	Full scale
L_{pp}(m)	1.12	98.0
B (m)	0.22	19.0
D (m)	0.07	6.44
d (m)	0.06	5.18
W (t)	0.011	7247.
GM (m)	0.035	3.03
T (sec)	1.33	12.4

Figure 1: Body plans and principal particulars of the cargo ships used in the experiment

lift, eddy making and bilge keel, are calculated respectively. The frictional, wave making and lift components are linear terms, and the eddy making and bilge keel components are nonlinear terms in Equation(2). Since the roll damping coefficient is dependent on the roll angle due to nonlinear effect, the roll damping and roll angle can be iteratively obtained by an equivalent linearlized method.

3 EXPERIMENTS OF CAPSIZING IN IRREGULAR BEAM WAVES

Model experiments of capsizing were carried out to validate the calculation method of capsizing probability. Although the calculation method assumed an irregular wind, no wind was generated in this experiment. The model, however, had a heel angle on the lee-side by shifting a weight on the deck of it. This condition can be considered to simulate a constant wind, or a movement of cargos in the ship.

3.1 Models

Two cargo ships in Japanese domestic service were selected for this experiment. Body plans and principal particulars of them are shown in Fig.1. The GZ curves of the models in the experimental conditions are shown in Fig.2. The center of gravity of both ships is much higher than the standard condition to occur the capsizing of the model ships easily.

3.2 Experimental method

The experiments were carried out at the towing tank of University of Osaka Prefecture ($70m \times 3m \times 1.6m$ depth). Figure 3 shows the experimental system to prevent large drifting motion of a model in beam waves. The wire rope constrains the model to keep the initial position. The weight under the water acts as restoring force for the drifting motion.

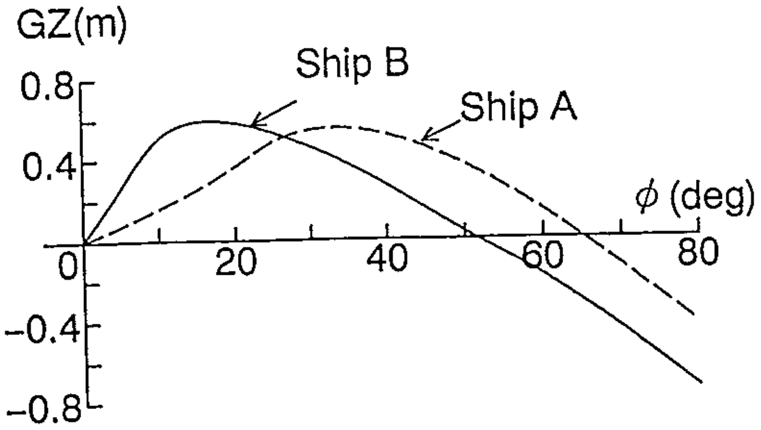


Figure 2: GZ curves of the full scale ships in the experimental condition

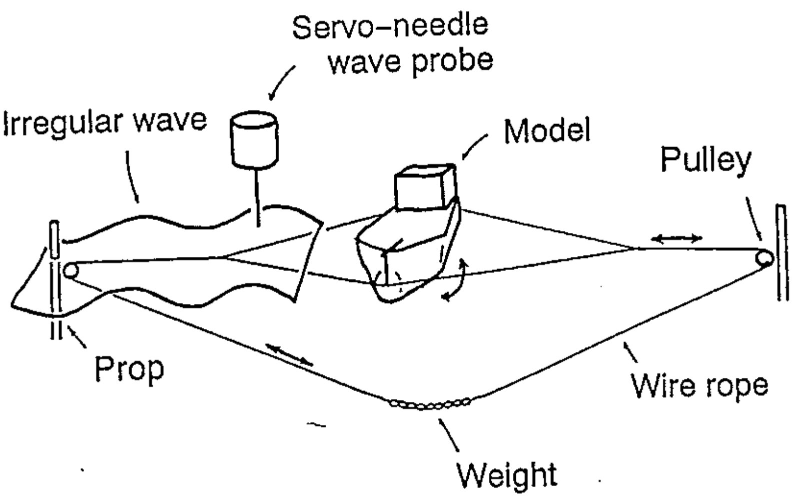


Figure 3: Experimental system

Table 1: Experimental result of the capsizing probability of Ship A

Heel angle (deg)	22.4
Experimental number (times)	44
Capsizing number (times)	11
Capsizing probability (% per 5 minutes)	25.0
Capsizing probability (per roll cycle)	1.63×10^{-3}

This mechanism simulates hydrodynamic reaction force against wave drifting force, whose center is assumed to be at the point of half depth of the ship draft. Therefore the model is captured at the point, and roll motion of the model is in free about the point. The model can drift between two props.

The irregular waves with ITTC(1978) spectrum are used for experiments. The wave height is measured by a servo-needle wave probe. The significant wave height $H_{1/3}$ of irregular waves is about 7.0cm. The mean wave period T_{01} is 0.89 sec. The measurements are repeated for different irregular wave packages with same wave spectrum obtained by changing the phases of the components. Duration of each measurement is 5.0 minutes in these experiments. The capsizing probability is obtained by dividing the capsizing number in times by the total experimental number.

3.3 Results and discussion

The result of Ship A will be shown as follows. Table 1 shows the capsizing probability of Ship A obtained from the experiment. The model is inclined to the lee-side by the heel angle of 22.4 degree. There are 11 times of capsizing in 44 times of experiments. The capsizing probability during 5.0 minutes is 25 %. This value is transformed into the capsizing probability per roll cycle to be 1.63×10^{-3}.

In this experiment, it is found that the wire rope and the weight under the water have a significant effect on the roll damping because the model is comparatively small for the experimental system. Therefore this effect is taken into account for calculating the capsizing probability.

Figure 4 shows the comparison between the capsizing probability obtained by the calculation and by the experiment. The abscissa and the ordinate are a heel angle and a capsizing probability per roll cycle, respectively. The solid line shows the calculated result, and the closed triangle means the experimental result. There is a little difference between the calculated result and experimental one.

The result for Ship B is shown in Fig.5. The calculation method is in fairly good agreement with experimental one in the heel angle of 4.3 degree. This result may suggest that the calculation method can be available for calculating the capsizing probability of a ship.

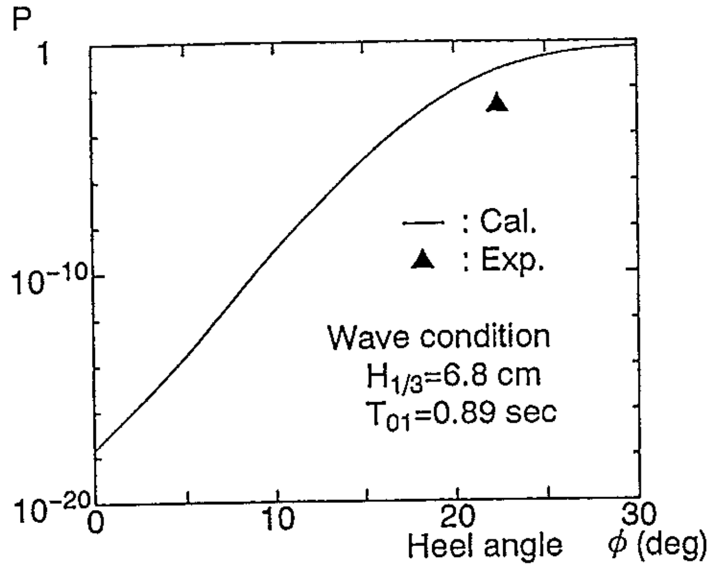


Figure 4: Comparison between experimental and calculated results of the capsizing probability of Ship A

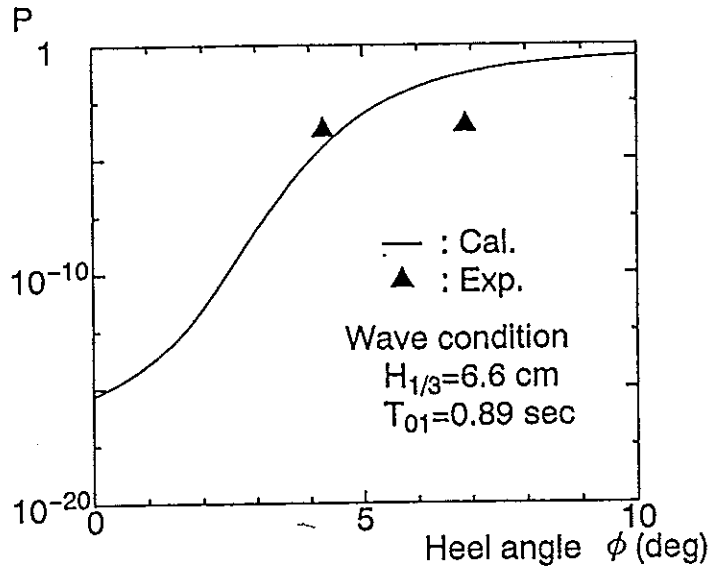


Figure 5: Comparison between experimental and calculated results of the capsizing probability of Ship B

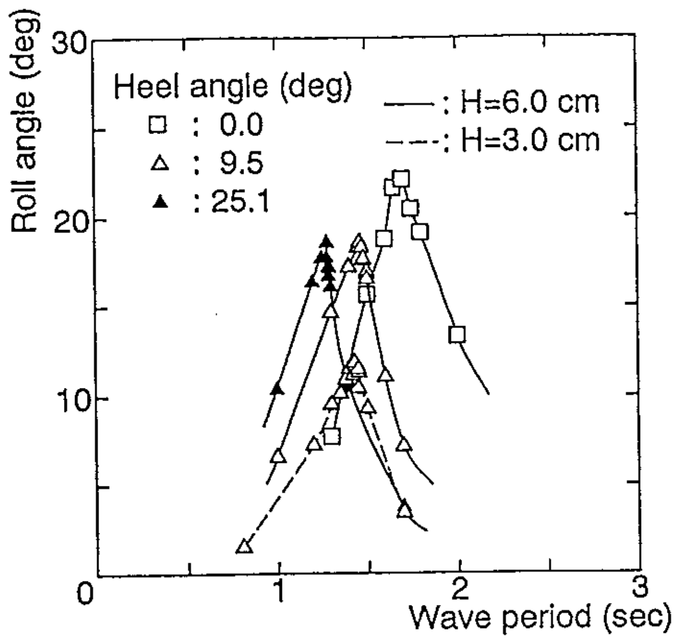


Figure 6: Roll amplitude of Ship A in regular waves

4 NONLINEAR EFFECT OF A RESTORING TERM

In the above calculation, the restoring term is assumed to be linear as shown in the Equation(1). Nonlinearity of the restoring force, however, may be considered to have a significant effect on large roll motion like capsizing motion of a ship. Since the resonance roll amplitude is one of the important factors of the nonlinear effect, the roll amplitudes of the two models were measured in regular beam waves.

4.1 Measurement of roll motion in regular beam waves

The same experimental system mentioned in the previous chapter was used. The roll angle of the model was measured by a position sensor device. The condition of the model, for example, the draft and the position of center of gravity, is same as the capsizing experiments.

The experimental results for Ship A are shown in Fig.6. The abscissa is a wave period, and the ordinate is a roll angle of the ship. The open square shows the case that the model has no heel angle, and the open triangle shows it that the model is inclined to lee side by the heel angel of 9.5 degree; the closed triangle means it by the heel angle of 25.1 degree which is similar condition as the capsizing experiment. The wave height used in the experiment is 6.0cm for the solid lines and 3.0cm for the broken line. These results show the resonant roll period of Ship A decreases with increasing the initial heel angle. This is because of the movement of the center of gravity by shifting a weight on the deck of the model, and because the second moment of area of waterplane increases by taking a large heel angle. Though the resonant period is changed, the skew of the roll response curve is not observed. The difference of the resonant roll period between two lines of the open triangle is slightly appeared. This suggests that the effect of the roll amplitude on the resonant motion is not so significant up to about 20 degree of roll amplitude if the slope of restoring arm curve at the steady heel angle is used as a modified metocentric height for a linear calculation. The similar result is shown for Ship B.

4.2 Effect on the capsizing probability

To investigate the nonlinear effect of a restoring term on the capsizing probability, the probability is calculated with the equivalent linearlized technique [8]. The equation of roll motion can be formulated as follows:

$$\ddot{\phi} + b_e\dot{\phi} + c_e\phi = Q(t) + \varepsilon(t) \tag{3}$$

where

$$\varepsilon(t) = b_e\dot{\phi} + c_e\phi - 2\alpha\dot{\phi} - \beta|\dot{\phi}|\dot{\phi} - WGZ(\phi)/(I_{44} + A_{44}) \tag{4}$$

The $GZ(\phi)$ is represented by an odd-power polynominal up to the third order. In case that a ship has a initial heel angle ϕ_0 by beam wind, the equivalent linear restoring coefficient c_e can be obtained by minimizing the mean square of ε as follows:

$$c_e = \omega_0^2 - 3C_1\sigma_\phi^2 W/(I_{44} + A_{44}) \tag{5}$$

$$C_1 = \{(GZ(\phi_0) - GM\phi_v)/\phi_v^3 - (GZ(\phi_0) - GM\phi_0)/(\phi_v\phi_0^2)\}/(1 - \phi_0/\phi_v) \tag{6}$$

where σ_ϕ is the standard variation of the roll angle, ϕ_v is a vanishing angle obtained from the GZ curve.

The comparison between calculated results using the linear and nonlinear restoring terms for Ship A and Ship B are shown in Fig.7. The solid line shows the result when the linear restoring term whose coefficient is determined by the GZ curve is used, and the broken line show the results for using the nonlinear restoring term. These calculation results demonstrate the nonlinear effect is not so significant for capsizing probability when the stability quality of ships are evaluated with a order of ten to power.

5 APPLICATION OF THE METHOD TO CARGO SHIPS

5.1 Effect of the bilge keels on capsizing probability

The stability qualities of cargo ships for Japanese domestic service are evaluated by using the capsizing probability. The calculation method described in Section 2 is used. Figure 8 shows the profiles, body plans and the principal particulars of the cargo ships used in this calculation. These ships are assumed to be in their full load condition and the GM value of each ship is set up in the standard value of operating condition of it. Figure 9 shows the GZ curves of the ships without a super structure. The weather condition in the calculation is determined using the relations shown in Table 2 for corresponding values of the Beaufort wind scale.

