DEPARTMENT OF SHIP AND MARINE TECHNOLOGY

NUMERICAL & PHYSICAL SIMULATION OF SHIP CAPSIZE IN HEAVY SEAS
(Collaborative Research Programme between UK & Japan)

24 - 25 July 1995
Ross Priory, University of Strathclyde, Scotland, UK

FINAL PROGRAMME

24 July - Intact Stability

09:00 - 10:00	Arrival of Participants and Coffee
10:00 - 10:15	*Welcome and Introduction to the Workshop:* Prof. Chengi Kuo
10:15 - 12:15	SESSION 1: *Ship Capsize Simulation in Stability Research*
12:30 - 13:30	Lunch
13:45 - 15:45	SESSION 2: *Model Capsize Experiments in Heavy Seas*
15:45 - 16:00	Tea
16:00 - 18:00	SESSION 3: *Non-Linear Dynamics and Ship Capsize*
19:30	Workshop Dinner

25 July - Damage Stability

09:00 - 11:00	SESSION 4: *Simulation of Damaged Ship Motions with Progressive Flooding*
11:00 - 11:15	Coffee
11:15 - 12:15	SESSION 5: *Dynamic Stability Software Validation Techniques*
12:30 - 13:30	Lunch
13:45 - 15:45	SESSION 6: *Model Experiments with Damaged Ship Models*
15:45 - 16:00	Tea
16:00 - 17:00	SESSION 7: *Standardisation of Model Experiments for Extreme Tests*

TECHNICAL SESSIONS

SESSION 1: ***Ship Capsize Simulation Programs for Stability Research***

Discussion Leader: Prof. Olle Rutgersson (KTH, Sweden)
Presenter 1: Dr Jan de Kat (MARIN)
Presenter 2: Prof. Alberto Francescutto (TRIESTE, Italy)

SESSION 2: ***Model Capsize Experiments in Heavy Seas***

Discussion Leader: Dr Ian Dand (BMT Ltd, UK)
Presenter 1: Professor Masami Hamamoto (OSAKA, Japan)
Presenter 2: Mr John Hall (HASLAR, UK)

SESSION 3: ***Non-Linear Dynamics and Ship Capsize***

Discussion Leader: Prof. Yucel Odabasi (ITU, Turkey)
Presenter 1: Prof. Michael Thompson (UCL, UK)
Presenter 2: Dr Naoya Umeda (NRIFE, Japan)

SESSION 4: ***Simulation of Damaged Ship Motions with Progressive Flooding***

Discussion Leader: Prof. Maciej Pawlowski (GDANSK, Poland)
Presenter 1: Dr Dracos Vassalos (STRATHCLYDE, UK)
Presenter 2: Mr Bruce Hutchison (SNAME, USA)

SESSION 5: ***Dynamic Stability Software Validation Techniques***

Discussion Leader: Prof. Apostolos Papanikolaou (NTUA, Greece)
Presentation 1: Prof. Kazuhiko Hasegawa (OSAKA, Japan)

SESSION 6: ***Model Experiments with Damaged Ship Models***

Discussion Leader: Dr Tor Svensen (DNV, Norway)
Presenter 1: Prof. Yoshiho Ikeda (OSAKA PREFECTURE, Japan)
Presenter 2: Mr Michael Schindler (DMI, Denmark)

SESSION 7: ***Standardisation of Model Experiment Techniques for Extreme Tests***

Discussion Leader: Dr Stefan Grochowalski (NRC, Canada)
Presenter 1: Dr Vidar Aanesland (MARINTEK, Norway)

WORKSHOP: *NUMERICAL AND PHYSICAL SIMULATION OF SHIP CAPSIZE IN HEAVY SEAS*

24-25 July 1995
Ross Priory, University of Strathclyde, Scotland, UK

Details of the transportation:

Transportation to and from Ross Priory will be provided to the workshop participants. Details are as follows:

Monday 24 July 1995

Meeting place:

Department of Ship & Marine Technology, University of Strathclyde,
100 Montrose Street, Glasgow

Meeting time: **8.00 am**

Departure time for Ross Priory: **8.15 am**

Note: Transportation for the workshop participants, who are planing to fly to Glasgow on Monday morning, can be arranged, if the flight details are given in advance.

Departure time from Ross Priory: **10.00 pm**
This service is for the participants, who are travelling to Glasgow. Those, whose accommodation is arranged at Ross Priory for the night of 24 July 1995, are advised to take personal belongings with them.

Tuesday 25 July 1995

Meeting place:
Department of Ship & Marine Technology, University of Strathclyde,
100 Montrose Street, Glasgow

Meeting time: **7.15 am**

Departure time for Ross Priory: **7.30 am**

Departure time from Ross Priory: **6.00 pm**
This service is for all the participants, who are travelling to Glasgow. Those, who need to travel on Tuesday evening and require transportation earlier than 6.00 pm, the transportation can be so arranged if advance notice is given.

