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## Nonlinear Time-Domain Simulation of Ship Capsizing in Beam Seas

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# NOTICE



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16. Abstract (Maximum 200 words)

Ship design practice has been to measure stability by static criteria and to compensate for dynamic effects through a margin of safety. However, there is a fundamental difference between static and dynamic stability. Certain factors which result in favorable static stability characteristics may actually present greater danger when considered in light of a dynamic analysis. The traditional linear strip-theory method is not suitable for assessing ship capsizing. This report presents a state-of-the-art nonlinear simulation method, LAMP (Large-Amplitude Motion Program), for the evaluation of ship operation in extreme seas. The intent of the study was to model maritime casualties, including a time-domain simulation of a ship capsizing in beam seas. A 400-foot (122-meter) Series 60, $C_B = 0.7$ ship with the center of gravity, CG, located amidships and 2.07 ft (0.63 m) below the design waterline is used as an example in this study. This ship satisfies the U.S. Coast Guard's minimum Metacentric Height, or GM, requirement for large cargo ship. It is shown in the report that capsizing can happen due to dynamic effects even for ships that satisfy the minimum GM requirement.

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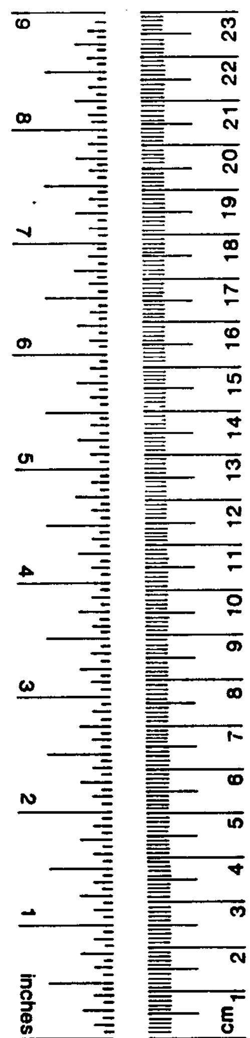
# METRIC CONVERSION FACTORS

## Approximate Conversions to Metric Measures

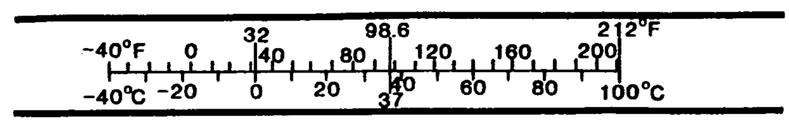
| Symbol | When You Know | Multiply By | To Find | Symbol |
|---|---|---|---|---|
| | | **LENGTH** | | |
| in | inches | *2.5 | centimeters | cm |
| ft | feet | 30 | centimeters | cm |
| yd | yards | 0.9 | meters | m |
| mi | miles | 1.6 | kilometers | km |
| | | **AREA** | | |
| $in^2$ | square inches | 6.5 | square centimeters | $cm^2$ |
| $ft^2$ | square feet | 0.09 | square meters | $m^2$ |
| $yd^2$ | square yards | 0.8 | square meters | $m^2$ |
| $mi^2$ | square miles | 2.6 | square kilometers | $km^2$ |
| | acres | 0.4 | hectares | ha |
| | | **MASS (WEIGHT)** | | |
| oz | ounces | 28 | grams | g |
| lb | pounds | 0.45 | kilograms | kg |
| | short tons (2000 lb) | 0.9 | tonnes | t |
| | | **VOLUME** | | |
| tsp | teaspoons | 5 | milliliters | ml |
| tbsp | tablespoons | 15 | milliliters | ml |
| fl oz | fluid ounces | 30 | milliliters | ml |
| c | cups | 0.24 | liters | l |
| pt | pints | 0.47 | liters | l |
| qt | quarts | 0.95 | liters | l |
| gal | gallons | 3.8 | liters | l |
| $ft^3$ | cubic feet | 0.03 | cubic meters | $m^3$ |
| $yd^3$ | cubic yards | 0.76 | cubic meters | $m^3$ |
| | | **TEMPERATURE (EXACT)** | | |
| °F | Fahrenheit temperature | 5/9 (after subtracting 32) | Celsius temperature | °C |

*1 in = 2.54 (exactly).