Figure.10 and 11 show the calculated results for Ship C and Ship D with and without bilge keels. In these figures, the abscissa and the ordinate are an average wind velocity and a calculated capsizing probability per roll cycle, respectively. The size of bilge keels

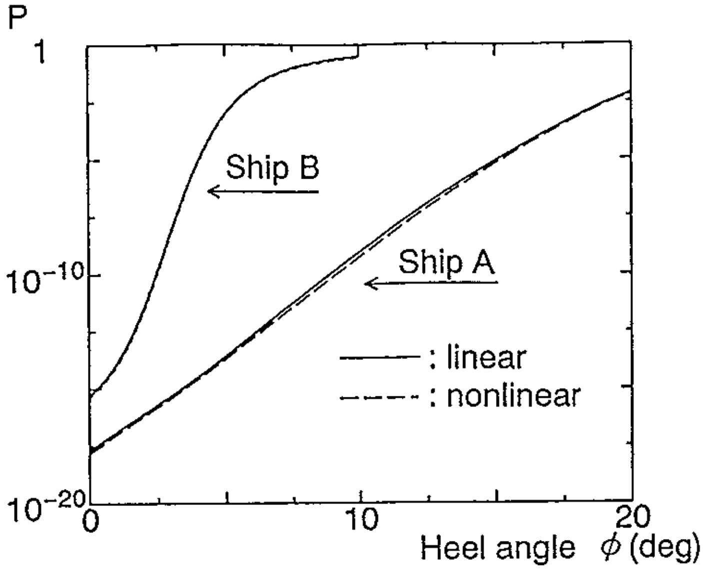
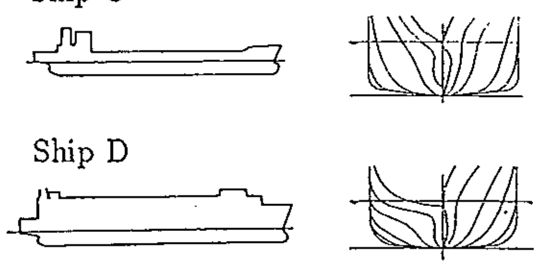


Figure 7: Effect of the nonlinear restoring term for Ship A and Ship B



	Ship C	Ship D
W (t)	2471.4	5835.4
L_{pp}(m)	72.0	105.0
B (m)	12.0	18.0
d (m)	4.16	5.52
GM (m)	0.42	1.10
T (sec)	12.3	12.1
A(m^2)	372.1	1481.4
h (m)	2.78	6.90

T: Natural roll period A: Project area

h: Distance of the center of the wind force from the waterline

Figure 8: Body plans and principal particulars of ships used in the calculation

Table 2: Sea states

Beaufort	U_T (m/s)	$H_{1/3}$ (m)	T_{01} (sec)
5	9.40	2.0	5.5
6	12.35	3.0	6.7
7	15.55	4.0	7.7
8	19.00	5.5	9.1
9	22.65	7.0	10.2
10	26.50	9.0	11.6
11	30.60	11.5	13.1
12	34.85	14.0	14.1

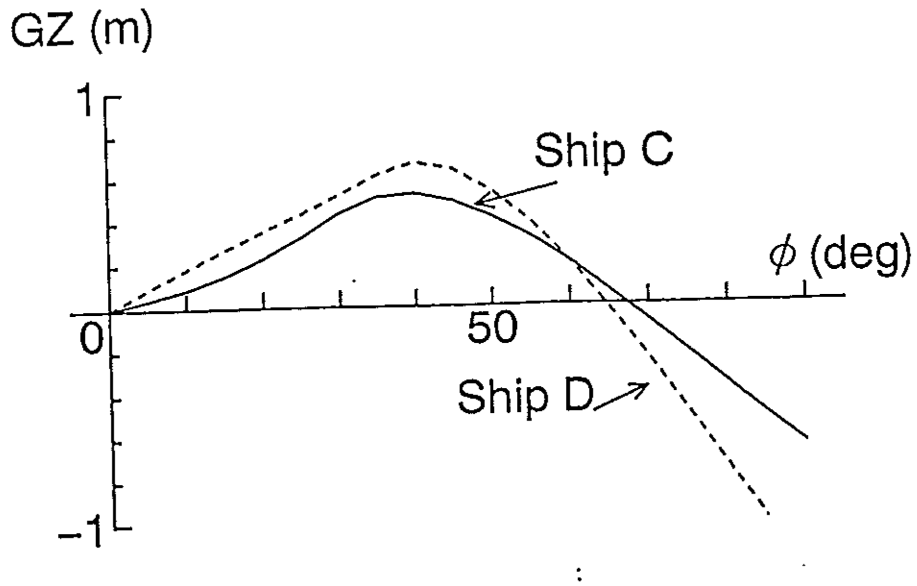


Figure 9: GZ curves of the cargo ships

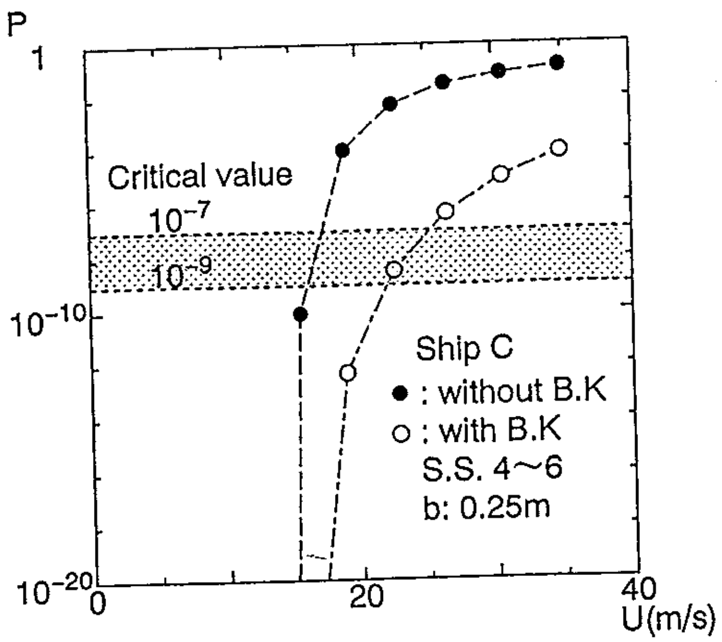


Figure 10: Capsizing probability of Ship C with and without bilge keels vs mean wind speed

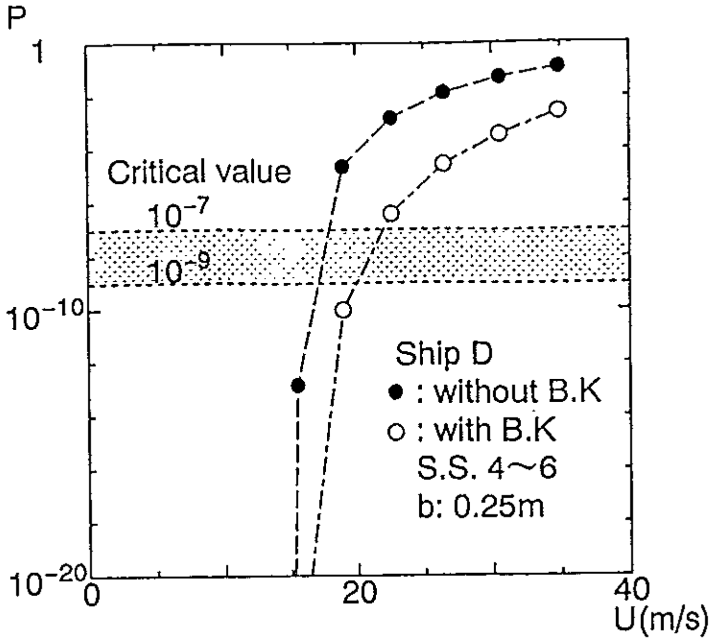


Figure 11: Capsizing probability of Ship D with and without bilge keels vs mean wind speed

in the calculation is assumed to be 0.25m of the breadth and S.S.4∼6 of the longitudinal location. These results show the safety level of ships significantly increases by the effect of the bilge keels.

Umeda temporarily used $10^{-7} \sim 10^{-9}$ as a critical value of capsizing probability on the basis of the calculation results of conventional ships[9]. If we assume the value is proper, the condition of these ships without bilge keels is dangerous in more than 17m/s. Therefore, they need to have an equipment to increase the roll damping for improving the stability quality. Figures 10 and 11 also show that the capsizing probability with the assumed bilge keel in wind velocity of 26.0m/s that is used in Japanese stability standard, larger than the critical value. These results suggest that the condition of those ships are dangerous one from the point of view for capsizing, and that larger bilge keels should be fitted on them.

The effect of the size of bilge keels for Ship C and Ship D is shown in Figs.12, 13. The stability quality of each ship is evaluated to satisfy the critical value. The weather condition is set on the Beaufort No.10 whose average wind velocity is 26.5m/s. The abscissa is a breadth of bilge keels, and the ordinate is a capsizing probability per roll cycle. The closed circle shows the results for the case of S.S.4∼6 longitudinal location of bilge keels and the open circle shows the case of S.S.3∼7. In case of Ship C the capsizing probability in 26.5 m/s wind is less than the critical value of 10^{-7} when the location of bilge keels is S.S.4∼6 and the breadth of them is more than 0.3m, or location of S.S.3∼7 and more than 0.15m of its breadth. This result suggests that the size of bilge keels significantly affects on the stability quality of a ship.

As the efficiency of the bilge keels is considered, the capsizing probability of Ship C with 0.25m of the breadth, S.S. 3∼7 is larger by 15% than that with 0.5m breadth, S.S. 4∼6 in spite of the same bilge keels area. In other words, the shorter the length of bilge keels is, the better the efficiency of it is for Ship C with the same area of bilge keels. The result of

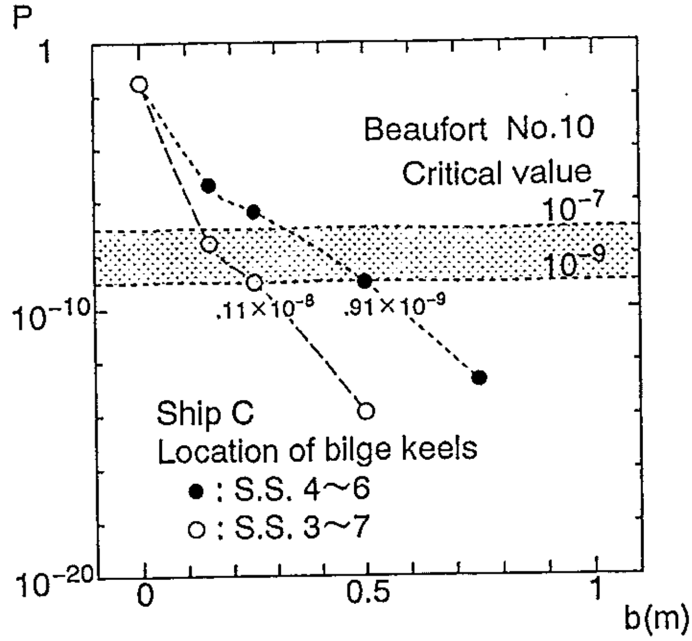


Figure 12: Effect of the size of bilge keels on the capsizing probability of Ship C in 26.5m/s mean wind speed

Ship D in Fig.13 is the opposite. The capsizing probability with 0.5m of the breadth, S.S. 4~6 is larger by 18 % than that with 0.25m breadth, S.S. 3~7. It is important point to decide the size of bilge keels for making the capsizing probability small.

5.2 Effect of GM value

The effect of GM value on the capsizing probability is investigated. The calculated result is shown in Fig.14. The abscissa and the ordinate are GM value and a capsizing probability per roll cycle, respectively. Ship C is evaluated using the capsizing probability in weather condition of Beaufort No.10. The closed circle shows the result in the condition that the ship does not have bilge keels, and the open circle shows the result when the bilge keel is 0.25m of breadth and S.S.4~6 of location. The natural roll period is estimated by using of the assumption that the gyro radius of rolling inertia does not depend on the loading condition of the ship. Figure 14 shows the capsizing probability is less than 10^{-7} when the ship with bilge keels has more than 0.8m of GM value. The capsizing probability has the largest value when GM value equal to about 0.5m, and is small on the part of the small GM value. Because the natural frequency of the ship coincides with the dominant wave frequency of Beaufort No.10 waves at about 0.5m of GM value.