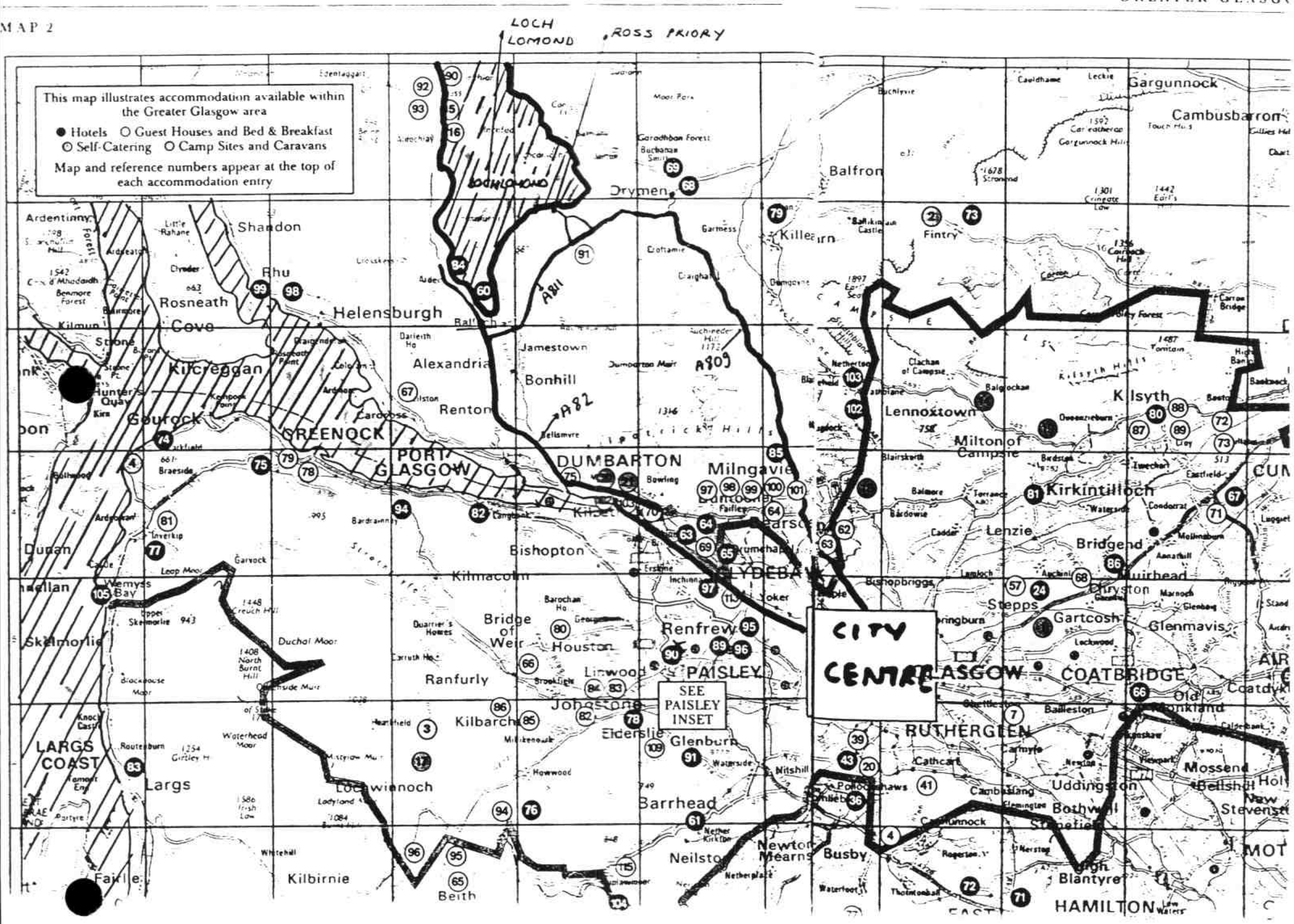
WORKSHOP
NUMERICAL AND PHYSICAL SIMULATION OF SHIP CAPSIZE IN HEAVY SEAS

24-25 July 1995
Ross Priory, University of Strathclyde, Scotland, UK

List of Attendants

	NAME	AFFILIATION
1	Prof. Olle Rutgersson	KTH - Sweden
2	Dr. Jan de Kat	Marin - The Netherlands
3	Dr. Giorgio Contento	Trieste - Italy
4	Pror. Masami Hamamoto	Osaka - Japan
5	Prof. Yücel Odabasi	ITU - Turkey
6	Prof. Michael Thompson	UCL - UK
7	Mr. John Hall	Haslar - UK
8	Prof. Maciej Pawlowski	Gdansk - Poland
9	Mr. Bruce Hutchison	SNAME - USA
10	Prof. Apostolos Papanikolaou	NTUA - Greece
11	Prof. Kazuhiko Hasegawa	Osaka - Japan
12	Dr. Ian Dand	BMT-UK
13	Prof. Yoshiho Ikeda	Osaka pref. - Japan
14	Mr. Michael Schindler	DMI - Denmark
15	Dr. Stefan Grochowalski	NRC - Canada
16	Dr. Vidar Aanesland	Marintek - Norway
17	Dr. Naoya Umeda	NRIFE - Japan
18	Dr. Dracos Vassalos	Strathclyde University - UK
19	Dr. Tor Svensen	DNV - Norway
20	Mr. Paul O'Brien	MOD - UK
21	Dr. Atilla Incecik	Glasgow University - UK
22	Prof. Chengi Kuo	Strathclyde University - UK
23	Dr. Osman Turan	Strathclyde University - UK
24	Mr. Luca Letizia	Strathclyde University - UK
25	Mr. Michael Thangaris	Strathclyde University - UK
26	Mr. Hyun S. Kim	Strathclyde University - UK
27	Mr. Toru Katayama	Osaka pref. - Japan
28	Mr. K. Roby	Osaka pref. - Japan
29	Mr. A. W. Vredeveldt	TNO - The Netherlands
30	Mr. I. Winkle	Glasgow University- UK

MAP 2



MAP FOR GLASGOW - LOCH LOMOND

Session 1

The practical role of time domain capsize simulation

J.O. de Kat
MARIN, Wageningen

1 INTRODUCTION

The purpose of this brief essay is to put into perspective ongoing research involving the simulation of intact ships in waves and wind, including conditions leading to capsize. A major challenge lies in the conversion of theoretical knowledge and numerical models to practical and useful information. One necessary condition for such knowledge transfer to be successful is a strong link between the research world and the "real" world, like ship operators and regulatory bodies.

In conjunction with the present Cooperative Research Navies project on Dynamic Stability (summarized by De Kat et al., 1994), work has been carried out with the following objectives:

- Modeling of capsize physics and risk analysis
- Development of "rational" design criteria
- Development of decision support tools (ship operation)
- Improvement of training/education

The numerical simulation of ship capsizing and broaching plays a central role in this process. To establish confidence levels in the time domain tool developed in recent years, Quality Assurance is essential. This includes the development of QA procedures, benchmark tests, and validation through model tests and full scale data.

Without any further elaboration, I would like to mention that research is carried out along similar lines for damaged ships within the same navy framework. This work relies heavily on the methods and procedures developed for intact ships, as discussed below.

2 CAPSIZE PHYSICS AND RISK ANALYSIS

In a capsize-physics sense, we can distinguish the following aspects:

- Development of simulation model
- Identification of capsize and broaching mechanisms
- Influence of irregular waves on capsizing
- Ocean wave characteristics
- Capsize risk analysis

2.1 Simulation model

The time domain simulation model, embodied in the computer program FREDYN, consists of an extended strip theory approach. It has been described briefly by De Kat et al. (1994) and De Kat (1994). It contains some empirical models, such as propeller-rudder interaction and rudder stall, derived from model tests. General capabilities of the program are given in Table 1.

Mono-hulls with watertight superstructure Zero to moderate speeds (Fn < 0.5) Six degrees of freedom Small and large amplitude motions, including broaching and capsizing Autopilot steering or specified manoeuvres Regular and irregular waves of small to extreme steepness Long-crested seas, any wave direction Prediction of wide range of capsize modes Nonlinear manoeuvring model Roll stabilization devices Skegs Interactive simulations on PC (active control of rudder and RPM) • Dynamic behaviour of damaged ship in waves and wind (1995) • Extensive validation using model tests and full scale measurements

Table 1 *General capabilities of FREDYN (time domain, nonlinear strip theory)*

Platform: the program runs on any computer, including PC. The performance on a Pentium 90 MHz PC is about 20 times faster than real time in a regular wave and a little faster than 1 to 1 in irregular waves.

Some of the current FREDYN developments include the improvement of maneuvering in waves. In certain conditions, especially in stern quartering waves, the maneuvering characteristics of the ship can influence its wave-induced motions such as yaw and roll. As yet there is no theory capable of dealing with this problem comprehensively. Accurately modeling the maneuvering behavior in calm water presents already problems of such complexity (such as the important nonlinear cross-flow drag), that one wonders to what extent the presence of waves compounds these difficulties. For the moment being and foreseeable future, we have to rely on a mix of empiricism and theoretical models for the flow behavior and hull forces in calm water. Traditionally, maneuvering has focused on motions in the horizontal plane only. Recent studies are adding to this the influence of roll - this allows one to predict the yaw-induced moment as a function of heel angle and ship speed, for example; the influence of wave height and period on such coupling is unknown, but may certainly be relevant.

It is evident that as the simulated ship is self-steered, some form of autopilot is necessary. Most autopilot models used in time domain simulations are of a relatively simple nature and may not resemble present-day (possibly adaptive) control algorithms used on board ships. Furthermore, in heavy weather a human is likely to take control of the wheel. Bearing these deficiencies in mind, it is necessary to satisfy oneself that the autopilot in the simulation model yields reasonable behavior, before embarking on extensive simulations.

2.2 Capsize mechanisms

In the analysis of numerous simulations and capsize model test data, various capsize modes have been encountered and "classified" into different categories. A list with such modes is given in Table 2; a number of these modes have been validated by means of model test data. In some cases, however, and particularly in irregular wave conditions, it is not clear to what extent the capsize (or broach) prediction is correct because of the lack of data. To obtain proper capsize validation data in irregular waves, it would be necessary to measure also the spatial wave profile encountered by the ship model (see 2.3). Validation as part of QA is addressed in section 6.

- pure loss of transverse stability in a wave crest
- parametric rolling (low cycle resonance)
- broaching in successive overtaking waves
- surfriding and broaching
- surfriding and loss of stability in wave crest
- surging, yawing and rolling with dynamic loss of stability
- resonant beam waves
- excessive wave-induced roll moment in beam seas

Table 2 *Capsize modes predicted by FREDYN*

2.3 Influence of irregular waves on capsizing

Time domain simulations in irregular waves suggest that the spatial character of the waves is of paramount importance in causing capsizing or broaching (De Kat,1994). Analysis of the wave conditions leading to such events suggests that quite a large range of spatial wave height and wave length combinations can result in the same capsize mode in different realizations of the same sea state, all other conditions (speed, heading) being equal. These waves are considered to be critical waves, having a critical spatial wave length and steepness. We can identify the critical wave characteristics as a function of ship speed, heading angle and loading condition by performing a sufficiently large number of simulations in a given sea state.

Moreover, when the same mode of capsize occurs in a sea state with different significant wave height and peak period, all other conditions being equal, this capsize seems to be caused by waves having the previously identified critical spatial wave length and steepness. In other words, the ship does not care whether it is in a severe or moderate sea state, it will capsize when it encounters a wave system with critical spatial parameters for a given heading and ship speed. Obviously, the probability of encountering critical waves will vary as a function of sea state parameters.

2.4 Ocean waves

Storm wave measurements form an important part in the present work. The simulation model uses the principle of linear superposition to model long-crested irregular waves. In case a large number of wave components is used to define the seaway, the statistical distribution of the simulated wave elevation will be close to Gaussian. The appropriateness to model moderate seaways in this manner is well known, but in storm conditions one would expect large deviations from Gaussianity.

To assess the linear wave model, comparisons have been made between numerical simulations, model basin measurements and ocean wave measurements, including storm seas. For comparison purposes, joint probability distributions provide useful information: wave height versus period, crest height versus wave height, etc. (De Kat, 1994). As far as available, spatial properties have been considered also. The following conclusions have been drawn so far:

- joint probability of wave height and zero-crossing period is predicted well by the linear model, even for storm sea conditions of high significant steepness (where significant steepness is a function of significant wave height and peak period)
- skewness of the wave elevation distribution tends to increase with significant steepness
- crest nonlinearities (and deviations from predictions by linear model) increase with increasing significant steepness and skewness of the wave elevation distribution
- joint probability of wave height and wave length (based on temporal analysis of zero-crossing period at a point) is predicted well by the linear model, even for storm sea conditions of high significant steepness
- joint distribution of spatial wave steepness and spatial wave length compare well with considerations based on temporal analysis at a point for low to moderately steep (in terms of significant steepness) sea states
- for high significant steepness, spatial wave lengths tend to be longer for a given wave period due to nonlinear effects (as is the case for regular Stokes waves); linear theory underpredicts the spatial wave lengths on the average in the order of 10%
- analysis of storm wave properties from measurements performed in the North Atlantic suggests that the mean significant steepness is of a moderate magnitude (De Kat et al., 1994), which implies that in these conditions simulations using the linear superposition principle for waves can model storm wave properties quite realistically

2.5 Capsize risk analysis

From many perspectives there is an interest in teh ability to predict the risk of capsizing. We consider two approaches:

(1) Computation of conditional probabilities of capsizing through a large number of simulations in low to severe sea states (McTaggart, 1995; De Kat et al., 1994).

(2) Probability of occurrence of critical spatial waves; the risk of capsize is assumed to be equal to the risk of encountering waves of critical wave length and steepness - see 2.3 - as a function of heading angle, ship speed and loading condition (De Kat, 1994).