## Approximate Conversions from Metric Measures

| Symbol | When You Know | Multiply By | To Find | Symbol |
|---|---|---|---|---|
| | | **LENGTH** | | |
| mm | millimeters | 0.04 | inches | in |
| cm | centimeters | 0.4 | inches | in |
| m | meters | 3.3 | feet | ft |
| m | meters | 1.1 | yards | yd |
| km | kilometers | 0.6 | miles | mi |
| | | **AREA** | | |
| $cm^2$ | square centimeters | 0.16 | square inches | $in^2$ |
| $m^2$ | square meters | 1.2 | square yards | $yd^2$ |
| $km^2$ | square kilometers | 0.4 | square miles | $mi^2$ |
| ha | hectares (10,000 $m^2$) | 2.5 | acres | |
| | | **MASS (WEIGHT)** | | |
| g | grams | 0.035 | ounces | oz |
| kg | kilograms | 2.2 | pounds | lb |
| t | tonnes (1000 kg) | 1.1 | short tons | |
| | | **VOLUME** | | |
| ml | milliliters | 0.03 | fluid ounces | fl oz |
| l | liters | 0.125 | cups | c |
| l | liters | 2.1 | pints | pt |
| l | liters | 1.06 | quarts | qt |
| l | liters | 0.26 | gallons | gal |
| $m^3$ | cubic meters | 35 | cubic feet | $ft^3$ |
| $m^3$ | cubic meters | 1.3 | cubic yards | $yd^3$ |
| | | **TEMPERATURE (EXACT)** | | |
| °C | Celsius temperature | 9/5 (then add 32) | Fahrenheit temperature | °F |



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## Executive Summary

Ship design practice has been to measure stability by static criteria and to compensate for dynamic effects through a margin of safety. However, there is a fundamental difference between static and dynamic stability. The Fastnet Yacht Race disaster in 1979 revealed that "... certain factors which result in favorable static stability characteristics may actually present greater danger when considered in light of a dynamic analysis." (Stephens, Kirkman and Peterson, 1981)

The existing linear strip-theory method cannot be used for assessing capsizing. Advanced nonlinear simulation methods are required. As we shall see, such advanced methods are now under development and their application to the assessment of the vessel dynamic stability problem is a realistic practical goal today.

The main objective of the present project has been to investigate the capabilities of a 3-D nonlinear time-domain Large-Amplitude Motion Program (LAMP) for the evaluation of ships operating in extreme waves. The intent was to build upon previous LAMP development and extend it to the modeling of maritime casualties, including the time-domain simulation of a ship capsizing in beam seas. This modeling capability will allow both the analysis of recorded casualties and the identification of potential safety concerns.

Ship motions in beam seas are extremely complicated since the roll motion is highly nonlinear and the viscous effects may be important. A typical example of beam sea capsizing is illustrated in a time sequence in Figure 1. The simulations are for a 400-foot (122-meter) Series 60, $C_B$=0.7 ship with center of gravity, CG, located at the mid ship and 2.07 ft (0.63 m) below the design waterline. This ship satisfies the U.S. Coast Guard's minimum GM requirement for large cargo ship. A linear large-amplitude regular beam wave (wave height, h=32 ft or 9.67 m, and wave length, l = 402 ft or 122.56 m) approaches from starboard (from right to left in the pictures). The ship is rolling in the counterclockwise direction while the wave crest is approaching. The ship capsizes near the crest of the wave.

In the original LAMP formulation (Lin & Yue, 1990; Lin, et al., 1992), it is assumed that the ship motion may be large relative to the wave amplitude. In the current version, the LAMP formulation has been extended to allow the presence of large-amplitude incident waves (Lin, et al., 1994). The incident wave amplitude can be of the same order of magnitude or larger than the transverse dimensions of the ship. The effects of exact hydrostatic restoring forces, wave exciting forces, and a more correct formulation of large-amplitude hydrodynamics are all included in the new version of the LAMP code. With these extensions, the LAMP code is now capable of performing nonlinear time-domain motion simulations in large-amplitude waves including capsizing.

In this report, the methodology used to assess vessel stability and safety is discussed. It emphasizes the importance of performing analyses of dynamic stability rather than applying margins of safety to static stability criteria. The need for computational tools for accurate stability and safety assessment is addressed, especially in view of the recommendation by the 1993 Sub-Committee on Stability, Load Lines, and Fishing Vessel Safety (SLF) of the International Maritime Organization (IMO) to consider the use advanced computational methods within the new Load Line Convention.

Using the current LAMP code, the present study shows some examples of ship capsizing in beam seas. These results clearly demonstrate the necessity and power of a nonlinear time-domain simulation tool for the study of vessel stability and for the assessment of ship safety. However, an extensive validation study and possibly further improvements to the present method may be required for accurate predictions of extreme ship motions in more general extreme sea conditions.