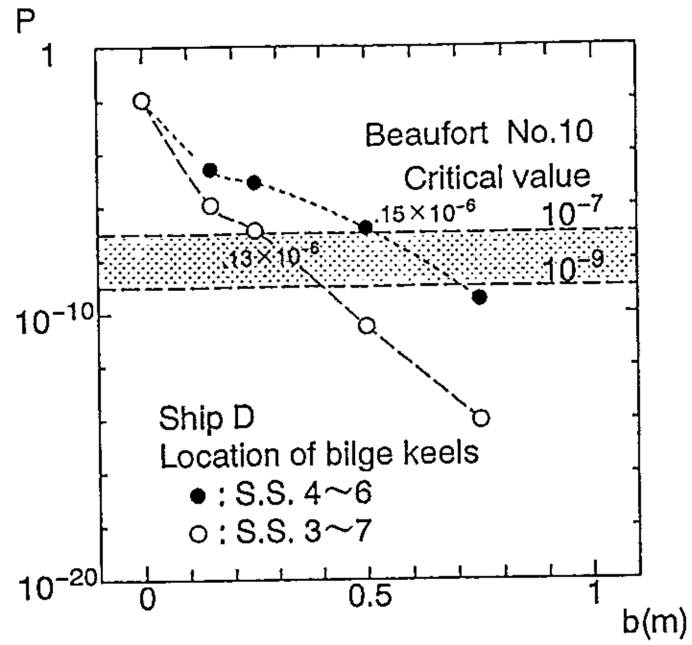


Figure 13: Effect of the size of bilge keels on the capsizing probability of Ship D in 26.5m/s mean wind speed

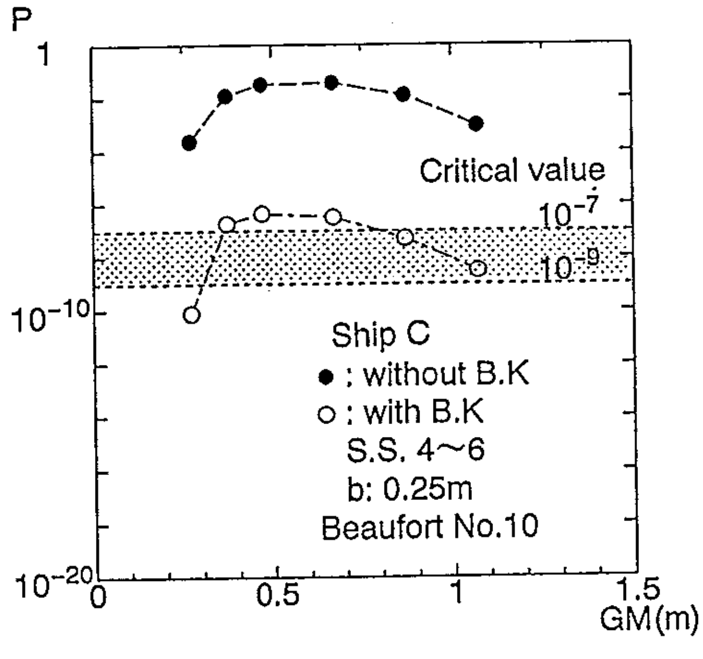


Figure 14: Effect of GM value on the capsizing probability of Ship C in 26.5m/s mean wind speed

6 CONCLUSIONS

On the basis of the present study, the following conclusions can be drawn.

1. Since the comparison between the calculated result and the experimental one shows in fairly good agreement, the present calculation method is valid for evaluating the stability quality of ships.

2. The nonlinear effect of the restoring term on the capsizing probability is not so significant in this calculation and experiments if the equivalent linear GM value is determined by the GZ value at the steady heel position by beam wind. This result suggests that the calculation method using the roll equation with the linear restoring term is effective.

3. Using the capsizing probability, the stability quality of a ship can be evaluated for any weather conditions.

4. The effect of bilge keels and GM value on the capsizing probability can be calculated by the present method. These results suggest that the method is useful to decide the optimum GM value and the optimum size of bilge keels to keep the sufficient stability quality of a ship in design stage.

5. The stability quality with the roll damping divices, anti-rolling tanks, fin stabilizers and so on, can be evaluate by using the capsizing probability if the roll damping of it can be predicted.

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DAMAGE STABILITY ASSESSMENT – STATE OF THE ART

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Abstract

At the present state of knowledge it is not possible to determine, with a reasonable degree of accuracy, criteria for capsizing of ships in waves which are crucial for the assessment of damaged ship safety. However, based on the results of damage stability model experiments, it is possible to derive a simplified relationship which takes into account only certain parameters and disregards others. Such a situation is reflected in the present criteria on damage stability. The background to these criteria is briefly discussed.

The paper emphasizes that relying on model tests only is not sufficient to make much progress and therefore more research is essential. Damage stability is a complex problem involving a large number of variables and the results from only one or two given ship shapes are not sufficient to identify the significant parameters governing the problem. Full benefit from test results will not therefore be achieved until there is a consistent logical theory which will enable the results to be generalised to other ship forms and sizes. Such theory at the moment does not exist. The paper discusses briefly two possibilities of creating such a theory, one based on a simplified model of damaged ship behaviour and the other – on a comparative method.

Finally, the paper discusses practical calculation of the factor 's', accounting for the damage stability of the ship in subdivision calculations. There is a need to improve 's' calculation in several points such as intermediate stages of flooding, heeling moments, the minimum possible vertical extent of damage, permeability for dry cargo spaces and – the most important of all – the mode of calculation which is overlooked by the rules of all certifying authorities and which has a great impact on the GZ-curve. As these calculations should be always carried out for the freely floating ship, longitudinally balanced at each angle of heel, a new angle of heel, different from those in use, has been proposed in the paper consistent with physical considerations.

DAMAGED SHIP CRITERIA FOR CAPSIZING IN WAVES

At the present stage of knowledge it is not possible to determine, with a reasonable degree of accuracy, criteria for capsizing of ships in waves. However, it is possible to derive experimentaly a simplified relationship which takes into account only certain parameters and disregards others.

On the basis of model tests carried out separately in the United Kingdom, as reported by Bird [1,2], and the USA, as reported by Middleton [3] it was noted that for any given sea state (characterised by significant wave height $h_{\frac{1}{3}}$) and freeboard the critical range of *GM* within which capsizing or survival was uncertain was quite narrow. Consequently, it was considered justifiable to treat the relationship as determinate, even though some degree of randomness may be present, especially in natural sea conditions.

From the results of the model tests it was decided to use *GM* and effective freeboard rather than the righting arm to estimate damaged ship safety. This is supported also by numerical calculations which show that the righting arm curve in the damaged condition is approximately proportional to the righting arm at an angle at which the weather deck edge becomes immersed. Observations in the model tests, confirmed also by new tests reported by Graham [4], Pucill [5] and Dand [6], showed that in some cases there could be ambiguity as to the effectiveness of the righting arms depending on the direction of heel (whether away or towards waves) and upon internal structural arrangements. The tests generally showed that the minimum righting arm required for survival relative to waves varies in a complex manner with freeboard and the same relationship for *GM* and freeboard was much less complex.

Supplementary model experiments showed that for any given sea state the *GM* necessary, in association with any given freeboard, is approximately proportional to the ship's breadth. Consequently, the critical significant wave height may be considered as a function of *GM* and an effective angle of the weather deck edge immersion:

$$h_{\frac{1}{3}critical} = f\left(\frac{GM_1 F_e}{B}\right)$$

where

F_e = $F_1 - \frac{1}{2}B \tan \varnothing$, is the effective freeboard after damage including an allowance both for the virtual increase of freeboard due to erections and/or sheer and for a decrease of freeboard due to heel;

F_1 – the effective mean damage freeboard in the upright position defined in [7];

GM_1 – the metacentric height flooded, calculated for the ship in the upright position in the final stage of flooding, using the constant displacement method;

B – the extreme moulded breadth at mid-length of the bulkhead deck.

THE CONDITIONAL PROBABILITY OF COLLISION SURVIVAL

The safety of ships in the damaged condition is nowadays assessed using the probabilistic concept. It is therefore important that damage stability criteria are expressed in terms of probability.

The probability s that a ship with a given value of the parameter GM_1F_e/B will not sink and not capsize during collisions is equal to the probability that the critical significant wave height related to this parameter is not exceeded. Therefore the probability s can be derived from the significant wave height distribution at the moment of collision $F = F(h_{\frac{1}{3}})$, with a combination of the damage stability criterion $h_{\frac{1}{3}} = h_{\frac{1}{3}}(GM_1F_e/B)$, as a composite function of the damage stability parameter GM_1F_e/B:

$$s = F[h_{\frac{1}{3}}(GM_1F_e/B)]$$

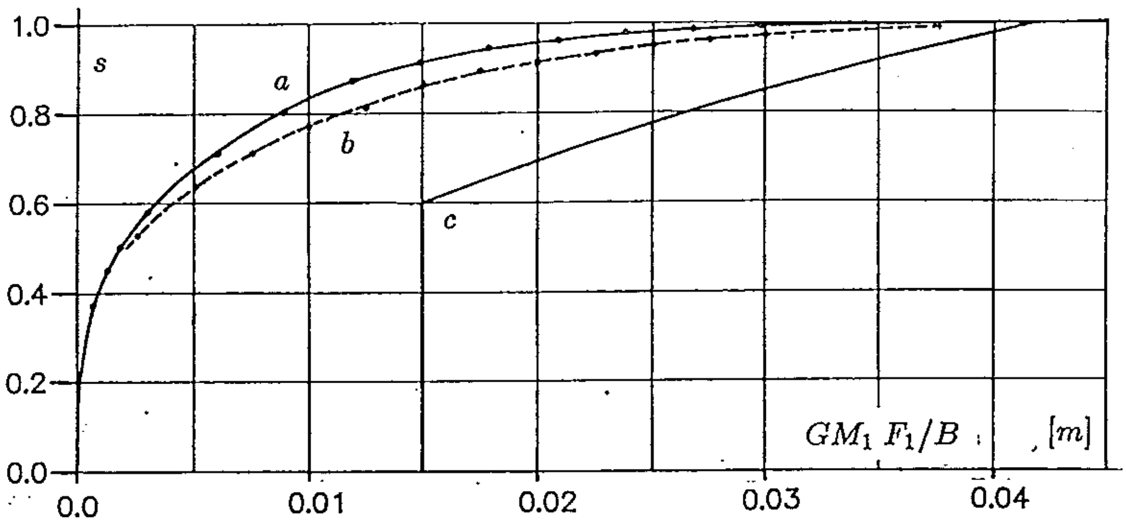

Fig.1 Probability of collision survival: a – based on model tests and sea state distribution, b – based on a comparative method, c – approximation adopted by IMO.

A graph of this probability is shown in Fig. 1, obtained using data given in [2, 7]. A simple but adequate approximation of this probability is

$$s = 1.7\left(\frac{GM_1F_e}{B}\right)^{\frac{1}{6}} = \left(\frac{x}{x_m}\right)^{\frac{1}{6}} \tag{1}$$

but not more than 1, where x stands for the damage stability parameter in metres and $x_m \approx 0.0416$ m is a value of x yielding $s = 1$. IMO, however, approximates this probability with a large under-estimation (see Fig. 1), using the formula

$$s = 4.9\left(\frac{GM_1F_e}{B}\right)^{\frac{1}{2}} = \left(\frac{x}{x_m}\right)^{\frac{1}{2}} \tag{2}$$

but not more than 1. If s is less than 0.6, which happens if $GM_1F_e/B < 0.015$ then IMO requires s to be taken as zero [7]. The philosophy behind this procedure is difficult to accept without reservation but it is still in force for passenger ships.

The calculation of the probability $s_i = s$ would therefore be simple if the damage stability parameter GM_1F_e/B was determinate for each group of compartments. However, this parameter is not determinate since it depends on such random quantities as the loading condition (draught T, trim t and GM), vertical extent of flooding H and the permeability μ at the moment of collision. Therefore, in order to obtain the composite probability s_i for all possible combinations of μ, H, T, and GM (before collision), it is necessary to average s with respect to all these random variables. This follows from the formula for entire probability. Hence:

$$s_i = E(s) = \int_\mu \int_T \int_t \int_{GM} s\, f(\mu, T, t, GM)\, d\mu\, dT\, dt\, dGM$$

where $s = s(\mu, T, t, GM)$ is a function of actual quantities at the moment of collision, averaged previously with respect to possible different vertical extents of flooding, for a ship with horizontal subdivisions.

As can be seen, to find s_i it is necessary to know the joint distribution density $f(\mu, T, t, GM)$ which can only be derived from statistical data, and which in practice is virtually impossible to obtain. Such distribution might also be related to the ship type and possibly to the ship's route, but again the understandable lack of data would prevent these parameters from being considered.

Remembering that the method is aimed at arriving at an assumed rather than the actual probability of survival, the averaging procedure may be largely simplified by accepting draught and possibly vertical extent of flooding as the only random variables and assuming the others to be determinate either as constants or as functions of draught. Hence, the following may be obtained for the s_i factor:

$$s_i = \int_T s(T)\, f(T)\, dT \qquad (3)$$

where $s(T)$ is obtained by averaging s relative to different vertical extents of flooding, and $f(T)$ is the marginal distribution density of draughts.

Some possible modifications of the factor s

The damage stability parameter GM_1F_e/B was partly chosen to facilitate the determination of the attained index of subdivision A by reducing the amount of damage stability calculations. In the latter half of the 1960's when the probabilistic concept was developed this was an important consideration, but that is not the case nowadays, with easy access to computers. For this reason and also due to the fact that the effective damage freeboard F_e is somewhat inadequate and contrived, the regulations for dry cargo ships developed in the latter half of the 1980's abandoned this parameter and used the righting arm curve in the damaged condition directly for calculating the s_i factor. This factor contained in [8] is, however, largely simplified and based on common standards used for damage stability calculations rather than on the results of model experiments.

The damage stability parameter can be easily expressed in terms of some other parameters related to the GZ-curve. For example, it may be observed that GM_1F_e/B equals roughly half the righting arm at an angle at which deck edge becomes immersed. Hence, from formula (1) the following can be obtained:

$$s = \left(\frac{GZ}{GZ_m} \right)^{\frac{1}{6}} \qquad (4)$$

but not more than 1, where GZ_m is a value of GZ yielding $s = 1$, i. e. $GZ_m \approx 0.083$ m.

According to the latest tests [4 – 6] the value of GZ_m should be considerably higher, but not more than 0.20 m – preferably 0.1 m. To facilitate applications for all cases of flooding, including end compartments, it may be assumed that GZ is the maximum value of the righting lever within an *initial range* extending from the angle of equilibrium up to $\frac{2}{3}\,\phi_{max}$, taken as an assumed angle of deck edge immersion, where ϕ_{max} is the angle at which GZ_{max} occurs. However, the initial range should neither extend beyond an angle ϕ_w – the angle at which weathertight openings immerse – nor 22° (= *arc* tan 0.4 – to satisfy a requirement in [7] that F_1 should not be greater than 0.2 B), whichever is less. The angle 22° represents the maximum permissible angle of equilibrium ϕ_m, due to asymmetric flooding and/or other heeling moments, if any.

The above factor *s*, in association with the initial range, caters in a natural way for cases with initial heel. In such a case, *s* reduces gradually with an increase of heel and becomes zero if the initial range vanishes, i.e. when the initial heel exceeds the admissible angle. Therefore, contrary to the situation in the existing formula [8], there is no longer any need for the use of coefficient *c* in the formula for *s*, arbitrarily accounting for the effect of the final equilibrium angle of heel ϕ_e. Furthermore, step changes in *s* values no longer exist as the initial range is now related to the angle ϕ_w. These step changes, inherent in the present formula, are obviously reflected in step changes in the index *A* as the height of the openings varies, as shown by Sen [9], which is not desirable when optimising watertight arrangement for the ship as it encourages paragraph designs. As a rule, step changes in regulations should be identified and eliminated whenever possible.