McTaggart (1995), who concentrated on further developing the first approach, points out the possibly important influence of wind on capsize risk analysis. Especially in combined beam seas and winds the likelihood of capsizing is significantly larger than without wind. The risk of capsize determined in this way is quite strongly dependent on the number of realizations (random "seeds") used to define the seaway. The second approach is still in the development stage and seems to be quite feasible.

3 DESIGN GUIDELINES

As is the case for merchant ships, there is a need to revise the intact stability criteria for naval ships. Present criteria are based on outdated hull forms, they do not take into account the potentially dangerous conditions in severe following or stern quartering seas (they are based on combined beam sea and wind capsize events, whence the "weather criterion"), and they provide an unknown and possibly unequal margin of safety for different ships. Through a large series of time domain simulations with systematically varied and actual hull forms, it was possible to derive practical design guidelines that provide an equal measure of safety against capsizing in waves for different ships (in this case frigate-type ships); the procedure for deriving these guidelines has been described by De Kat at al. (1994).

A major finding of this work is the clear connection between wave-induced capsizing and the calm water GZ curve. In contrast to the usual "regulatory" GZ properties, a direct link was found between safety against capsizing and the following large angle properties:

- range of positive stability (i.e., angle of vanishing stability)
- total area underneath the positive GZ curve

Minimum acceptable values have been proposed for the above parameters, where a distinction is made between normal stability conditions (unlimited operation, including heavy weather) and marginal stability conditions (e.g., minimum fuel level, which is assumed to not coincide with operation in heavy seas). It is of interest to note the inclusion of the angle of vanishing stability in the intact stability requirements of the Russian Register of Shipping Rules.

At present, the various navies are evaluating the proposed guidelines (as a supplement to existing criteria) in terms of existing ships. The intention is to apply the same procedure in the development of intact stability criteria for ships with low length-to-beam ratio (L/B < 6) and moderate block coefficient.

4 SHIP OPERATION SUPPORT

In an attempt to provide practical information to ship officers, a methodology has been developed for producing polar diagrams on the basis of a matrix of time domain simulations. These diagrams apply to an individual ship with a given loading condition, and indicate the following as a function of ship speed and heading angle in a seaway: likelihood of broaching, surfriding, motion amplitudes and capsizing. Developments for on-board support continue.

5 EDUCATION / TRAINING

Time domain simulation tools can be used also for educational and training purposes. In this respect, efforts have focused on the following developments:

- Videos with high-resolution 3-D animations of ships in waves and wind (including what-if scenarios resulting in, for example, broaching or loss of stability conditions)
- Interactive simulation on PC (active control over propeller RPM and rudder, with 3-D graphic display of ship in waves)
- Seminars at ship officer schools

6 QUALITY ASSURANCE

As the time domain program FREDYN is evolving from a research program toward a practical decision making tool in ship design and operation, it is necessary to assure its quality and establish its confidence levels. Guidelines have been generated for program development and validation procedures, including the definition of standard benchmark tests. The QA process has been started for the simulation of intact ships. In due time, a similar approach will be applied to the simulation of damaged ships in waves and wind.

QA measures concentrate on the following subjects:

- Theory
- Coding
- Validation data
- Validation of simulation predictions
- QA documentation

In the performance evaluation of the simulation program, several levels of application are distinguished, ranging from preliminary design to safety-related decision making. Validation comprises performance as regards seakeeping, maneuvering and capsizing. For seakeeping and maneuvering (turning circles, zig-zag maneuvers), both model test and full scale data are available, while capsize data are available through model tests. Knowledge about simulation performance levels is highly relevant, ship safety being a key issue.

7 CLOSURE

This document provides an overview of current activities as regards simulating the dynamic behavior of intact ships. The work described here is an international effort involving navies from Australia, Canada, Netherlands, UK and USA, US Coast Guard and MARIN. It is our opinion that time domain simulations can provide practical engineering information toward the safe design and operation of ships.

8 REFERENCES

De Kat, J.O., Brouwer, R., McTaggart, K., and Thomas, W.L., "Intact Ship Survivability in Extreme Waves: Criteria from a Research and Navy Perspective," *Proceedings STAB '94, Fifth International Conference on Stability of Ships and Ocean Vehicles*, Melbourne, Fla., Nov. 1994

De Kat, J.O., "Irregular Waves and Their Influence on Capsizing," *Proceedings of the 20th Symposium on Naval Hydrodynamics*, Santa Barbara, Aug. 1994
McTaggart, K.A., "Capsize Risk Prediction Including Wind Effects," To be presented at the *Third Canadian Marine Hydrodynamics and Structures Conference*, Halifax, August 1995

McTaggart, K.A., "Capsize Risk Prediction Including Wind Effects," To be presented at the *Third Canadian Marine Hydrodynamics and Structures Conference*, Halifax, August 1995

Workshop on Numerical and Physical Simulation of Ship Capsize in Heavy Seas
24-25 July 1995
University of Strathclyde, Scotland, UK

Current Views about Ship Stability

The Role and the Methods of Simulation of Ship Behaviour at Sea Including Ship Capsizing

Vincenzo ARMENIO, Giorgio CONTENTO and Alberto FRANCESCUTTO

Dept. of Naval Architecture, Ocean and Environmental Engineering - DINMA
University of Trieste, via Valerio, 10
34127 Trieste - ITALY
phone: +39-40-6763425 fax: +39-40-6763443
e-mail:francesc@univ.trieste.it

1. INTRODUCTION

Safety of life at sea and the protection of marine environment is receiving a growing interest by the different relevant parties. This is due to the too high rate of tragedy-level casualties at sea occurred during the last decade. The actions taken in this sense follow the intense phase devoted to the development of Reliability based methods of assessment of the resistance of structures and on the parallel development of techniques for Quality Management and Assurance. These last are based on the application of ISO9000 Standard Series of rules for quality control and are rapidly spreading in the world of ship construction. In line with this, IMO in his resolution A741 (18) of 4 November 1993 adopted the International Safety Management Code which constitutes a new approach to safety at sea *(Chauvel, 1994)*. Discovery, Learning, Understanding and Becoming are all practically new keywords of this new approach. Sentences as "Concern for safety is no longer focused on a product-centred response requiring technical improvements, but on human contribution and participation, in order to create a safety-conscious environment" are worth noting. In the following we shall use the word "safe" in the double sense of human life and environmental protection.

Since 1980, the SSRG, the authors belong to, has been involved in the investigation of large amplitude rolling motion. In the present paper recent results of the group in the framework of non linear formulation of the problem, including the application of CFD techniques are discussed. Finally the way the authors intend to pursue in the next future is outlined.

2. PRESENT WORK

2.1 The importance of Nonlinear Dynamics

Rolling motion has been considered since long time the motion relevant to the hydrodynamic part of the ship safety *(Francescutto, 1993a)*. Great interest has been and is presently being paid to situations leading to large amplitude rolling, namely the rolling motion directly generated by the action of wind and waves as in the beam sea condition and the rolling motion indirectly generated by the action of waves in the following sea conditions.

Rolling motion is also crucial in the context of seakeeping and the related features of seaworthiness and seakindliness. In this perspective, large amplitude is not the only undesirable feature of roll motion where large accelerations, often obtained as a side effect of a roll stabilising system, are considered equally dangerous *(Sellars et al., 1992)*.

This twofold interest in ship rolling originates from the particular characteristics that this motion features for conventional ship forms. Roll motion is indeed the motion to which the ship opposes the minimum restoring moment and contemporaneously the minimum damping ability. As a consequence, large amplitudes can be experienced also in moderate sea states, provided that the sea spectrum is sufficiently narrow and the centerpeak frequency sufficiently close to the natural frequency of the ship *(Francescutto, 1991; Francescutto, 1993b)*. Unfortunately, these events are not so rare since the natural rolling frequency of the most part of existing ships falls within the range of frequencies of the most energetic part of the sea spectrum.

Finally, it is known that the rolling motion is a strongly nonlinear phenomenon. Its intrinsic nonlinearities appear both in a qualitative and quantitative sense. Let's think for example to resonant frequency shifts and ultimately jump phenomena *(Francescutto et al., 1994a)* as a result of a nonlinear restoring and/or to saturation of the amplitude at peak as a result of a strongly nonlinear damping *(Contento et al., 1995a)*. Biased oscillations may also occur in extreme situations and coupling effects may play dramatic roles.

Talking about all these features of the roll motion, typically experienced in Towing Tank tests and at sea, we implicitly refer to a simplified decoding of the physical problem: the traditionally adopted mathematical model (ODEs) allows us to approach different aspects of the fluid-body interaction problem. However the draft of a realistic equation of motion still stands as an open problem. Approximate roll motion equations are often used in the practice. These equations use mixed hydrodynamic/hydrostatic approaches and consider that linear or quasilinear hydrodynamic assumptions allow reliable descriptions well beyond their intrinsic validity limit. Recently, the methods to study the complex dynamics of nonlinear systems have received an extraordinary development. As a consequence, it is easier to study the possibilities of strange phenomena *(Falzarano et al., 1992)* (bifurcations, chaos, symmetry breaking, etc.) hidden in the rolling motion equation than to write down a correct nonlinear equation of motion for rolling. This explains the huge amount of published papers on complex roll dynamics. The results obtained in this field are very interesting as they disclose a new world of possibilities, some of them being very dangerous. Despite that, the forecasting capability of these processes strongly depend on the reliability of the coefficients employed. In other words, even a mathematical model based on linear or quasilinear assumptions can often work well if used with 'ad hoc' parameter values, the possibility of the mentioned phenomena being often tied to very precise values for these coefficients. Moreover, bifurcations and chaos are usually studied in the deterministic case, i.e. in the presence of a regular excitation.

These coefficients may be obtained from experimental records of the motion post-processed with sophisticated parameter identification techniques. The fully theoretical calculation through analytical/numerical hydrodynamics is still to come *(Brook, 1990)* and the poor forecasting capability of the conventional seakeeping codes in the case of large amplitude motions witnesses this lack.

The growing capabilities of computers allow to face optimistically a fully hydrodynamic approach to the fluid-body interaction problem. Nevertheless simplified models based on nonlinear ODEs are still needed in the perspective of a global/realistic shiphandling simulation system *(Francescutto, 1992; Francescutto, 1993a)*. The use of these 'simple' models, tuned on specific ship/sea conditions on the basis of more complicated 'pre-runs' of simulations/experiments, are undoubtedly on the side of human safety and environment protection

With the aim to gain a deeper knowledge of the nonlinear phenomena in ship rolling, according to the 20th ITTC Seakeeping Committee, campaigns of experiments on ship models in beam sea have been carried out and are presently in progress at the Hydrodynamic Laboratories of the University of Trieste *(Cardo et al., 1994)*. A numerical procedure similar to that proposed by Spouge *(1992)* for free decay tests has been developed at DINMA for the steady state oscillations in waves at constant incident wave slope and/or at constant incident wave frequency. Sophisticated models for the damping function, restoring and effective wave slope may be used, the results (coefficients) obviously being dependent on the limits of precision of the minimisation procedure and on the confidence range of the measurements *(Contento et al., 1995a)*.

In the particular case of a model of a RoRo vessel, a campaign of measurements of the steady state roll in beam sea has been carried out *(Contento et a;., 1995a)*. The frequency (ω) and the amplitude of the incident wave have been varied in a wide range of values of practical interest. The roll response has been analysed both in the frequency domain and in the wave steepness (s_W) domain. Both families of response curves have shown distinct nonlinear features.

Those obtained at constant wave steepness exhibits a bent resonance peak. The backbone curve shows an S-shape according to the nonlinear restoring characteristics of the ship. It has been obtained through a polynomial fitting and its consistence with the locus of peaks of the experimental curves is encouraging. However the mild peak bending in the considered range of wave steepnesses leads to the impossibility of bifurcations *(Francescutto et al., 1994)* in full agreement with experiments.

The family of curves at constant wave frequency also exhibits characteristic nonlinear features. The first consists in the saturation phenomenon, i.e. the quite sharp reduction of slope of the roll amplitude curves at moderate excitation amplitudes. The second feature is the appearance of multiple intersections of the curves at different frequencies. These are connected with both the nonlinear restoring and the nonlinear damping. In particular, near resonance, the S-shape of the backbone curve is responsible for back-and-forth variations of the peak frequency.

The fitting procedure above mentioned has been applied to obtain estimates of the hydrodynamic coefficients both as a function of s_W at constant frequency and as a function of ω at constant wave steepness.

The trend shown by the coefficients as a function of the excitation frequency ω appears of great interest. While the data at the lowest tested frequency could be suspect, being obtained in a frequency range where the waves generated by the used wavemaker present non negligible higher order harmonic components, the minimum value shown at resonance by both damping and forcing coefficients is somehow an index of why discrepancies appeared between the experimental data and values obtained through the fitting procedure with constant coefficients. Starting from results like these, a further attempt has been made to improve the results of the fitting using frequency/amplitude dependent coefficients. The desired result of capturing the ends of the resonance curves has been obtained. The above analysis indicates that a very good simulation capability is obtained by an isolated rolling motion differential equation with separate contributions (added inertia, damping, restoring and excitation), provided the coefficients are derived by a technique looking at the experimental results in both the frequency and the wave steepness directions. From the preliminary results, it appears that a good quality prediction model of nonlinear rolling cannot be based on constant coefficients time domain simulations (for example from roll decay tests). These can infact lead to incorrect estimates of rolling amplitudes even when the parameters have been obtained through high level parameter estimation procedures based on experimental data. An attempt to include a frequency/wave steepness dependency of the different contributions of the uncoupled roll equation can lead to an extremely good agreement between experimental data and estimated roll response.

2.2 A fully hydrodynamic simulation of motions in waves - The development of a numerical Towing Tank

As mentioned before, the attractiveness of a simple mathematical model in ship motions is often frustrated by its own poor capability in describing properly the physical problem. The capability usually (but not necessarily) grows if detailed grids of experimental data, and consequently coefficients, are available. Full scale coefficients are in any case an 'a posteriori' option. As far as the theoretical predictions of loads/motions are concerned, they are fundamentally based on linearization/inviscid fluid assumptions apart from equivalent linearizations in the roll damping term. The nonlinear effects from wave-floating body interaction are implicitly thought to be small if compared with the linear part and are therefore neglected. Kishev et al. *(1981)* have conducted a theoretical analysis up to the second order for the roll moment in forced oscillations in calm inviscid fluid. There they show that the traditional superposition of inertia, damping and restoring moment becomes inconsistent. As a consequence, in the presence of large amplitude incident waves and/or large amplitude motions the role of the excitation and of the response characteristics of the body is hardly distinguishable in the sense of the Nonlinear Dynamics. Transient phenomena, slow drift motions, parametric oscillations, subharmonic responses, springing vibrations of the hull may therefore occur as a result of nonlinear wave-body interaction. Recently, Contento et al. *(1995a)* have conducted an extensive campaign of experimental measurements on the roll motion of a scale model of a RoRo vessel in regular beam sea and in free decay. On the other hand, conclusions similar to those of Kishev have been drawn in our analysis as shown in 2.1. Recently, the so called 'body-exact approach' in numerical wave tanks has shown its attractiveness *(Faltinsen, 1977; Vinje et al., 1981, Isaacson, 1982; Dommermuth et al., 1987; Sen et al., 1989; Cointe et al., 1990; Sen, 1993; van Daalen, 1993; Zhao, 1993; Contento, 1995a; Tanizawa, 1995; Contento et al., 1995b)*. The main idea consists in simulating, without compromises on the amplitude of the incident wave and of the motion of the body, a 'physical' towing tank experiment in the time domain. A floating body in a closed domain is therefore subjected to an incoming wave train. Appropriate boundary conditions and/or a moving boundary allow to simulate an absorbing beach and the wavemaker respectively. The fully nonlinear boundary conditions are applied both on the free surface and on the instantaneous wetted hull. Even if the inviscid fluid assumption is typically made, the method needs orders of magnitude of computational 'efforts' more than that required by linear frequency-domain solutions. In any case this matter doesn't justify the giving up of these methods when the need is stringent.

In the particular case of perfect fluid flow, after the appearance of the pioneering paper of Longuet-Higgins and Cokelet *(1976)* significant steps have been conducted in the direction of capturing second and higher order diffraction pressures on fixed structures *(Isaacson, 1982; Isaacson et al., 1991; Yeung et al., 1992; Kim et al. 1994)* or of calculating radiation/impact pressures on rigid bodies with prescribed motions in calm water *(Dommermuth et al., 1987; Zhao et al., 1993)*. Fully 3D computations seems to be a privilege of few *(Isaacson, 1982; Dommermuth et al., 1987; Tanizawa, 1995)* nevertheless several papers and applications have appeared in the 2D case *(Cointe, 1990; Contento, 1995b; Faltinsen, 1977; Sen et al., 1989; Sen, 1993; Vinje, 1981; Zhao, 1993)*. An exhaustive review of the 'numerical wave tank approach' to the wave-body interaction problem was given in The Hague (ISOPE '95) by Kim *(1995)*.

Nowadays, nonlinear wave loads predictions through the numerical wave tank approach are likely to become a standard procedure in ocean/coastal engineering. On the contrary, strong difficulties are encountered in ship motions computations. For example, the kinematic and dynamic characteristics of the incident waves generated by numerical wavemakers with a non-linear free surface, evidently affect the results, both as far as the pressures/loads are concerned and as regards the motion amplitudes in the floating body problem. Being obviously dependent on the amplitude and on the frequency of the motion of the wavemaker (boundary conditions), these characteristics have been shown to be dependent on computational features such as the size of the domain or the number of wavelenghts in the tank *(Lee et al., 1987)*, the effectiveness of the non-reflective boundary *(Yeung, 1992; Jagannathan, 1988)* or damping sponge *(Israeli, 1981; Cointe, 1990)*, the regridding *(Dommermuth, 1987)* and the interpolation-extrapolation techniques *(Saubestre, 1991; Sen, 1993; Contento, 1995b)*.

Operating at the moment in two dimensions, an accurate and robust algorithm has been developed and implemented at DINMA: the mathematical model for the wave generation and interaction with a fixed or free floating arbitrarily shaped body has been presented and deeply discussed *(Contento et al., 1995a)*. Wave generation by a flap-type wavemaker and absorption by a non reflecting boundary condition are discussed in detail evidencing some physical and numerical aspects which respectively characterise and may affect the solution. In the free floating body problem, a stable and accurate procedure for the calculation of forces and moments is proposed and systematically used with good results *(Contento, 1995a-b)*. Large amplitude motions can be simulated both prescribing the amplitude and frequency of the motion in calm water (radiation problem) and simulating free motions (decays in calm water and motions in waves).