The maximum angle limit ϕ_m in determining GZ is essential to avoid panic and to allow the safe operation of life-saving equipment (which is of prime importance for passenger ships) and also to reduce the danger of cargo shift. There is no justification for this limit being different from usual values used for the maximum admissible final angle of equilibrium. The angle ϕ_m should therefore be taken as

20° – for passenger ships

25° – for all cargo ships, if the deck edge is immersed; otherwise, ϕ_m can be taken as the angle at which the deck edge immerses but not more than 30°.

Formula (4) accommodates most typical cases of righting arm curves with small ranges of about 20° or less. It is felt, however, that to encourage good design practice some credit ought to be given for greater ranges of righting arm curves. This view is also supported by the model tests referred to earlier [1 – 3]. If this credit is in proportion to the range of stability, the following expression results for the factor *s:*

$$s = \left(\frac{GZ}{0.1} \frac{range}{\phi_m} \right)^{\frac{1}{6}} \qquad (5)$$

but not more than 1, with the ratio of range and ϕ_m to be taken as 1 if range $< \phi_m$, where:

GZ – maximum positive resultant righting lever (in metres) within the initial range;

initial range – range of positive righting levers beyond the angle of equilibrium and terminated at the angle $min(\phi_w, \frac{2}{3}\phi_{max}, \phi_f, \phi_m)$;

range – range of positive righting levers beyond the angle of equilibrium (in degrees) and terminated at the angle $min(\phi_f, \phi_v)$;

$\varnothing_v$ — angle of vanishing stability;
$\varnothing_f$ — angle of flooding (angle of heel at which significant down-flooding occurs);
$\varnothing_m$ — maximum permissible final angle of equilibrium.

As can be seen, the range is considered as large and worth giving credit if it is greater than the angle $\varnothing_m$ — the maximum permissible final angle of equilibrium.

The evidence from the model tests indicates that GZ_{max} measured in some initial range is of basic importance for the safety of the damaged ship, as capsizing is a highly quasi-static phenomenon. Other parameters like 'range' or 'area ' are, therefore, of minor importance. Nevertheless, it seems to be merit in giving some credit for greater ranges or areas under righting arm curves in such a way that excess area or range could compensate for deficient GZ_{max}. The minor importance of other parameters comes from the fact that they are strongly correlated with GZ_{max}. Range is more simple in practical applications than area, therefore the former has been chosen for the use in formula (5).

If, however, the credit mentioned above is taken in proportion to the product of the range of stability and the GZ_{max} value occuring at this range, the following expression results then for the factor *s*:

$$s = \left(\frac{GZ}{0.1} \frac{GZ_{max}}{0.18} \frac{range}{\varnothing_m} \right)^{\frac{1}{6}} \tag{6}$$

but not more than 1, with the two last ratios to be taken as 1, if they are not greater than 1. In cases when the range is not affected by the angle of flooding, it may be assumed that

$$area \cong \frac{2}{\pi} GZ_{max}\, range \frac{\pi}{180} = \frac{GZ_{max}\, range}{90}$$

and the above formula reduces then to

$$s = \left(\frac{GZ}{0.1} \frac{area}{0.002\, \varnothing_m} \right)^{\frac{1}{6}} \tag{7}$$

but not more than 1, with the area ratio to be taken as 1 if it is not greater than 1. The last formula could be treated as another alternative for the factor *s*.

It is worth repeating here that the survival capability of the ship is not related to the GZ_{max} at the whole range but at some intial range and this fact is reflected by the above formulae in which GZ is taken from the initial range. Due to this reason GZ_{max} produces a complex relationship when it is taken for a criterion, clearly revealed by the model experiments.

MODEL EXPERIMENTS AND SOME OF THEIR IMPLICATIONS

Following the tragic capsize of the "Herald of Free Enterprise" in March 1987, the UK Department of Transport initiated an extensive Ro-Ro Passenger Ferry Safety Research Programme, comprising a number of studies into damage survivability of these ships, including model tests in waves [4 — 6]. The main objective of the above experiments 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. Regrettably, it can be said that this primary objective has only been partly achieved. The results obtained are fully valid only for the particular ships investigated and whether they can be extended to other ships is a matter of speculation, although some attempts to generalise them have been presented.

Need for more research

Damage stability is a complex problem involving a large number of variables and with results only from one or two given ship shapes it is difficult to identify the significant parameters governing the problem. Full benefit from test results will not therefore be achieved until there is a consistent logical theory which will enable the results to be generalised to other ship forms and sizes. Such theory, at the moment, does not exist. Observation of the behaviour of floodable models during tests provides an insight into main factors which need to be taken into account and the model results provide some basis for judging the validity of any theory proposed.

The only firm conclusion which may be drawn from the recent tests is that the residual stability standards for ro-ro passenger ships are largely inadequate and should be increased. In addition, the model tests show that the minimum damage stability required for survival at a given sea state varies in a complex manner with freeboard, ship form and structural arrangements. In this situation, until an appropriate theory is developed, it is possible to get only a simplified relationship which takes into account certain parameters and disregards others.

At present there are two basic methods of evaluating damage stability of ships – the probabilistic evaluation method for cargo ships which have recently come into effect [8] and a whole collection of others [10 – 15], including the so-called SOLAS '90 [16] which are deterministic in nature. An alternative probability based standard exists for passenger ships [7] but it is seldom used because of the available deterministic option. This is a frustrating situation because the assessment procedure for damage stability should not be a function of ship type. In the interest of rationality and in anticipation of the future harmonization of all damage stability regulations, it would be clearly sensible that the basic procedure for all ship types should be probability based.

A possible way ahead, in this situation, could be to use the existing SOLAS '90 criteria as a basis, eliminate any parts not justified from the evidence of the damaged model tests, convert the criteria into a probabilistic format, and ensure that full credit is given for buoyancy above the bulkhead deck. As a matter of fact, this is what formula (5) largely does.

It is possible to argue that the SOLAS '90 standard has been adopted in something of a rush and perhaps to some extent under public pressure, being only briefly discussed at the Maritime Safety Committee, without detailed consideration by the more technically orientated SLF Sub-Committee. This has meant that some basic deficiencies of the method could not be attended to. For example, the SOLAS '90 standard has a clearly recognized deficiency: the minimum stability required depends on whether a bulkhead is in way of damage or not, which is clearly meaningless. Nevertheless, it can be said that these requirements as a whole are broadly in line with the results of the latest tests, with the proviso that a fairly low sea state is assumed at time of collision.

Theoretical model of damaged ship behaviour

It may be thought that a satisfactory model for depicting the behaviour of a damaged ship in waves would be far too complicated for use in standards. However, the recent model tests, referred to above, once again confirmed that roll motion of a damaged ship nearly entirely vanishes as flood water appears to be a very effective roll damping medium. Similar observations were made during experiments with a flooded model of a ro-ro cargo ship, carried out in the early 1970's in Poland [17 – 20]. The same observation was also made by Froude in his classic experiments in 1880 with the "Inflexible" [21].

Thus, it seems entirely feasible to configure a simplified mathematical model simulating the motion of the damaged ship in random seas, taking into account the most important degrees of freedom like heave and possibly pitch, in case of forward or aft compartments flooded, and ignoring the less important degrees. Such a model was recently proposed in [31, 32] and the analysis took the form of a generalised green seas approach. The tests shed a lot of light regarding this matter. It was found, generally, that capsizing behaviour is to a large extent quasistatic, and may be thought of as due to heeling moment and stability reduction as a result of water on deck due to wave action and/or heeling moment due to the impact of waves against the windward side. The latter could be entirely ignored as the tests showed that a damaged model invariably capsized towards damage, irrespective of the fact whether it was away or towards the waves. In this simple way the critical *GZ*-curve for preventing a damaged ship from capsizing in a given sea state could be arrived at.

Capsizing is simply due to lack of statical stability caused by water inside the ship whose level is higher from that outside the ship. The increase of water level is due to action of waves and could be directly related to the amplitude of the relative motion of water at the ship side. It is clear that such a model would produce a deterministic relationship between critical stability parameters and sea states, depending on ship form and size, with no uncertain zone. The probability of ship survival could then be derived directly on the basis of distribution of the relevant sea states at the moment of collision.

Critical *GZ*-curve related to sea states based on a comparative method

As yet, there are no damage stability criteria that are supported by theoretical investigations. It would be instructive, therefore, to formulate this problem on a mathematical basis with a view to arriving at a quantitative relationship linking design particulars with survival capability, as proposed above. There is yet another possibility, promising even quicker but nevertheless reliable results.

Broadly speaking, it is possible to develop a comparative method based on freeboard requirements contained in the rules applied for different categories of ships (e.g. river, lake, coastal and sea-going), and extrapolate them for the damage case. Assigning each category of ship a sea state which ships belonging to that category are supposed to survive with certainty provided its intact freeboard and stability are not less than those required by the rules, it is possible to arrive at a relationship relating the sea state to the critical *GZ*-curve for the damaged vessel of a certain size. This comparative method could provide a damage stability criterion relatively quickly. Values of the factor *s*, based on such an approach are shown in Fig. 1 [22]. A good agreement with the model results can be observed in this Figure, although some assumptions adopted there have not been later supported by the results of model tests. However, the approach is well worth investigating further.

PRACTICAL CALCULATION OF THE FACTOR *s*

As mentioned earlier, the *s* factor, for regulatory purposes, is calculated under many simplifications. Some practical aspects of computation are discussed below. The cargo space permeability, trim and the position of the centre of gravity at any draught are random variables but are treated as constant or determinate functions of draught.

Calculation of factor *s*

1. In general, for any vertical extent of flooding at any initial draught *s* should be calculated for the final stage of flooding using formula (5).
2. However, $s = 0$ if an angle during intermediate flooding or prior to equalisation exceeds the maximum admissible angle of heel ϕ_m or the angle at which downflooding may take place.

Thus far stability during intermediate stages of flooding has not attracted the attention it deserves. Work done to date by the author supports the intuitive notion that the intermediate conditions are not usually a problem if the final condition is acceptable, provided the angle of heel is not so large as to cause cargo shift and the water can freely spread over the entire compartment. The deck edge then remains above the water all the time during transient flooding [22].

Thus if there are efficient cross- or down-flooding arrangements it is entirely sufficient as far as damage stability is concerned to check only the maximum angle of equilibrium during flooding, and focus attention on the safety of the ship in the final stage of flooding. Hence, the above theoretical developments, being novel to the naval architects, can be expected to have a significant impact on the simplification of damage stability assessments.

3. In making stability calculations the ship shall be at the most unfavourable intact service trim anticipated at each initial draught condition, having regard to stability in the final stage of flooding. For cargo ships in loaded conditions, level trim may be used.
4. A residual righting lever for passenger ships is to be obtained as *resultant* lever taking into account the greatest of the following heeling moments:
   - due to crowding of all passengers towards one side;
   - due to launching of all fully loaded davit – launched survival crafts on one side;
   - due to wind pressure;

   whereas for cargo ships – assuming solely the heeling moment due to wind pressure which may only be meaningful for ships with excessively large windage area.

The KG value should be taken as KG_{max} for each draught which complies with the relevant intact stability requirements, or if a lower KG value is specified in stability information for the master, that value may be used.

Owing to physical reasons, stability calculations should be always carried out for the *freely* floating ship, longitudinally balanced at each angle of heel, using the constant displacement method. There are usually marked differences between the GZ-curves calculated for the free trim condition and for fixed trim, discussed in detail by Vassalos [23] and Wimalsiri [24], particularly for ships in the damaged condition if the deck edge becomes immersed, that is for ships with longitudinal asymmetry. The importance of the mode of calculation is clearly shown in Fig. 2 which speaks for itself. Despite this fact, internationally accepted rules give no attention regarding this matter which thus may lead to largely distorted results of calculations from different computer program packages.

Regarding stability during intermediate stages of flooding, it should be analysed using the added mass method. However, in the case of horizontal subdivision without efficient arrangements to put the flooding water down, it should be assumed that after the immersion of the edge of the watertight deck, the level of water above such a deck coincides with the level of water outside. This covers the case of a small hole below and a very large one above the horizontal subdivision which is fairly likely nowadays because of bulbous bows associated with large flare.

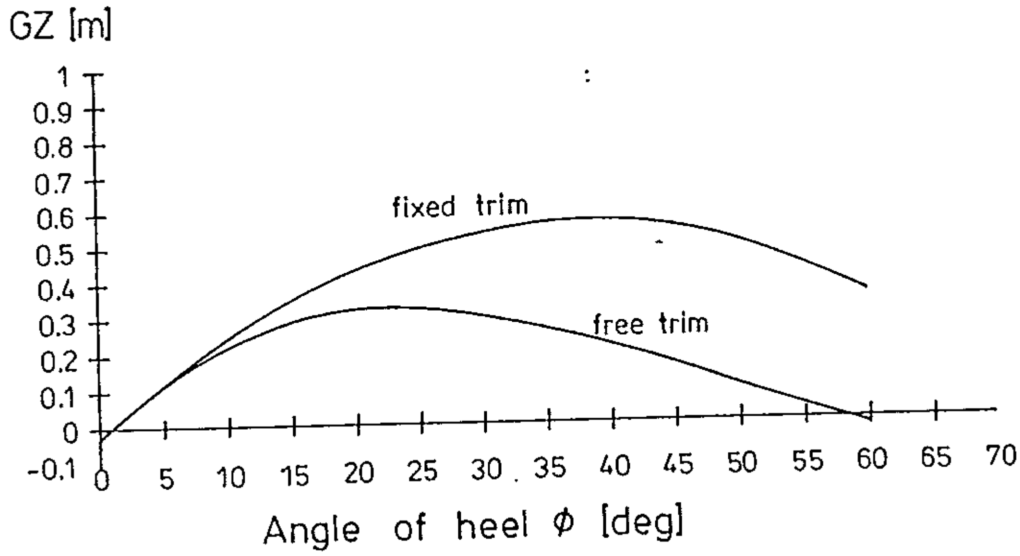

Fig. 2 The effect of mode of calculation on the *GZ*-curve

There is yet one more problem, usually overlooked in stabibility calculations. Namely, for a freely floating ship, it is essential to define clearly what is understood by the angle of inclination. Two different angles of heel are commonly used, but neither of them is correct. These matters are discussed in detail in [25 – 27]. The angle of inclination of the *y*-axis (normal to the plane of symmetry), relative to the water level should be taken as the angle of heel in such a case, the direction of the righting moment is then constant in space and the centre of buoyancy moves along a *strictly flat* curve lying on a plane perpendicular to the axis of rotation [27]. Aparat from the inconsistencies, wide use of the proper angle of heel and mode of floating is of importance for the uniform presentation and interpretation of stability calculations. The lack of guidelines in this respect is one reason why there are so large discrepancies between the results from different sources.