At present, a Sommerfeld radiation condition with 'numerical' celerity is enforced to allow a long time simulation *(Contento et al., 1995b)*. Any other absorbing boundary condition can be easily implemented as well.

A flap-type wavemaker is chosen with axis of rotation at the bottom of the tank. Some significant quantities of the computed waves, such as the wave elevation, the potential and velocities, are monitored and plotted to detect the stability and accuracy of the scheme. Mass, inflow-outflow and energy-rate are systematically calculated during the simulation; moreover the Fourier analysis of the wave elevation is performed with reference to a fixed set of stations along the tank. Some results from the application of the numerical wave tank approach to the free floating body problem has been recently presented *(Contento, 1995a)*. Prescribed motions in still water or in waves, free decays and motions in waves can be simulated. According to the 20th ITTC recommendations, an extensive campaign of numerical tests for internal consistency and validation against experimental data has been conducted *(Contento, 1995b)*. The goodness of the comparison of numerical computation with experimental data from Vugts *(1967)* and linear theory *(Porter, 1960)* for the radiation and diffraction problem stands as a preliminary validation of the code. The intrinsic nature of the 'body exact approach' to ship motions avoids the use of traditional assumptions on the dynamics/hydrodynamics of the wave-body interaction problem (effect superposition, linear-quasi linear, ...). The results from the wave tank may however appear as a 'black box' so the application of a parameter identification technique to the records of the motion of the body from the simulations allows information about the nonlinear terms in the traditional ODEs. From an analysis like this, corroborated by Kishev et al. *(1981)*, it comes out that even a complicated nonlinear ODE hardly simulates correctly free decays at moderate/large amplitudes. A computation carried out in the case of a scaled cross section after the bulb of a RoRo vessel with a pronounced flare, has shown that the immersion of the flare itself introduces sensible deviations from linearity in the decay record both in heave and in roll.

Finally, it has to be pointed out that the problem of accurate predictions of large amplitude motions for complicated geometries still remains open, mainly due to large impacts resulting in water jets and breaking waves at the liquid solid interface.

2.3 The hydrodynamic coupling between liquid sloshing and ship motions

One of the main challenges for safe ship design and operation is represented by the presence on board of liquids with free surface both desired as liquid cargo or consumable and undesired as water on deck or as a result of a flooding process. The motions connected with the presence on board of liquids are important in both aspects of ship safety: structural and hydrodynamic. The impulsive pressure peaks are important for structures, while dynamic pressures are relevant to transversal inclining moments and hence capsizing.

The introduction of double hull tankers as a response to the demand of pollution safe ships in case of grounding or collision, increases the importance of dynamic pressures since they are now relevant to structural safety too.

The importance attributed since long time to the possible danger represented by liquids with free surface on board is witnessed by the presence on intact ship stability rules of a specific regulation regarding first a correction to the initial metacentric height and successively the full curves of static and dynamic stability. In spite of the use of the term "dynamic", this is a static approach and is fully valid in this limit, i.e. in the case of infinitely slow or quasi-static inclinations of the ship.

Nowadays, it is completely understood that the hydrodynamic aspects of ship safety (and also part of the structural and operational ones) can only be treated as really dynamic phenomena, hence the consideration of sloshing motions and loads *(Francescutto, 1992; Francescutto, 1993a)*. This is particularly true when one realises that the loss of a ship at sea is generally a complex phenomenon involving the simultaneous action of different causes which superpose non linearly, to feedback phenomena and to the possible loss of structural integrity. Transient and large amplitude motions play the most relevant role. This is the reason why static, linear or quasi linear approaches cannot be used with sufficient reliability when capsizing is involved. They fail in the description of the phenomena, and therefore don't possess sufficient forecasting capability, in quantitative aspects, qualitative ones and often in both.

Previous statements are by no means questioned by some experimental observation *(Grochowalski, 1989)* reporting the correlation between actual behaviour at sea and complying to intact stability rules is quite good. In first place it is known that intact stability rules don't represent absolute safety. Furthermore, ships are an object with quite a high reliability, i.e. it is sufficient a small specimen of bad correlation between static stability and ship loss to explain the observed casualties at sea *(Francescutto, 1992)*.

From a mathematical point of view, the analysis of the roll motion of a ship with free surface liquids shipped on board constitutes a difficult task, due to the strong interaction between ship motions and liquid sloshing inside the tank. The problem as a whole can be split into two different subproblems. The first concerns the appropriate simulation of large amplitude motions of the ship, including the coupling between roll, sway and heave motion; the latter is related to the appropriate modelling of large amplitude liquid sloshing inside the partially filled tank.

In the past, several mathematical models have been proposed for the solution of such a problem. As a general rule, linear ship motion computer codes have been matched to algorithms which solve the partial differential equations modelling liquid sloshing. When the liquid depth inside the tank is small enough, the shallow water equations have been considered

(Pantazopoulos, 1990; Armenio, 1992), whereas in the other cases the sloshing problem has been solved by the use of BEM techniques in the hypothesis of inviscid liquids *(Francescutto et al., 1994b)*.

The shallow water equations, hyperbolic in nature, are usually solved by means of 'shock capturing' techniques developed in the framework of gasdynamic. The 'Random Choice' method, *(Chorin, 1976)*, has been widely used in the past. The method allows to treat sharp discontinuities effectively, nevertheless it is not conservative both in mass end energy. The main effects of such gaps consist in a wrong prediction of the wave speed inside the tank and in the numerical variation of water volume during computations. For the above reasons a new powerful technique (CE-SE) *(Chang et al., 1992)* recently developed has been successfully applied *(La Rocca, 1994)* for the solutions of the Shallow Water equations.

In the meantime, in order to simulate accurately large amplitude liquid sloshing in arbitrary shaped tanks (for instance equipped with internal baffle), an improved MAC method (SIMAC) *(Armenio, 1994)* has been developed. The algorithm is able to simulate large amplitude free surface flows, and, at the same time, to solve viscous stresses accurately. This last circumstance is due to its own ability in treating effectively very stretched grids. In order to validate the previous mathematical models experimental tests have been carried out considering a 0.50 meter breadth rectangular tank in roll motion *(Armenio et al., 1995a)*.

Summarising, it has been proved that the use of the shallow water equations allows accurate evaluations of the wave pattern of the wave speed and of the pressure distribution over the rigid walls for filling ratios (λ/b) up to 0.10-0.12. This circumstance is basically due to the nature of the shallow water approximation and it is independent on the algorithm used for the solution of them. A detailed discussion on this topic is in *(Armenio et al., 1995b)*.

The RANSe provide accurate evaluation both of the pressure distribution and of the wave patterns in the whole range of filling ratios investigated. Nevertheless, break down of computations have to be expected when the phenomenon becomes rather violent including large splashes, breaking waves and air inclusion.

Then, the coupled problem, concerning the interaction between the ship motion and liquid sloshing has been dealt with by matching the computer code solving the RANSe by means of SIMAC and the SWe by CE-SE with the uncoupled nonlinear roll motion equation. In order to validate the above mathematical model experimental tests considering the scale model of a fishing vessel equipped with a rectangular tank, in a regular beam sea, have been carried out at the Hydrodynamic Laboratories of DINMA, University of Trieste.

This study has provided very interesting physical and numerical considerations *(Armenio et al., 1995c; Armenio et al., 1995d)* briefly summarised in the following.

The system is similar to a slightly damped two degrees of freedom oscillating system. The most significant effect due to the presence of free surface liquids on board results in the appearance

of two different resonance frequencies at which ship rolling or liquid sloshing can experience large amplitude motion.
The values of the new resonance frequencies are strictly related to the characteristics and the amount of the liquid shipped on board, the tank geometry and its own position inside the ship.
The presence of small quantities of liquid (shallow water case) provides the reduction of the maximum roll angle as compared with the case without liquids on board. Moreover, in shallow water cases, in the zone of the anti-symmetric resonance the ship roll motion can experience the maximum roll amplitude in the frequency domain, whereas the opposite is true by increasing the liquid depth inside the tank.
As regards the numerical model developed, a general good agreement with experimental data has been observed in the whole range of encounter frequencies of practical interest.
When SWe are used, large disagreement between numerical and experimental results are evidenced in the first resonance zone for filling ratios greater than 0.10. As previously pointed out this circumstance is due to the shallow water approximation, that reduce the pressure field to an hydrostatic one.
Despite the analysis described in 2.1, the use of constant hydrodynamic coefficients as derived by the roll decay experiments has constituted a fairly good approximation. This is mainly due to the fact that the sloshing induced moment is more than one order of magnitude larger than the roll damping moment in the whole range of wave frequencies investigated.
According to *(Rakhmanin, 1995),* in the next future a more complete mathematical model solving the coupled large amplitude roll heave and sway motion of the ship will be considered. This improvement will overcome the restrictive hypotheses that considers fixed the roll axis during the ship motion

3. FUTURE WORK

3.1 Short-term research: fully hydrodynamic approach to ship motions with free surface liquids on board

In the previous paragraphs, an analysis has been conducted on the reasons leading to the choice of CFD approaches to ship motions forecasting and to on board free surface liquids motions/loads computations. In principle, these procedures can (and at present they do) work separately. The motion of the ship is indeed governed by Newton law where the force/moment vectors summarise the whole *external fluid actions* without any distinction between excitation and own response. Nothing forbids us to add forces and moments, if any, from the interaction of the body with other systems like moorings, wind or sloshing. From the other side, the simulation of the liquid motion in a tank needs the instantaneous knowledge of the displacements, velocities and accelerations of the wall of the container, i.e. of the ship (body forces in a fixed to the tank frame of reference). So at a specific time instant the motion of the body induced by the external hydrodynamics acts as an excitation for the liquid sloshing which in turn acts as a feedback on the motion of the ship. By this fully coupled ship/sloshing systems, the simulation of the ship in waves with free surface liquids is free from any assumption on the position of the roll axis *(Hutchinson, 1991),* i.e. important effects from sway and heave (plus surge, pitch and yaw in 3D) are no longer neglected. Moreover the inflow-outflow inside the flooded tank can be easily simulated, providing an accurate tool for the evaluation of the transient flooding after damage.

3.2 Long-term research - A "must" for future: domain decomposition

The scenario given in 2.1 to 2.3 about non linear dynamics and CFD tools for 'state of art' ship motions computations with the optional presence of free surface liquids on board, does not exhaust the problem at all. In fact, viscous effects often can affect wave loads on the hull and in a inviscid towing tank approach they can be accounted only in a 'concentrated' form (vortex shedding techniques) *(Standing et al., 1992)* and only in special cases (sharp sections where the influence of the Reynolds number on the separation point is scarce). In the field of ship resistance, the problem is rather similar in the sense that wave and viscous effects are strongly coupled. The effectiveness of an inviscid or viscous numerical approach in terms of resolution, computing time, ... has been widely discussed in the dedicated literature. Both methods are efficient in the range of validity of their formulation/application. Hence the domain decomposition has been first introduced as the 'magic potion' to the solution of the ship resistance problem: 'viscous' methods are applied on the part of the fluid domain where viscous effects are expected, namely near the body, whereas 'inviscid' methods are applied in the outer zone where the vorticity is expected to be vanished.
At present this kind of simulations of unsteady oscillating flows around a surface piercing body are not available in literature. It is our intention to pursue this fascinating approach.

4. A BRIEF REMARK

Computing time - Spending less rarely means saving money

High reliability and precision of codes for numerical simulation is unfortunately paid in terms of computing time. Each numerical simulation requires a few hours in a medium speed workstation. While this result could be optimised, we don't consider this order of magnitude of computer time a too great limitation to the practical application of the developed methods. Apart the specific results, it allows to obtain in terms of flexibility, possibility of parameter variation, absence of scale effects, possibility of tank geometry variation, and so on. In addition we know that computations of such complexity and even more, are considered very attractive in the analysis of the circumstances of a casualty *after* its occurrence, i.e. when the problem to be solved is to find which of the parties involved in the processes of ship design, construction, maintenance, certification, operation *has to pay* for the ship loss (and its consequences as regards human life and environment). Our hope, corroborated by IMO actions as the ISM Code above mentioned, is that these computations and other relevant to safety, become more and more attractive *before* the occurrence of casualties, i.e. passing from an *understanding and learning* to a *prevention* scheme.

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Workshop: Numerical And Physical Simulation
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Application of Nonlinear Dynamical System Approach to Ship Capsize in Regular Following and Quartering Seas

by **Naoya Umeda**
National Research Institute of Fisheries Engineering, Japan
Currently, a Visiting Research Fellow, University of Strathclyde, United Kingdom

ABSTRACT

This paper describes a method for applying a nonlinear dynamical system approach to capsize and broaching of a ship running in regular following and quartering seas. First, we should find coexisting steady states of both autonomous and nonautonomous systems representing a coupled surge-sway-roll-yaw-rudder motion. Then we can identify the final steady state where the ship is captured as a result of sudden change of operational factors.

INTRODUCTION

A nonlinear system can possess several coexisting attractors. Thus, it is very difficult to clarify a global feature of the system by only repeating time domain simulations with a limited number of initial conditions. In particular, safety problems in engineering require us to exclude all potential danger in advance. For this purpose, the nonlinear dynamical approach is the most suitable. It investigates recurrent behaviours of trajectories in a phase space. For dissipative systems, this approach produced many and useful fruits because trajectories are attracted to subsets of the phase space. It is found that the forced Duffing equation, one of the simplest models that this approach was applied, has several steady states including even chaos attractors. Since the phase space of this system is three dimensional, several geometric methods are very effective. (Thompson & Stewart, 1986)

In the seakeeping theory, a motion of a ship drifting in beam seas can be described with a coupled equation for sway and roll. If a wave length is much longer than ship breadth, this equation can be simplified to a one degree of freedom equation in roll. Because, the roll radiation moment due to sway cancels out the roll diffraction moment. The nonlinear restoring moment can be represented with a third order polynomial or its equivalent. As a result, this roll equation is regarded as a kind of the forced Duffing equation. Thus the nonlinear dynamical system approach has been directly applied to the roll motion and capsizing of a ship in beam seas. (e.g. Kuo & Odabasi, 1975; Thompson, 1989; Kan & Taguchi, 1990)

On the other hand, the IMO stability criteria, especially weather criteria, almost succeed to prevent capsizing of a ship drifting in beam seas, except for capsizing due to a large-scale breaking wave. It is also true, however, that the ship satisfying the criteria may easily capsize when she runs in following and quartering seas. (e.g. Umeda, Hamamoto et al., 1995) The Doppler effect can make the encounter frequency so low as the roll natural frequency or the time constant of manoeuvring motion. Thus, some instabilities may occur as coupled motions in surge, sway, roll and yaw. For these phenomena, the above-mentioned geometric methods used for beam sea cases meet difficulties because increase of degrees of freedom prevents visualisation of the phase space. In the meanwhile, the ship master can avoid dangerous phenomena in following and quartering seas by changing ship speed or heading angle with propulsive power and a rudder. Thus, the IMO is developing an operational guidance for the master to avoid dangerous situation.

Responding this practical requirement and expectation for the nonlinear dynamical system approach, the author proposes a method for applying the dynamical system approach to capsizing of a ship running in following and quartering seas. Here, as the first step, only ship motions in regular waves are discussed. Obviously effects of wave irregularity should be discussed near future.

NOMENCLATURE

A_{11}	surge added mass coefficient	B_{11}	surge damping coefficient
c	wave celerity	F_n	nominal Froude number
F_1	amplitude of wave-induced surge force	g	gravitational acceleration
H	wave height	I_{xx}	moment of inertia in roll
I_{zz}	moment of inertia in yaw	J_{xx}	added moment of inertia in roll
J_{zz}	added moment of inertia in yaw	k	wave number
K_P	rudder gain	K_w	wave-induced roll moment
L	ship length	m	ship mass
m_x	added mass in surge	m_y	added mass in sway
n	propeller revolution number	N_w	wave-induced yaw moment
p	roll rate	r	yaw rate
R	ship resistance	t	time
T	propeller thrust		
T_D'	nondimensional time constant for differential control	T_E'	nondimensional time constant for steering gear
u	surge velocity	U_0	ship cruising velocity
v	sway velocity	X_w	wave-induced surge force
Y_w	wave-induced sway force	z_H	height of centre of lateral force
δ	rudder angle		
ε_F	phase lag of wave-induced surge force	ϕ	roll angle
		λ	wave length
ξ_G	longitudinal position of C. G.	χ	heading angle
χ_c	desired heading angle for auto pilot	ω_e	averaged encounter frequency

MATHEMATICAL MODELS

When a ship runs in following and quartering seas, the encounter frequency of the ship to waves becomes much smaller than the natural frequencies in heave and pitch. Therefore, heave and pitch motions can be approximated to just trace their static equilibria. In addition, because of a low encounter frequency, hydrodynamic forces due to shedding free vortices are dominant and wave-making effects are almost negligible. Thus, a manoeuvring model is more suitable than a seakeeping model.

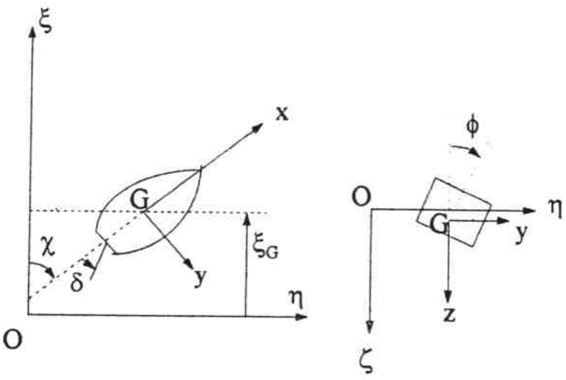

Fig. 1 Co-ordinate systems

As can be seen in Fig. 1, two co-ordinate systems are used: wave fixed with origin at a wave trough, ξ axis in the direction of wave travel; and upright body fixed with origin at the centre of ship gravity, the x axis pointing towards the bow, the y axis to starboard and the z axis downwards. The latter co-ordinate system is not allowed to turn round the x axis. (Hamamoto, et.al., 1994) The symbols are defined in the nomenclature.