5. If any equalizing arrangements are necessary to increase stability in the final stage of flooding, these arrangements shall, where practicable, be self-acting. However, if

controls are necessary, they should be operable from above the highest relevant bulkhead deck. All such arrangements shall be acceptable to the approving authority.

6. The time for equalization shall not exceed 10 minutes, otherwise the stability of the ship at the stage after 10 minutes of equalization shall be taken for calculating the factor s. In making calculations for heel prior to equalization and for equalization time, the flooding shall be assumed to be completed prior to commencement of equalization. These calculations are to be carried out in accordance with IMO resolution A.266(VIII) [7].

Calculation of factor $s(T)$

Wherever a horizontal subdivision is fitted above the waterline in question, the factor $s(T) \equiv s_T$ is obtained by averaging factor s with respect to different vertical extents of flooding. Otherwise, $s(T)$ is taken as s.

1. The s_T-value for the lower compartment or group of compartments shall be obtained by multiplying the s-value as determined in the previous sub-chapter by the reduction factor v, which represents the probability that the spaces above the horizontal subdivision will not be flooded.

2. In cases of simultaneous flooding of the spaces below and above the horizontal subdivision, the resulting s_T-value for such a compartment or group of compartments shall be obtained by an increase of the value as determined for the spaces below the horizontal subdivision by the s-value for simultaneous flooding according to the previous sub-chapter, multiplied by an increase of the reduction factor v due to the increase of the height of damage H above the baseline.

   For instance, if there are three watertight decks above the waterline with the heights H_1, H_2 and H_3, the following is obtained:

$$s(T) = s_1 v_1 + s_2(v_2 - v_1) + s_3(1 - v_2) \tag{8}$$

where

s_1, s_2, s_3 — s calculated assuming vertical extent of flooding up to the height H_1, H_2 and H_3 respectively;

v_1, v_2, v_3 — v calculated for heights H_1, H_2 and H_3 respectively, with $v_3 = 1$.

3. The probability factor 'v' shall be calculated according to:

   $v = 1$ for the uppermost horizontal subdivision, and for other horizontal subdivisions when they are above a height of H_{max}; otherwise

   $v = \dfrac{H - H_{min}}{H_{max} - H_{min}}$ for the assumed flooding up to the horizontal subdivision above a height of H_{min}, (9)

where

'H' is the height of the horizontal subdivision above the baseline (in metres) which is assumed to limit the vertical extent of damage,

'H_{max}' is the maximum possible vertical extent of damage above the baseline (in metres), given by:

$$H_{max} = T + 0.056L(1 - \frac{L}{500}) \quad \text{if } L \leqslant 250 \text{ m;}$$
$$H_{max} = \text{T}+7 \text{ otherwise} \tag{10}$$

'H_{min}' is the minimum possible vertical extent of damage above the baseline (in metres):

$$H_{min} = H_{max}\,(\tfrac{1}{6}L) \text{ but not less than } T + 1.5. \tag{11}$$

As can be seen, the factor 'v' varies from zero for horizontal subdivision at the level 'H_{min}', linearly upwards to the value of one at the level of 'H_{max}' conforming to the minimum bow height according to the 1966 Load Line Convention. As a large majority of ships have bow heights that are greater in value, H_{max} is currently underestimated and should be upgraded in future. A height of 'H_{min}' is such as to conform to the minimum bow height of the smallest ships which are supposed to collide with the ship in question. It may be assumed that such ships are 6 times smaller than the ship in question. The use of H_{min} is also advisable because of waves which may occur at the moment of collision. Within the present regulations, however, H_{min} is taken as the draught T under consideration [8], giving thus far too much credit for the horizontal subdivision located close to the waterline.

Where the height of a horizontal subdivision above the baseline is not constant, the height of the lowest point of that subdivision should be used in calculating 'v'.

Calculation of factor s_i

The factor s_i is given by an integral formula. Applying numerical integration, the following is obtained:

$$s_i = \sum_{j \in n} w_j\, s\,(T_j) \tag{12}$$

where

n — number of draughts used in calculating s_i

$w_1, w_2, ..., w_n$ — weighting factors, depending on distribution density of draughts at the moment of collision and the number of draughts used, and

$$\sum_{j \in n} w_j = 1$$

The present calculation of the attained subdivision index A for dry cargo ships [8] is based on two draughts only. Because of the fact, however, that

- flooding information for the master is supposed to be carried out for a wide range of draughts and
- the problems arising with the determination of KG_{max} value if the calculation of the subdivision index is based on two draughts only

three draughts instead of two are desired to be employed in this paragraph. It must be remembered that three draughts of five are used in IMO resolution A.265(VIII) for calculating A index, although passenger ships have much lesser draught variation than cargo ships. Three draughts are entirely sufficient for practical applications.

The two current draughts could be supplemented by the light ship draught. Thus the procedure for the s_t factor could be as follows:

"For each compartment or group of compartments, s_t shall be weighted according to draught consideration as follows:

$$s_t = 0.45 s_l + 0.33 s_p + 0.22 s_b \tag{13}$$

where s_l, s_p, s_b are the s_T factors calculated for the ship at full, partial and ballast (light ship) load conditions at arrival".

Partial loading condition is a load condition with 60% utilisation of load capacity. Each loading condition should correspond to *arrival* condition as most collisions happen in the proximity of ports. For this reason, all tanks intended for consumable liquids should be assumed to be empty, to err on the side of safety.

The weighting factors in formula (13) were originally derived assuming a triangular distribution of draughts; they represent the relative frequency of a ship operating at the given loading condition. If further operating data is analysed, these factors might be altered. However, the influence of these weighting factors have been found to be modest.

For ships carrying dangerous goods, only two draughts may be adopted corresponding to full load condition and ballast condition, with weighting factors taken as 0.50.

Permeability

The operational performance of dry cargo ships is characterised by substantial differences in permeabilities, depending on the type of cargo. This feature is crucial for the safety assessment and, therefore, should be accounted for in a satisfactory manner, otherwise the subdivision index will largely be distorted as a measure of subdivision of the ship. Due to these reasons, the suggested values of permeability μ based on actual data for dry cargo spaces are as follows:

Volumetric permeability

Spaces	Net	Gross
intended for ro-ro cargo	0.85	0.90
intended for containers	0.80	0.85
intended for other dry cargo	0.60	0.70

For lighter draughts, some cargo holds shall be assumed to be empty, according to the anticipated loading conditions. Permeabilities of other spaces can be based on common standards as recommended in [7, 8].

The actual volumetric permeability can be represented by:

$$\mu = 0.95 - \frac{V_{net}}{V_{gross}} (0.95 - \mu_{net}) \tag{14}$$

but not more than the gross value, where V_{net} and V_{gross} are net (occupied by cargo) and gross (theoretical) volumes of the cargo hold. Permeability of net volume of containers average at about 0.71 [28] and such a value is recommended by the Polish Register of Shipping [29]. Necessary data for ro-ro ships can be found in [30]. Differentiating between net and gross value of permeability is necessary to allow for different cargo hold arrangements like double sides or double deck. There is no need, however, to vary permeability with the draught [28].

The above values of volumetric permeability are definitely to excessive for the assessment of damaged stability, particularly in the case of containerships and partly of ro-ro vessels as the water trapped inside containers contributes nearly nothing to the loss of moment of inertia of the waterplane. The walls of containers operate here as weathertight bulkheads reducing largely the free surface effect. This is why there is need to defferentiate between the two cases and introduce volumetric and surface permeabilities: The former is simply the degree of damage volume occupied by the flooded water and applies for sinkage calculations, the latter - the degree of damage waterplane occupied by the free surface and applies for stability calculations. The suggested values of surface permeability are as follows:

Surface permeability

Spaces	Net	Gross
intended for ro-ro cargo	0.75	0.85
intended for containers	0.45	0.55
intended for other dry cargo	0.60	0.70

The actual surface permeability for containerships can be assessed more accurately assuming $\mu_{net} = 0$ in formula (14), that is, as if cargo was impermeable. Thus:

$$\mu_s = 0.95 \left(1 - \frac{V_{net}}{V_{gross}}\right)$$

which may vary from about 0.35 for the midship holds up to about 0.65 for the holds nearest the ends of the ship.

These are realistic values of permeabilities for dry cargo holds and they should be taken for the assessment of ship safety in the damaged condition. Although in the case of ro-ro ships the increase in permeability is substantial, they are capable of meeting the safety standard without restricting their design features if there are provisions for reserve buoyancy above the vehicle deck [33].

In the case of ships carrying harmful cargo, it may be assumed that liquid cargo in a damaged hold is replaced by water.

CONCLUDING REMARKS

The paper provides a detailed analysis of a part of the probabilistic regulations for ship subdivision, dealing with damage stability assessment. The background to the present damage stability criteria has been discussed, emphasizing the need for more research in this area in order to arrive at a criterion based on a theoretical model. These possibilities were briefly discussed. But even if we knew a strict physical criterion, the whole procedure for the assessment of damage stability safety would still be far from perfect if we do not remove a number of omissions and weaknesses existing in the present regulations and indicated in the paper; in particular, if the proper angle of heel and mode of calculations are not specified in the regulations.

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Numerical Prediction of Roll Motion of a Ship with Liquids on Board in Regular Waves from Different Directions

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Abstract

In this paper the roll motion of a fishing vessel with free surface liquids on board is analysed both in a regular beam sea and in a longitudinal sea.

Firstly an analytical model based on the Lagrange equation representing the roll motion of the ship with free surface liquids has been developed. Successively a hydrodynamic approach involving the roll motion equation of the ship in conjunction with the Navier-Stokes equations for the determination of the sloshing loads, has been used, and computations were carried out considering the 1:12.5 model of a fishing vessel equipped with a rectangular tank.

In addition the effect of viscosity of the shipped liquids on the overall sloshing induced loads has been investigated in both the sea conditions.

1. Introduction

The analysis of the dynamic response of a ship with free surface liquids on board is generally a difficult task due to a strong interaction between the liquid oscillation and the ship motion. On the other hand the problem is important for several classes of ships, for instance for tankers of the last generation due to the sloshing induced loads over the walls of the tanks, for the analysis of damage survivability, for small boats with water shipped on deck.

In the past several approaches have been used for the determination of the stability characteristics of ships with free surface liquids, with particular regard to capsizing.

In particular they involve static, dynamic and hydrodynamic approaches[1], depending on the mathematical model used for the description of the liquid motion inside the compartment.

The first of them [2, 3] treats the liquid motion as a static shifting of the free surface according to the ship heeling, which means that the oscillation of the liquid is kept into account as a correction for free surface effects to the righting moment of the ship. It is known that this method allows to predict the static characteristics of the ship.

If one is interested to the behaviour in a seaway of a ship with free surface liquids on board, more accurate predictions can be obtained by the use of a mathematical model comprehensive of the dynamic contributions derived by the liquid oscillation.

In a simplified analysis the roll motion of the ship with free surface liquids can be described by a set of two coupled differential equations. In this case the roll angle and the slope of the free surface of the liquid in the tank are the unknown quantities. In the past, such a method, called 'dynamic', has been extensively used in the study of passive antirolling tanks [4, 5], and generally it has been developed for U-shaped tanks.

This last circumstance follows from the difficulties deriving by the reduction of the system of partial differential equations describing the liquid motion to an ordinary equation when there are no dominant flow directions, and this is the case of free surface tanks.

The dynamic approach allows to evaluate the main characteristics of the system in terms of natural frequencies of oscillation, corresponding to the square root of the eigenvalues of the coupled equations. Nevertheless the results obtained by the 'dynamic' approach are useful only in a qualitatively sense, due to the simplifications used for the determination of the ship-sloshing interaction. In particular the mathematical model representing the oscillation of the liquid involves the restrictive hypothesis that the amplitude of the free surface motion is small, reducing the physical system to a double pendulum equivalent system.

Finally more accurate predictions can be derived by solving the sloshing problem using an appropriate hydrodynamic model [6].

The problem is generally solved as follows: the roll angle, velocity and acceleration are calculated at each time step by means of a computer code solving the ship motions. Then those values are given as input to a subroutine that solves the sloshing problem. Finally the resulting moment, obtained by integrating the pressure field over the walls and the bottom of the tank is added as an exciting moment at the next time step in the ship motions calculation.

Several levels of accuracy can be derived by the use of the mentioned hydrodynamic method, depending on the mathematical model used both for the description of the ship motions and for the simumlation of the sloshing problem.

Actually the solution of the fully non linear three dimensional problem representing the ship motions in a seaway is unpracticable, and some hypotheses have to be done for the prediction of the ship response. In particular, if we are interested in the calculation of ship rolling, a system of three coupled equations representing the heave, sway and roll motions can be used. Nevertheless due to the difficulties arising in the calculations of some of the coupling coefficients of the ship motion equations, a more simplified model involving the uncoupled nonlinear roll equation can be considered. In the latter case the most stringent hypotheses consist in assuming known and fixed the roll axis and in neglecting the effects of the sway motion of the ship on the sloshing of the liquid.

Sloshing loads can be evaluated in several ways, depending on the geometry of the compartment, on the physical characteristics of the liquid and on the external excitation [7].

In some simple cases (constant mass of liquid, unbaffled tanks filled enough to eliminate the appearence of hydraulic jumps and in the presence of moderate amplitudes of excitation) the potential theory can be successfully used. Unfortunately, most of practical applications are characterized by complex geometries and physical conditions as well, suggesting the use of more sophisticated mathematical models.