The state vector **x** of this system are defined as follows:

$$\mathbf{x} = \{\xi_G/\lambda, u, v, \phi, p, \chi, r, \delta\}^T \tag{1}$$

The dynamical system can be represented by the following state equation:

$$\dot{\mathbf{x}} = \mathbf{F}(\mathbf{x}) = \{f_1(\mathbf{x}), f_2(\mathbf{x}), \cdots, f_8(\mathbf{x})\}^T \tag{2}$$

where

$$f_1(\mathbf{x}) = \{u\cos\chi - v\sin\chi - c\}/\lambda \tag{3}$$

$$f_2(\mathbf{x}) = \{T(u;n) - R(u) + X_w(\xi_G/\lambda, \chi)\}/(m+m_x) \tag{4}$$

$$\begin{aligned} f_3(\mathbf{x}) = \{&-(m+m_x)ur + Y_v(u;n)v + Y_r(u;n)r + Y_p(u)p \\ &+Y_\phi(u)\phi + Y_\delta(\xi_G/\lambda, u, \chi; n)\delta + Y_w(\xi_G/\lambda, u, \chi; n)\} \\ &/(\mathrm{m}+\mathrm{m_y}) \end{aligned} \tag{5}$$

$$f_4(\mathbf{x}) = p \tag{6}$$

$$\begin{aligned} f_5(\mathbf{x}) = \{&m_x z_H ur + K_v(u;n)v + K_r(u;n)r + K_p(u)p \\ &+K_\phi(u)\phi + K_\delta(\xi_G/\lambda, u, \chi; n)\delta + K_w(\xi_G/\lambda, u, \chi; n) \\ &-\mathrm{mgGZ}(\xi_G/\lambda, \phi, \chi)\}/(I_{xx}+J_{xx}) \end{aligned} \tag{7}$$

$$f_6(\mathbf{x}) = r \tag{8}$$

$$\begin{aligned} f_7(\mathbf{x}) = \{&N_v(u;n)v + N_r(u;n)r + N_p(u)p + N_\phi(u)\phi \\ &+N_\delta(\xi_G/\lambda, u, \chi; n)\delta + N_w(\xi_G/\lambda, u, \chi; n)\} \\ &/(\mathrm{I_{zz}}+\mathrm{I_{zz}}) \end{aligned} \tag{9}$$

$$f_8(\mathbf{x}) = \left[(U_0/L)\{-\delta - K_P(\chi - \chi_c)\} - K_P T'_D r\right]/T'_E \tag{10}$$

Since external forces are functions of the surge displacement but not time, this equation is nonlinear and autonomous.

The wave forces and moments are predicted as the sum of the Froude-Krylov force and hydrodynamic lift due to wave particle velocity by a slender body theory. Umeda et al. (1995) well validated this prediction method with a series of captive model experiments covering the typical broaching conditions, namely, the runs with zero encounter frequency in extremely steep quartering seas. As widely accepted, the wave effect on the righting moment can be estimated by integrating wave pressure up to a wave surface. The manoeuvring derivatives with respect of sway, roll and yaw can be obtained by conventional captive model tests in still water and rudder derivatives are calculated using the inflow velocity modified to take into account the orbital velocity due to the wave, together with the change in the propeller race. In addition, coupling inertia terms are neglected because of the low encounter frequency. The wave effect on the manoeuvring derivatives is taken into account in some previous studies. The model experiment (Fujino et al., 1983) shows that it is generally small as far as a ship is free in heave and pitch. The nonlinear terms of the manoeuvring derivatives are also assumed to be negligible because the sway velocity and yaw rate are much smaller than the forward velocity even during broaching, as observed by Fuwa et al. (1982).

An inertia co-ordinate system travelling with a mean ship velocity, $\overline{U}$, and mean ship course, $\overline{\chi}$, enables us to transform Eq. (2) based on the wave fixed co-ordinate system to a nonlinear and nonautonomous model. In this model the ship motions are represented by surge, $\widetilde{X}_G$, sway, $\widetilde{Y}_G$, roll, $\widetilde{\phi}$, yaw, $\widetilde{\chi}$ and rudder, $\widetilde{\delta}$ around the inertia co-ordinate system travelling with a mean velocity and course. Here we do not assume that the surge and sway motions are small. Because, no restoring forces exist for these motions. When we consider the ship motions whose frequency is equal to the encounter frequency, the following van del Pol transformation is useful.

$$(u_1 \quad v_1)^T = P\left(\widetilde{X}_G \quad \dot{\widetilde{X}}_G\right)^T \tag{11}$$
$$(u_2 \quad v_2)^T = P\left(\widetilde{Y}_G \quad \dot{\widetilde{Y}}_G\right)^T \tag{12}$$
$$(u_4 \quad v_4)^T = P\left(\widetilde{\phi} \quad \dot{\widetilde{\phi}}\right)^T \tag{13}$$
$$(u_6 \quad v_6)^T = P\left(\widetilde{\chi} \quad \dot{\widetilde{\chi}}\right)^T \tag{14}$$
$$(u_7 \quad v_7)^T = P\left(\widetilde{\delta} \quad \dot{\widetilde{\delta}}\right)^T \tag{15}$$

where
$$P = \begin{bmatrix} \cos\omega_e t & -1/\omega_e \sin\omega_e t \\ -\sin\omega_e t & -1/\omega_e \cos\omega_e t \end{bmatrix} \tag{16}$$

Substituting Eqs.(11)-(15) to Eq. (2) and averaging them over one period, that is,

$$0 < t < 2\pi/\omega_e \tag{17}$$

we obtained the following averaged equation.

$$\dot{\mathbf{v}} = \mathbf{G}(\mathbf{v}) = \{g_1(\mathbf{x}), g_2(\mathbf{x}), \cdots, g_{10}(\mathbf{x})\}^T \tag{18}$$

where

$$\mathbf{v} = (u_1, v_1, u_2, v_2, u_4, v_4, u_6, v_6, u_7, v_7)^T \tag{19}$$

$$\begin{aligned} g_1(\mathbf{x}) &= \frac{1}{2}\omega_e v_1 - \frac{F_1}{2\omega_e A_{11}}\cos\varepsilon_F - \frac{B_{11}}{2A_{11}}u_1 \\ &+ \frac{\mathrm{k}^2 F_1}{16\omega_e A_{11}}\Big\{3\sin\varepsilon_F\left(v_1^2\cos^2\bar{\chi} - 2v_1v_2\cos\bar{\chi}\sin\bar{\chi} + v_2^2\sin^2\bar{\chi}\right) \\ &+ \cos\varepsilon_F\left(u_1^2\cos^2\bar{\chi} - 2u_1u_2\cos\bar{\chi}\sin\bar{\chi} + u_2^2\sin^2\bar{\chi}\right) \\ &-2\sin\varepsilon_F\big(-u_1v_1\cos^2\bar{\chi} + u_1v_2\cos\bar{\chi}\sin\bar{\chi} \\ &\qquad + u_2v_1\cos\bar{\chi}\sin\bar{\chi} - u_2v_2\sin^2\bar{\chi}\big)\Big\} \end{aligned} \tag{20}$$

$$\begin{aligned} g_2(\mathbf{x}) &= -\frac{1}{2}\omega_e u_1 + \frac{F_1}{2\omega_e A_{11}}\sin s\varepsilon_F - \frac{B_{11}}{2A_{11}}v_1 \\ &- \frac{\mathrm{k}^2 F_1}{16\omega_e A_{11}}\Big\{3\sin\varepsilon_F\left(u_1^2\cos^2\bar{\chi} - 2u_1u_2\cos\bar{\chi}\sin\bar{\chi} + u_2^2\sin^2\bar{\chi}\right) \\ &+ \sin\varepsilon_F\left(v_1^2\cos^2\bar{\chi} - 2v_1v_2\cos\bar{\chi}\sin\bar{\chi} + v_2^2\sin^2\bar{\chi}\right) \\ &-2\cos\varepsilon_F\big(-u_1v_1\cos^2\bar{\chi} + u_1v_2\cos\bar{\chi}\sin\bar{\chi} \\ &\qquad + u_2v_1\cos\bar{\chi}\sin\bar{\chi} - u_2v_2\sin^2\bar{\chi}\big)\Big\} \end{aligned} \tag{21}$$

and so on. Similarly we can find different averaged equations for several subharmonic motions.

STEADY STATE

In order to obtain steady states of the ship motions, the fixed points, $\mathbf{x}_0$ and $\mathbf{v}_0$, should be calculated by solving the following equations:

$$\mathbf{F}(\mathbf{x}_0) = \mathbf{0} \tag{22}$$

$$\mathbf{G}(\mathbf{v}_0) = \mathbf{0} \tag{23}$$

Then we examine its local stability by calculating eigenvalues of the Jacobian matrix for locally linearlized equations at the fixed points.

Since $\mathbf{x}_0$ is a fixed point of the autonomous system on the wave fixed co-ordinate, it means that the ship will run with the wave celerity, a certain drift angle, a heading angle and a rudder angle. This is often called as surf-riding. If the auto pilot is appropriate , this fixed point can be stable at a wave down slope near a wave trough. This stable surf-riding has been also realised in model experiments. (e.g. Kan, 1987) If the auto pilot is not appropriate or heading angle is larger, this stable fixed point easily tends to be unstable. This unstable fixed point is usually saddle. (Umeda, 1994)

$\mathbf{v}_0$ is a fixed point of the averaged equation. The averaging theorem (e.g., Guckenheimer & Holmes, 1983) indicates that, if an averaged equation has a hyperbolic fixed point, $\mathbf{v}_0$, the original equation possesses a unique hyperbolic periodic orbit of the same stability type as $\mathbf{v}_0$. Therefore, $\mathbf{v}_0$ means a periodic motion whose frequency is equal to the encounter frequency and its local stability can be examined with eigenvalues as the case of $\mathbf{x}_0$. Similarly we can investigate subharmonic motions.

TRANSIENT STATE

If the control parameter in the mathematical model changes slowly, the ship motion simply follows the above-obtained steady states. In some cases a fixed point may emerge or disappear. Its local stability may change from stable to unstable or from unstable to stable. These are bifurcations. As slowly-changing control parameters, the ship displacement, trim, mass distribution, wave height, wave length and so on can be pointed out. For these parameters, the analysis of steady states is enough for practical purposes.

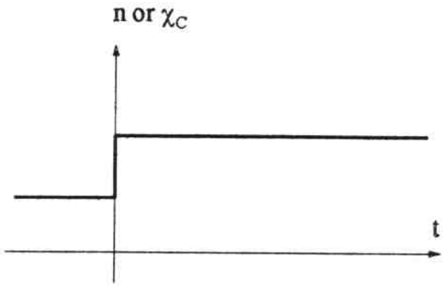

Fig. 2 Change of control parameter

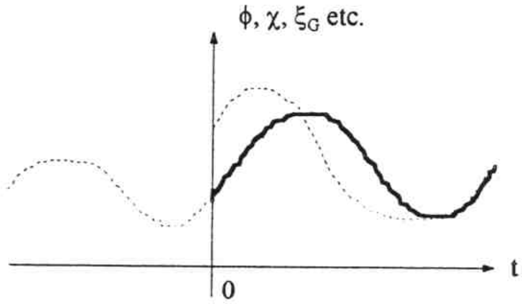

Fig. 3 Steady behaviours (·····) and transient behaviour (—)

On the other hand, the propeller revolution number, n, and the desired course for auto pilot, χ_c, cannot be assumed to be slowly changed. These changes are rather sudden as Fig. 2 compared with the time constant of ship motion. Before this operational action for the propeller and rudder, the ship motion without doubt exists in a stable steady state for the former control parameter. When time tends to infinity after the operational action, the ship must settle down to one of stable steady states for the new control parameter. However, it is not obvious which steady state realises among several coexisting steady states. Therefore, we start to numerically integrate with time the state equation involving the new control parameters from the preceding steady state, as shown in Fig. 3. If the preceding state is periodic, the phase lag of the ship motion to a wave is not unique although each phase lag among ship motions is unique. Thus, we have to repeat numerical integration from 0 to 2π as the phase lag of the ship motion to a wave. However, since the initial value set of the phase space is limited to be one dimensional, the procedure is still applicable for practical purpose.

Although this procedure is quite straightforward, we can further reduce our effort for numerical simulations by a method in some cases. That is the analysis of invariant manifold. The Hartman-Grobman theorem and stable manifold theorem (e.g., Guckenheimer & Holmes, 1983) indicate that the invariant manifolds representing all trajectories associated with a fixed point can be obtained by tracing trajectories backwards and forwards in time from eigenspace spanned by eigenvector at the hyperbolic fixed point.

In the case of the uncoupled autonomous system in pure following seas, stable invariant manifolds for an unstable fixed point identify a domain of attraction for surf-

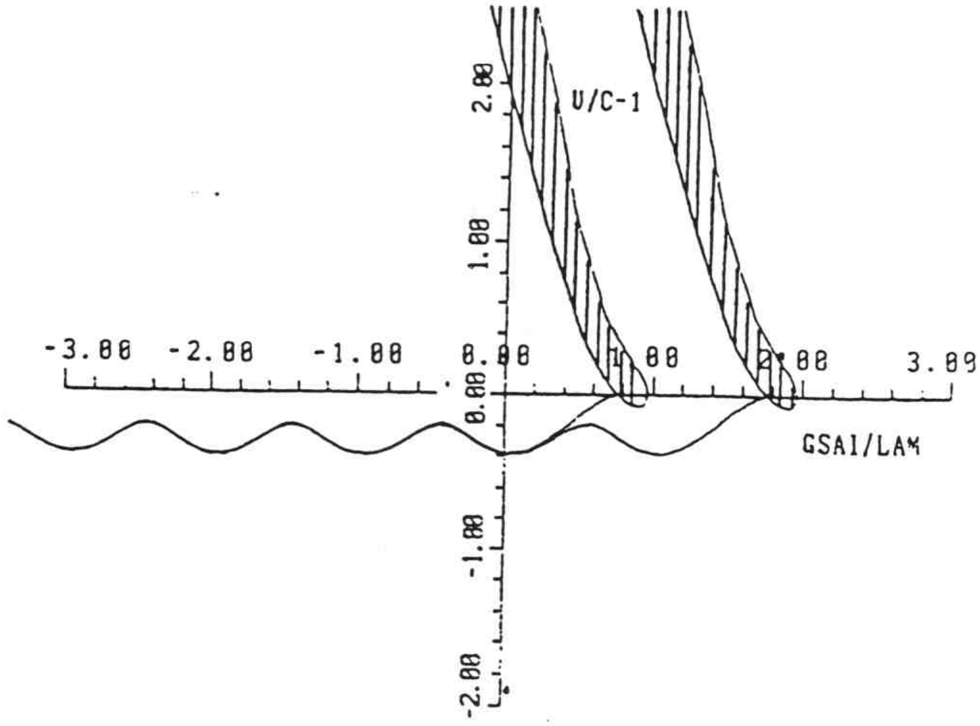

Fig. 4 Invariant manifolds in a phase plane of a ship longitudinal motion in following seas (H/λ=1/10, λ/L=1.0, Fn=0.2866) (Umeda, 1990)

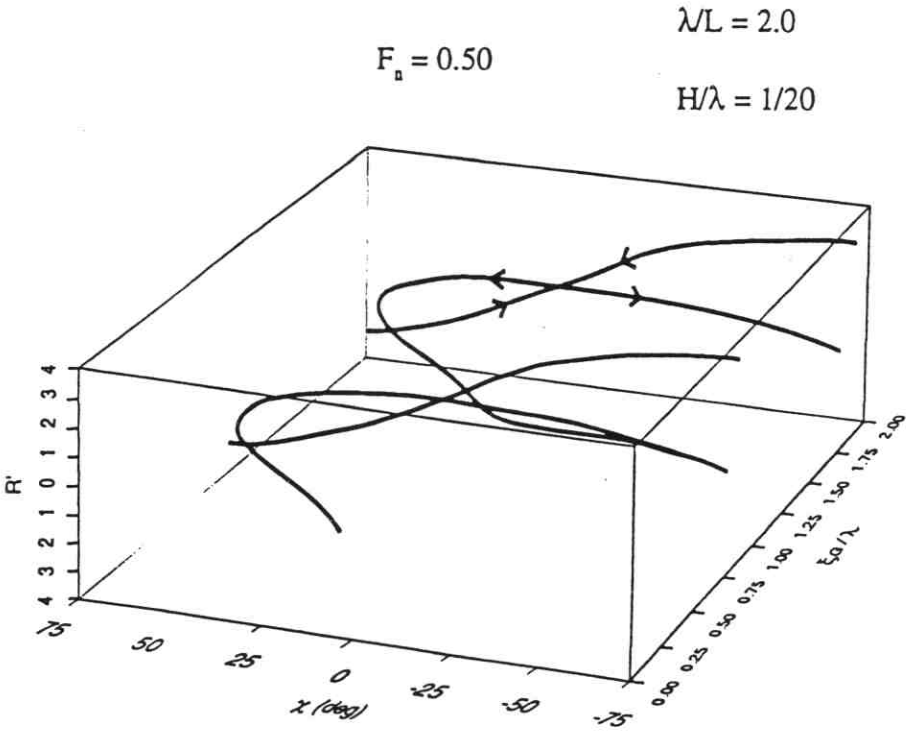

Fig. 5 Invariant manifolds in a projected phase space for a ship planner motion in quartering seas (Umeda & Renilson, 1994)

riding in the phase space, as shown in Fig. 4. Thus, we can estimate the final steady state only by judging whether the preceding steady state is within this domain or not. (Umeda, 1990)

Some practically important phenomena are transient. An example of them is broaching. This means that a helmsman cannot prevent turning of the ship despite his maximum steering effort. It is well known that this phenomenon may easily occur when a ship is nearly surf-ridded in steep quartering waves. Usually the maximum steering effort has a nature of the bang-bang control. Thus, the rudder angle fixed with its maximum opposite angle, δ=-35 degrees, the surge-sway-yaw motion was examined as an autonomous dynamical system. (Umeda & Renilson, 1992 & 1994) This study shows that the fixed points of this system are saddle and one of its unstable invariant manifold may have positive yaw rate for a while, as shown in Fig. 5. The positive yaw rate here is the increase of turning under the maximum opposite rudder angle. This means that broaching from a nearly surf-riding condition was explained with a dynamical system approach.

From practical application, the most urgent problem is which operational actions induce ship capsizing from normal periodic motions. The investigation for this problem is now under way by the author.

CONCLUSIONS

For assessing ship stability in following and quartering seas, it is essential to obtain steady states of both autonomous and nonautonomous systems. Then we can predict the final steady state as a result of slowly change in ship loading condition or environmental condition. Moreover, we can identify the final steady state as a result of sudden change of operational parameters by a limited number of time domain simulations from the preceding steady state.

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