Actually at the Department of Naval Architecture of the University of Trieste, the analysis of the roll response of a ship with liquids on boards is extensively carried out, both theoretically and by experiments. The topic of such an analysis is twofold: a complete understanding of the phenomenon resulting by the continuous tranfer of kinetic and potential energy between the ship motions and the liquid oscillation is the primary objective, and the second is the generation of an easy to use tool allowing the prediction of the characteristics of a ship with free surface liquids on board with respect to capsizing.

In the first stage of the project the stability of a fishing vessel with water on deck has been analysed. In order to solve efficiently the hydraulic jump which appears in a wide range of frequencies of excitation, the sloshing loads are calculated solving the hyperbolic shallow water equations. A numerical procedure solving the previous equations in conjunction with the non linear uncoupled roll motion equation has been developed [8] and at the same time experimental tests have been carried out using a 2.0 meters long model of a fishing vessel in beam sea [9]. Comparisons between numerical and experimental results for the free oscillation tests agreed very well.

In the meantime a more powerful method than that used in [8] for the solution of the shallow water equations has been implemented, in order to allows accurate solutions both in the two- and in the three-dimensional cases [10].

Successively a mathematical model for the high fill depth case has been developed using the potential formulation in the hypothesis of inviscid liquids [11]. Comparisons between computed and measured roll angles [12] in beam sea agree very well, except in the steep part of the resonance zone, where the large amplitude sloshing motions can not be well described.

The results obtained in the previous cases show that the main feature deriving from the presence of free surface liquids is the shifting of the resonance frequency in the range of large encounter periods, depending on the amount of liquid on board and on its own position on respect to the centre of mass of the ship. Moreover, in the case of water on deck an amplification of the maximum roll angle with respect to the 'frozen liquid' condition has been observed [13].

In this paper the roll motion of a ship with free surface liquids is studied in detail both in beam and in longitudinal sea.

In the first stage an analytical method based on the Lagrange formulation of the roll motion with free surface liquids has been developed. Despite its own simplicity the method allows to obtain powerful qualitative informations about the behaviour of the dynamic system, in terms of new resonance conditions.

Then, a more realistic evaluation of the roll motion of the ship with free surface liquids on board, has been made using a hydrodynamic approach. In particular the calculation of the sloshing loads has been performed by solving the fully Navier-Stokes equations with the appropriate non linear boundary conditions at the free surface.

Finally a parametric excitation has been considered for a ship sailing in a following regular sea. The threshold curves for the stability of the roll motion of the ship have been evaluated solving the previous analytical model by the 'harmonic balance method' and by means of the fully hydrodynamic procedure.

2. Mathematical Model

analysis in beam sea:

As previously discussed an analytical model to describe the dynamic behaviour of a ship with a free surface liquid has been developed. In the past such kind of models has been developed for U shaped tank [4] using the assumption that the equations of motion of the liquid can be simplified along the line representing the main direction of the flow. Unfortunately the previous models fail when the analysis of a free surface tank is performed, mainly due to the lacking of a dominant flow of the particles of the liquid inside the compartment.

In this case a new model reducing the liquid motion to the oscillation of the centre of mass of the liquid itself has been developed.

In particular the following hypotheses, deriving from the assumption of small amplitude oscillations, have been made:

- The roll motion is considered uncoupled from the others.

- The free surface of the liquid in the tank is considered flat during the motion.

An inertial frame of reference with origin in the centre of mass of the ship with the free surface liquid on board has been chosen. Reducing the liquid motion to the oscillation of its own centre of mass it follows:

$$x_{GL} = \frac{b^2}{12\ h} tg\Psi \cos\Phi - \left[\frac{h}{2} - R + \frac{b^2 tg^2\Psi}{24\ h}\right]\sin\Phi \tag{1}$$

$$y_{GL} = \frac{b^2}{12\ h} tg\Psi \sin\Phi + \left[\frac{h}{2} - R + \frac{b^2 tg^2\Psi}{24\ h}\right]\cos\Phi \tag{2}$$

being Φ and Ψ respectively the roll angle and the angle between the free surface of the liquid and the bottom of the tank , b the tank breadth, h the liquid depth, R the distance between the center of gravity of the ship and the bottom of the tank and x_{GL}, y_{GL} the coordinates of the centre of mass of the liquid.

The Lagrangian function describing the dynamical system constituted by the ship with a free surface liquid on board will be:

$$L = T - U = \frac{1}{2}\left[a_1 \dot{\Phi}^2 + \rho\ b\ h\ l\ \left(\dot{x}_{GL}^2 + \dot{y}_{GL}^2\right)\right] - \frac{a_3\ \Phi^2}{2} - \rho\ b\ h\ l\ g\ \dot{y}_{GL}^2 \tag{3}$$

being ρ the density of the liquid, g the acceleration of gravity, l the lenght of the tank and a_1 and a_3 respectively the inertial and the restoring coefficients of the roll motion of the ship.

If we are interested in the small amplitude motion of the system about the position of stable equilibrium ($\Phi = 0$, $\Psi = 0$) we can consider the linearized Lagrangian function and the following relation:

$$\frac{d}{dt}\frac{\partial L}{\partial \dot{q}_h} - \frac{\partial L}{\partial q_h} = Q_h \qquad (h=1,2) \tag{4}$$

being q_h the generalized coordinates represented in our case by Φ and Ψ, and Q_h the component of forces that cannot be derived by a potential (dissipative and time dependent forces), associated to the h-coordinate.

By means of Eq. 4 we obtain the following differential equations with constant coefficients representing the roll motion of a ship with free surface liquids on board:

$$\begin{bmatrix} a_1 & c_1 \\ c_1 & b_1 \end{bmatrix}\begin{pmatrix} \ddot{\Phi} \\ \ddot{\Psi} \end{pmatrix} + \begin{bmatrix} a_2 & 0 \\ 0 & b_2 \end{bmatrix}\begin{pmatrix} \dot{\Phi} \\ \dot{\Psi} \end{pmatrix} + \begin{bmatrix} a_3 & c_3 \\ c_3 & b_3 \end{bmatrix}\begin{pmatrix} \Phi \\ \Psi \end{pmatrix} = \begin{pmatrix} M_0\ e^{i\omega_e t} \\ 0 \end{pmatrix} \tag{5}$$

being the coefficients a_i, b_i, c_i given in Appendix 1.

By solving Eq 5 we can calculate the maximum roll angle at each value of the encounter frequency. Of course due to the simplification introduced, the results of (5) can be used only in a qualitative sense. Nevertheless the new resonance conditions of the roll motion of the ship can be accurately evaluated by means of the eigenvalues of (5):

$$\lambda_{1,2}^2 = \frac{b_1\ a_3 + a_1\ b_3 - 2\ c_1\ b_3 \pm \sqrt{D}}{2\ \left(a_1\ b_1 - c_1^2\right)} \tag{6}$$

with $D = (b_1\ a_3 + a_1\ b_3 - 2\ c_1\ b_3)^2 - 4\ \left(a_1\ b_1 - c_1^2\right)\left(a_3\ b_3 - c_3^2\right)$,

being the frequencies of resonance the square roots of (6).

Since only qualitative results are available from the analytical model just described a more sophisticated mathematical model that allows an accurate prediction of the roll motion of the ship and of the magnitude of the dynamic sloshing loads on the walls of the tank has been developed.

Actually the uncoupled non linear roll equation is used, in conjunction with a classical ship motion code for the evaluation of the frequency dependent radiation and diffraction coefficients due to the hydrodynamic interaction between the ship hull and the neighbouring incident waves.

The sloshing loads are calculated solving the two dimensional Navier-Stokes equations using a new algorithm able to solve accurately the boundary layer at the rigid walls and the free surface waves as well [14-16].

The matching between the roll motion of the ship and the sloshing in the tank is performed considering in the roll motion equation an exciting moment derived by integrating the pressure field over the rigid walls of the tank:

$$a_1 \ddot{\Phi} + a_2 \dot{\Phi} + R(\Phi) = F_0 \sin(\omega_e t) + M(\Phi, \dot{\Phi}, \ddot{\Phi}) \tag{7}$$

In Eq. 7 a_1 represents the sum of the inertial moment of mass of the ship about the centre of mass of the ship, and of the added mass coefficient, a_2 the damping coefficient calculated as the sum of the radiation contribution and the part due to the viscosity of water, $R(\Phi)$ the non linear restoring moment expressed as follows:

$$R(\Phi) = \alpha_1 \Phi + \alpha_3 \Phi^3 \ , \tag{8}$$

F_0 the amplitude of the exciting moment comprehensive of the contribution of diffraction and M the moment due to the sloshing of the liquid in the tank.

Eq. 7 is solved in the time domain by a fourth order Runge-Kutta method. At each time step the calculated roll angle, velocity and acceleration are given as input to the subroutine that evaluates the liquid motion in the tank by solving the Navier-Stokes equations, finally the sloshing moment about the roll axis is calculated and the cycle can start again.

analysis in longitudinal sea:

It is well known that a ship in following sea can experience large amplitude motions, often leading to broaching-to phenomena or capsizing.

Whereas the first is the result of several interacting causes , i. e. loss of directional stability in waves, loss of efficiency of the control surfaces, yawing moment due to the incident waves and loss of stability [17], the second is usually related to loss of static stability of the ship in waves or to the phenomenon of parametric resonance. In the following we will focus our attention on the last circumstance.

Froude [18] was the first to understand the dangerous effects that the following sea can produce on the stability of a ship on respect to capsizing. More than thirty years ago Grim [19] and Paulling [20] have given a rational theory that explains the mechanism of parametric resonance in terms of continuous tranfer of energy from the wave system to the roll motion of the ship through its vertical motions, when particular resonance conditions are verified.

In the mentioned studies it has been demonstrated that a wave of length equal to the ship hull length, coming from the stern and encountering the ship with a frequency twice the natural roll frequency can give rise to the loss of stability of the upright position $\Phi=0$.

Starting from the previous studies, in the past this phenomenon has been carefully analysed [21, 22, 23] including the effects of the interaction among the heave, pitch and roll motions.

From a physical point of view, it has been proved that the periodic variation of the restoring moment of the roll motion of the ship is the main cause of the loss of stability of the ship in following sea. In order to deal with a detailed description of the phenomenon, the coupled heave, pitch and roll equations in the hypotheses of low frequency and large amplitude excitation should be solved in the time domain. This approach needs the knowledge of the coupling coefficients between the vertical motions and the roll motion, in particular in the case of large amplitude slow motions. Moreover if the analysis of a ship with free surface liquids is carried out more complications are added due to the coupling between the liquid sloshing and the ship motions.

On the other hand focussing the attention on the analysis of the conditions that can cause the loss of stability of the roll motion of the ship in following sea, a simplified linearized approach can be adopted. The whole problem can be split into two parts. The first in which the vertical motion of the ship are calculated using a ship motion code, so that the relative position between the ship and the incident waves can be obtained, and the second that solves the uncoupled equation of the roll motion with parametric excitation in order to find the threshold zones for the stability of the roll motion.

Using this simplified approach the improvement with respect to the traditional methods [19,20] consists in including the dynamic effects produced by the vertical motions in the calculation of the relative variation of the transversal metacentric height [24].

Due to the complexity introduced by the coupling of the roll motion with the sloshing of the liquid inside the compartment, in this paper we use the simplified method above described. In particular the uncoupled linearized roll motion equation is considered in conjunction with the contribution of the oscillation of the liquid in the tank, the latter calculated neglecting the effects of the vertical motions.

Firstly the dynamic method described in the previous section, resulting in a set of two coupled ordinary differential equations, has been used and a mixed analytical-numerical solution was obtained in order to find the threshold zones of stability.

Finally the hydrodynamic method that involves the roll equation of the ship together with the solution of the Navier-Stokes equations for the sloshing in the compartment has been used and comparisons between analytical and numerical results have been carried out.

In the case we are going to examine, the equations of the roll motion of a ship with free surface liquids on board are the same as in the previous section (Eq. 5), nevertheless the external excitation $M_0\, e^{(i\omega_e t)}$ will vanish and a periodic variation of the restoring moment of the ship has to be introduced. A further simplification can be made considering a sinusoidal variation of the metacentric height, so we will obtain:

$$a_3 = \Delta\, GM \left[1 + \frac{\delta GM}{\overline{GM}} \sin(\omega_e t)\right] \tag{9}$$

being δGM the maximum variation of the metacentric height due to the wave action and $\overline{GM}$ the mean value in one period. It has to be remarked that the latter is usually different from the still water value.

In the past the determination of the threshold zones for the stability of the upright position of the ship with liquids on board has been carried out using a multiple scale method, and very interesting features have been shown [25]. In particular the fact that the most dangerous situation will occur when the encounter frequency is twice the resonance frequency of the roll motion with free surface liquids on board, and that the latter will act as an additional damping in the resonance conditions of the ship without free surface liquids.

Due to the complexity of the method adopted in [25] only a first order solution has been reached, so reasonable results was derived for small amplitude parametric excitation when the roll and the liquid sloshing damping coefficients were small enough.

In this paper the determination of the threshold zones of stability of the solution $\Phi = 0$ is carried out using the 'Energy Balance Method' [26] that, in the case herein investigated, allows to obtain much more accurate solutions than the previous. A brief description of the method will be given in the following.

The system (5) of coupled equations, written in the form representing the roll motion in following sea of the ship with free surface liquids, is put in a diagonal form, and we seek a solution:

$$\xi_1 = a \sin(\omega_e t) + b \cos(\omega_e t)$$

$$\xi_2 = c \sin(\omega_e t) + d \cos(\omega_e t) \tag{10}$$

in the zone $\omega_e=2*\omega_1$ and $\omega_e=2*\omega_2$, being $\omega_{1,2}$ the natural frequencies of the roll motion of the ship with free surface liquids on board. In 10, $\xi_{1,2}$ represent the unknowns of the diagonal system, and a,b,c,d are constants to be calculated. This operation is made by substuting 10 in the diagonal system and operating an energy balance. Finally the conditions for a stable solution are sought by determining the eigenvalues of the linear algebric system resulting by the above mentioned operations. More details about the method adopted herein are given in [27].

3. Numerical Results

analysis in beam sea:

Numerical simulations have been carried out considering a 1:12.5 model of the fishing vessel 'ANCONA'(Fig. 1) equipped with a rectangular tank filled with different liquids. The main characteristics of the model and of the tank are the following:

Ship model:

L = 2.0 m
B = 0.552 m
D = 0.397 m
T = 0.215 m
Mass = 100 Kg
KG = 0.210 m
GM = 0.077 m

Rectangular tank:

breadth = 0.42 m
length = 0.40 m
height = 0.280 m
liquid depth = 0.12 m

By performing the hydrostatic calculations it has been found out that the GZ curve is linear in a wide range of angles of heeling (Fig. 2), so the restoring coefficient in the roll equation can be written as $a_3 = \Delta$ GM without introducing any approximation.

The bottom of the tank is put 0.079 m below the centre of mass of the ship and two different liquids are considered inside the tank:

Reginol oil : $\rho = 860$ kg/m^3, $\nu = 0.000435$ m^2/s
85% glycerol/water solution : $\rho = 1230$ kg/m^3, $\nu = 0.00011$ m^2/s

The choice of such liquids is mostly due to the CPU time needed in the numerical simulations when more realistic liquids are used. Simulations involving the presence of water inside a compartment will be made in the next future.

Due to the presence of the liquid, the mass, and the metacentric height of the ship will change, so we obtain:

Shipping of Reginol oil:
Total Mass = 117.3 Kg
GM = 0.079 m

Shipping of 85% g/w sol. :
Total Mass = 124.8 Kg
GM = 0.081 m

At the two different load conditions the hydrodynamic added mass, damping and exciting force have been calculated by means of a ship motions code that uses the well established strip theory, while the viscous part of the damping coefficients has been obtained using the semi-empirical formulations of Ikeda et al. [28]. Whereas in this work a linearized damping coefficients has been used, by experiments has been observed that a more sophisticated non linear model should be used for an accurate prediction of the roll response near resonance.

Firstly free rolling numerical tests have been carried out, considering the ship model without any liquid on board and with reginol oil shipped in the tank previously described. The time domain roll angles are shown respectively in Fig. 3 and 4. It can be noticed that the natural roll period changes due to the effect of liquid sloshing, from T_{nat} = 1.38 s to T_{nat} = 1.6 s. In particular it has been observed that the shifting of the natural period will be larger by increasing the distance between the centres of mass of the ship and of the shipped liquid.

Successively the case of the ship in beam sea has been considered and calculations have been made by means of the analytical and of the hydrodynamic models.

In both cases, as previously discussed, the radiation and diffraction coefficients of the roll motion have been calculated using the strip theory. The roll angle versus the frequency of the incident waves is plotted in Fig. 5 considering the presence of reginol oil and 1/100 height to length ratio waves.

As previously discussed, it is shown that the dynamic approach predicts unrealistic roll angles in particular near resonance, whereas an excellent agreement with the more reasonable hydrodynamic approach has been found out in the calculation of the natural period of oscillation. This last circumstance allows the use of the simplified analytical model in the first stage of the ship design, in order to find the new resonance conditions due to the presence of free surface liquids.

Moreover the effect of viscosity and density of the free surface liquid on board with respect to the roll characteristics of the ship has been investigated. It is known that in sloshing problems, the liquid viscosity places an important role in the reduction of the speed of propagation of surface waves, and in the smoothing of high frequency waves resulting in spray and breaking wave phenomena.

In Fig. 6 the roll angle versus the encounter period is plotted considering the two liquids as previously seen.

Also in this case 1/100 height to lenght ratio waves have been used in the computations. Two different effects are shown, the first resulting in the variation of the roll response near resonance, the second in a weak variation of the phase between the exciting moment and the sloshing induced moment (Fig. 7).

Whereas the first is mostly due to the different densities of the liquids, the second is related to the variation of the velocity of propagation of the waves inside the tank, due to the different values of the kinematic viscosity of the liquids.

Looking at Fig. 7 a phase lag close to $\pi/2$ is observed, between the sloshing induced moment and the heeling moment due to the sea waves, in resonance conditions, whereas the two moments result in opposition of phase in the range of larger encounter frequencies; finally in the range of short waves, corresponding to very large frequencies of excitation, it has been noticed that the liquid inside the tank appears like 'frozen'. The analysis of the phase lag between the roll exciting moment and the sloshing moment suggests that in resonance conditions the liquid oscillation acts as an amplifier of the roll motion of the ship, whereas the latter will result strongly reduced in the range of smaller periods of excitation. Finally in the range of very short waves, the effect of the liquid sloshing on the roll motion of the ship is neglegible. It has to be pointed out that due to the depth and the viscosity of the liquids used in the numerical simulations and to the diffraction coefficients used in the roll motion equation the second resonance peak, rising up in the experimental tests at $\omega_e = \omega_2 = 6.8$ rad/s [12], has not been observed.

By the analysis of the wave systems excited inside the tank, it appears that in resonance conditions a train of travelling waves, often resulting in an hydraulic jump, superposed to a large standing wave, rises up, depending on the magnitude of the external excitation.

In this case the main part of the sloshing induced moment is due to the large standing waves, whereas the short and fast waves can induce impact loads over the walls of the tank, resulting in risks of structural damages. So, it is remarkable that the problem of the interaction of sloshing of liquids inside a compartment and the roll motion is important in the analysis of the safety of a ship in a more general sense than capsizing.

The appearance of the high speed short waves inside the tank is related to the magnitude of the external excitation and to the liquid viscosity. The increasing of the latter smoothes the short waves, so weak impact loads may be expected.

Longitudinal sea:

The analytical and the numerical analysis of the roll motion of the ship with liquids on board has been carried out also in following sea, using the mathematical models as previously described.

At this stage of the work the possibility of occurrence of unstable roll motion of the ship in longitudinal sea has been investigated by analysing the threshold zones of the relative variation of the metacentric heigth. Of course, as previously pointed out the study of large amplitude rolling motion in the mentioned conditions of excitation needs a more complete mathematical model than that used in this work.

In particular the same ship model as described in the previous section has been used, and the shipping of a 0.12 m depth reginol oil was considered. Comparisons between the analytical and numerical threshold curves in the most dangerous subharmonic resonance are presented in Fig. 8. It can be noticed that a 23% relative variation of the righting arm of the ship is enough to excite the parametric instability in the most critical condition ($\omega_e=2*\omega_1$). By the analysis of these preliminary results it is shown that the effect of shipping of free surface liquids on board is in a strong shifting of the parametric resonance in the range of lower encounter frequencies on respect to the case of a ship without free surface liquids on board.

Moreover the numerical results have shown that the effect of viscosity of the liquid is neglegible relatively to the shifting of the frequency for the occurrence of the parametric resonance, neverthleless, it can have some importance in the variation of the threshold values for the stability of the roll motion.

The situation can appears completely different when the free surface liquid is shipped in a baffled tank, and in general the latter represents a more realistic situation to analyse. Infact the presence of horizontal or vertical baffles can induce a strong variation in the natural frequency of oscillation of the liquid in the tank, and in this case viscosity plays a crucial role in the dissipation of the kinetic energy of the liquid. The analysis of this more interesting situation is in progress.

4. Conclusions

In this paper the analysis of the roll motion of a ship with free surface liquids on board has been carried out both in regular beam sea and in following sea.

An analytical model and a hydrodynamic approach have been developed. The first was derived for free surface any shaped tanks using the Lagrange equation representing the roll motion of the dynamic system ship-free surface liquid. The other has been obtained by the matching of a Navier-Stokes solver, for the determination of the sloshing induced moment, with the uncoupled equation of the roll motion of the ship.

As shown in previous studies it has been found out that the presence of a free surface liquid on board acts as a mechanism that shifts the natural roll period of the ship depending on the geometry of the tank, its own position on respect to the centre of mass of the ship and on the liquid depth.

Moreover it has been observed that several wave systems are generally excited inside the tank, resulting both in a large induced heeling moment and in impact loads on the walls of the compartment.

Preliminary results in longitudinal sea have shown that the presence of free surface liquids on board reduces the value of the encounter frequency at which the parametric resonance may occur.

Due to the effects that the sway motion of the ship can cause on the sloshing of liquids in compartments, in the next future the interaction between roll, sway and liquid motion will be studied, both by a numerical approach and by experiments.

Finally, actually the analysis of the sloshing induced loads in the case of baffled tanks is in progress. It is believed that the presence of internal rigid structures inside a tank can change drastically the roll response of the ship with respect to the case of unbaffled tanks.

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Appendix 1

In the following the coefficients of Eq. 5 are reported:

$$a_1 = I + \left(\frac{h}{2} - R\right)^2 Q$$

$$a_2 = 2\ X_s \sqrt{a_1\ a_3}$$

$$a_3 = \Delta\ GM - \left(\frac{h}{2} - R\right) Q\ h$$

$$b_1 = Q \left(\frac{b^2}{12\ h}\right)^2$$

$$b_2 = 2\ X_T \sqrt{b_1\ b_3}$$

$$b_3 = \frac{Q\ g\ b^2}{12\ h}$$

$$c_1 = \frac{Q\ b^2}{12\ h} \left(\frac{h}{2} - R\right)$$

$$c_3 = \frac{Q\ g\ b^2}{12\ h}$$

being:

I the inertial moment of mass of the ship about her centre of mass, **h** the liquid depth inside the tank, **R** the distance between the centres of mass of the ship and of the liquid, **Q** the mass of the shipped liquid, **b** the tank breadth, **Δ** the ship displacement, **GM** the trasversal metacentric height, X_s and X_T the non dimensional damping coefficients, respectively for the roll

Whereas X_S can be easily calculated using the strip theory, the determination of X_T is quite a difficult task, being related to the knowledge of the dissipation rate of energy during the liquid sloshing. Nevertheless the value of X_T affects the curve of the roll response only near resonance, where the analytical model becomes not consistent.

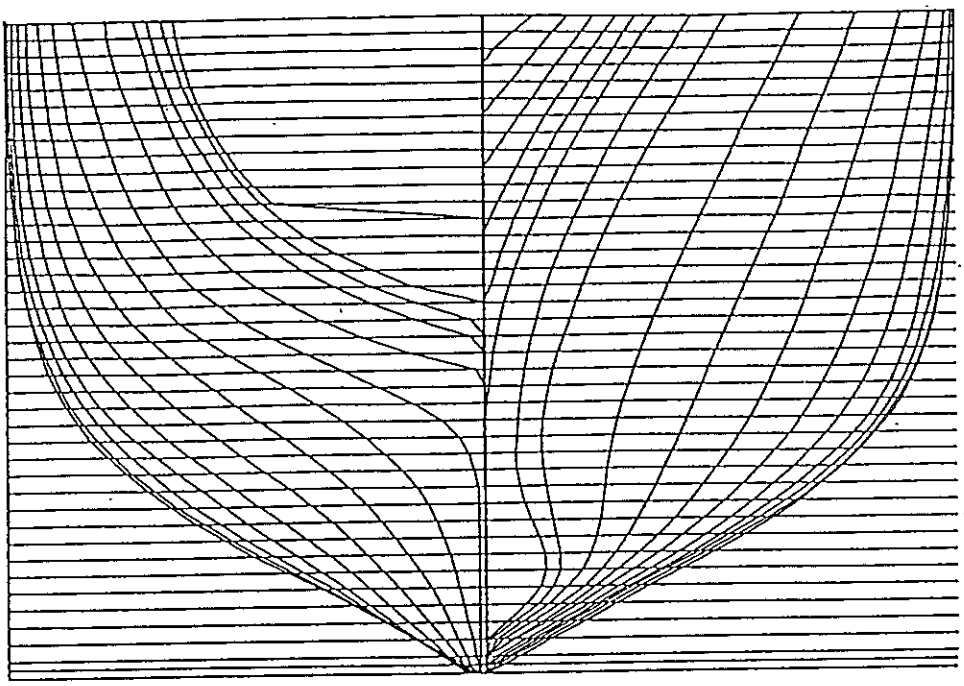

Fig. 1 Body plan of the fishing vessel 'ANCONA', used in our computations

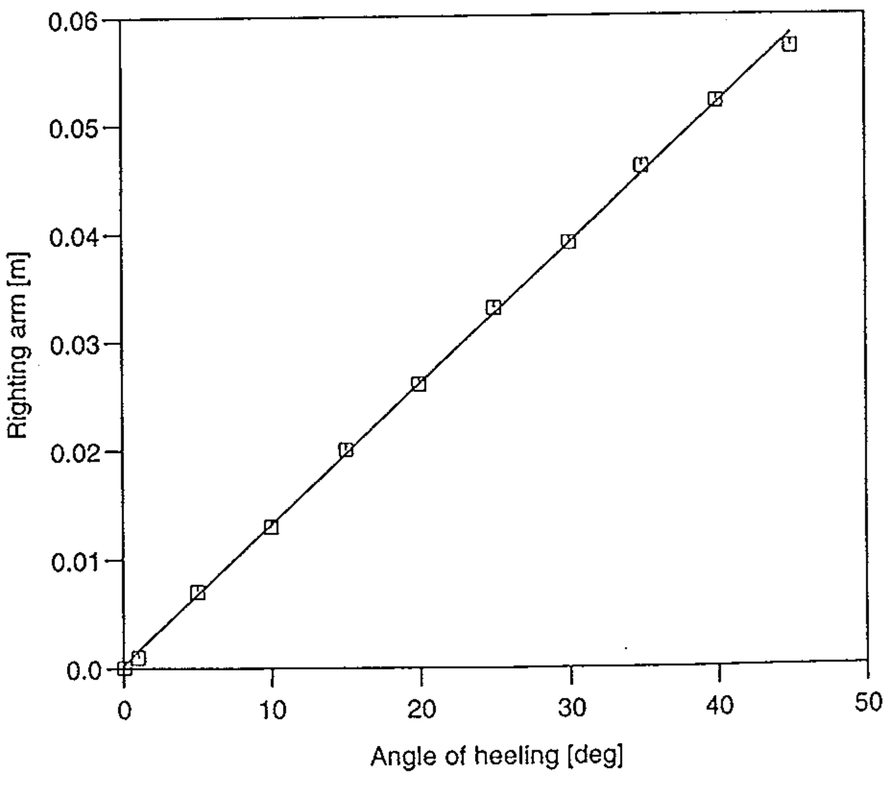

FiG. 2 Trasversal righting arms of the fishing vessels at the initial load conditions (without any liquid).

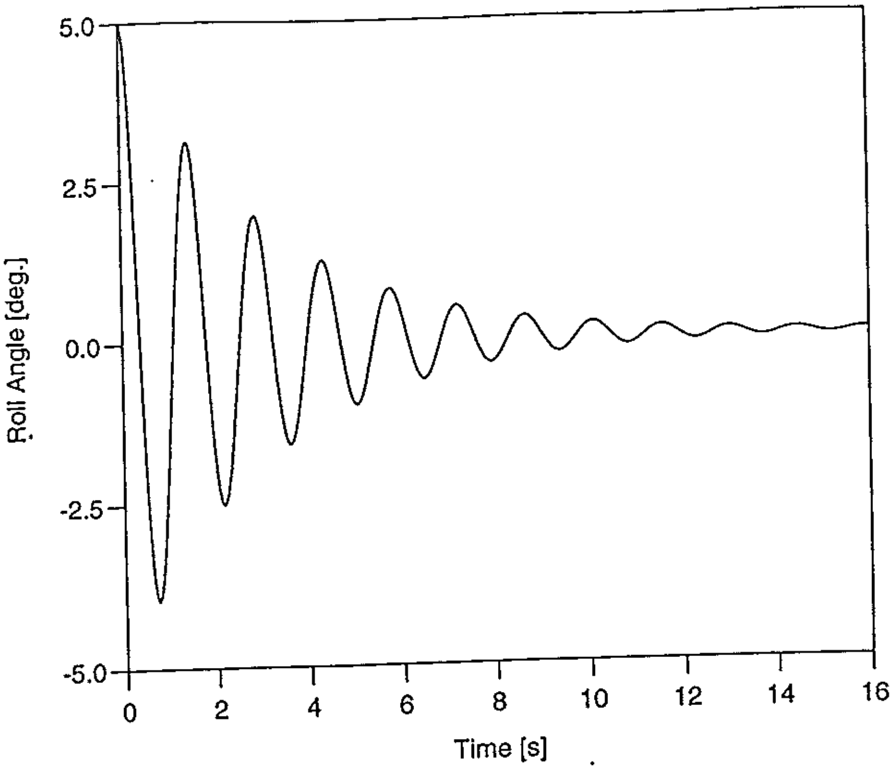

Fig. 3 Computed roll angles in the roll decay test, in the case of ship without free surface liquids on board: (T_{nat} = 1.38 s).

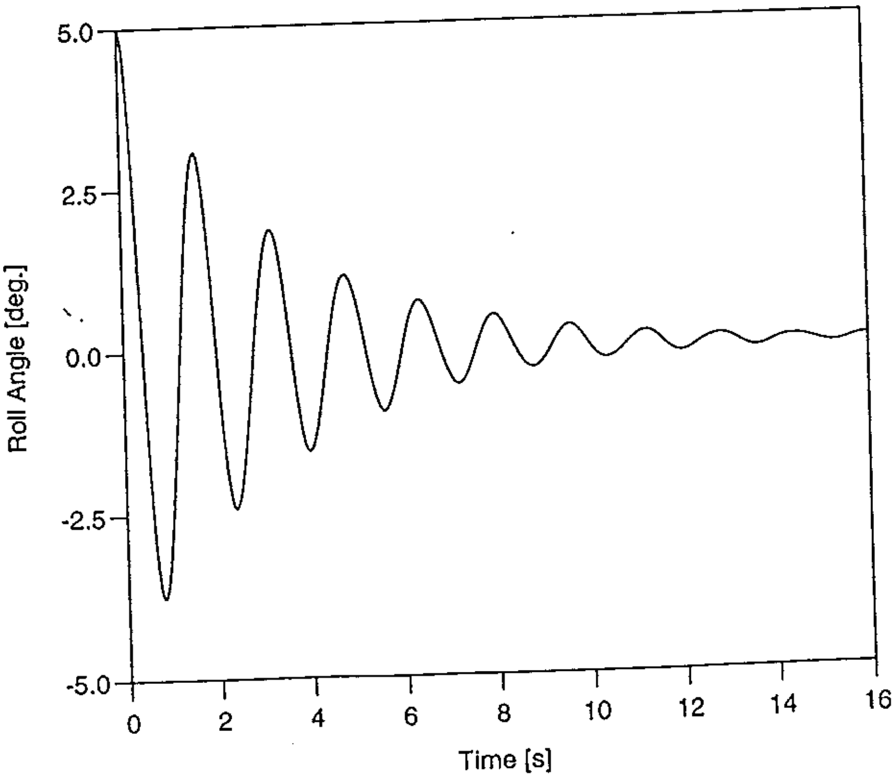

Fig. 4 Computed roll angles in the roll decay test, in the case of ship with a 0.12 m depth reginol oil shipped on board: (T_{nat} = 1.6 s).

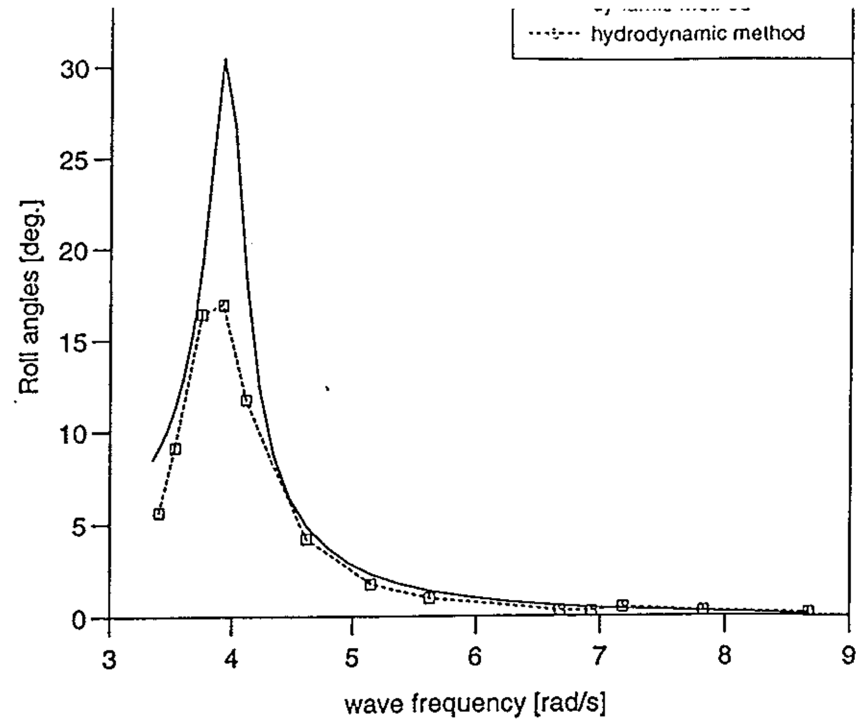

Fig.5 Roll response in the frequencies domain calculated by the analytical and by the hydrodynamic methods:
Ship model with 0.12 m depth reginol oil on board, 1/100 wave steepnes.

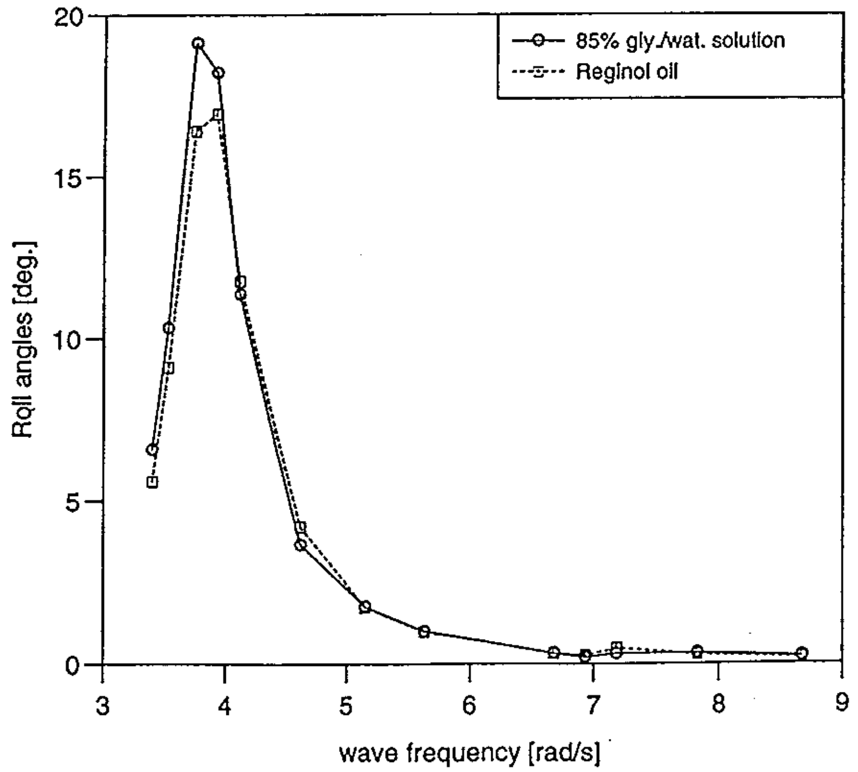

Fig. 6 Roll response in the frequencies domain for the ship model with: 0.12 m depth reginol oil and 0.12 m depth 85% glycerol/water solution.

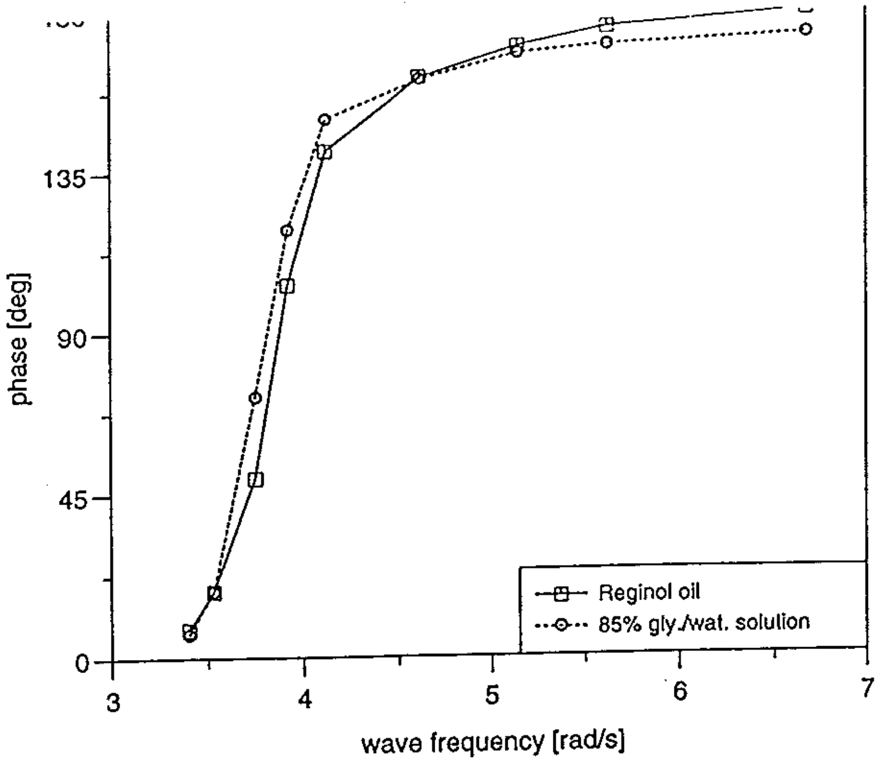

Fig. 7 Phase lag between the exciting moment and the sloshing moment versus encounter frequencies, in the case of shipping of reginol oil and 85% glycerol/water solution.

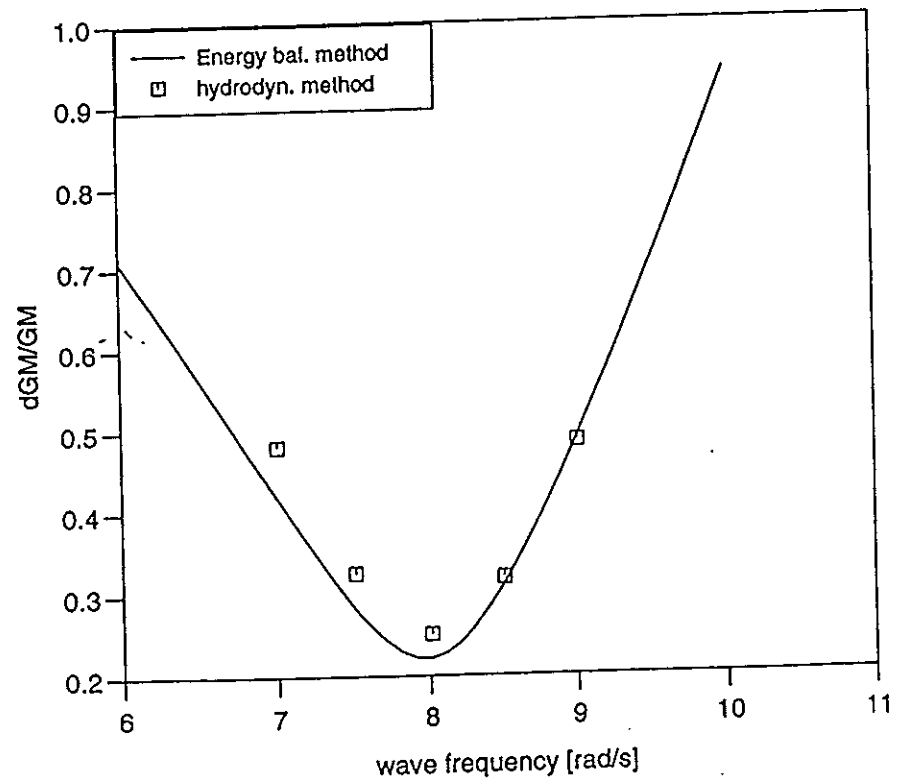

Fig. 8 Analitically and numerically computed threshold values of $\delta GM/GM$ in the parametric resonance conditions $\omega_e=2*\omega_1$

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