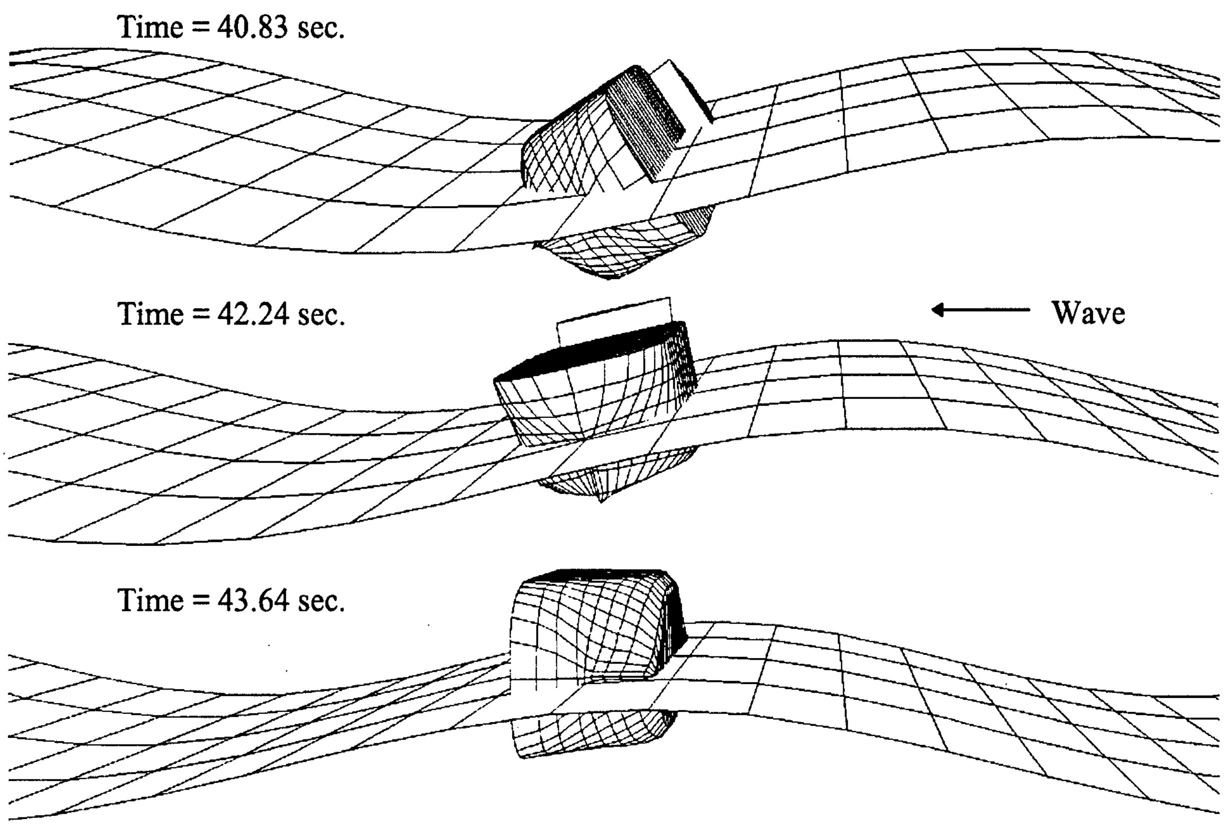


**Figure 1. Example of Ship Capsizing in Large-Amplitude Regular Beam Waves. Regular Wave, l = 402 ft (122.56 m), h = 32 ft (9.76 m)**

# 1 Introduction

One of the primary missions of the U.S. Coast Guard is the protection of life and property by the establishment and enforcement of marine safety standards. The ideal standard seeks to ensure safety without unduly affecting the ship's operability. A major safety concern is the prevention of the loss of life and property due to ship capsizing. Current regulations seek to prevent such occurrences by setting minimum stability and freeboard requirements. These regulations are based mainly on hydrostatics. They were developed from an analysis of, and experience with, traditional ship configurations.

The stability assessment of new, innovative ship forms and the assessment of capsizing accidents often require very expensive and time consuming experiments. The existing ship motion prediction tools are primarily based on hydrostatics and linear strip theory which can only be used for assessing small amplitude motions in moderate sea conditions. Therefore, an accurate ship motion simulation method may have a large impact on ship safety assessment.

Computational simulation techniques and computer architectures have finally reached such a level of sophistication that the development of a simulation system for vessel stability and safety assessment for extreme seas is a practical goal. The purpose of this report is to discuss the recent advances in computational hydrodynamics research and the related practical engineering systems, in particular the LAMP System (Lin & Yue, 1990, Lin, et al., 1992, 1993, 1994), for the assessment of the stability and safety of a vessel operating in extreme seas.

Since stability criteria are primarily based on static stability, it is extremely important to emphasize that the physics governing static stability is quite different from the physics for dynamic stability. To illustrate this point, we shall first look at a sailing yacht disaster which has been investigated extensively and which is quite well understood.

The Fastnet Race of 1979 is considered to be the greatest disaster in the history of the sport of yachting. Seventy-seven boats were completely capsized and fifteen sailors died (Rousmaniere, 1980). Stephens, Kirkman and Peterson (1981) analyzed the Fastnet disaster in their landmark paper on "Sailing Yacht Capsizing." They addressed the capsize mechanism, the environmental conditions, and the design approach which led to the terrible disaster. They pointed out that design practice has been to measure stability by static criteria and to compensate for dynamic effects through margins of safety. Their investigation of the Fastnet Race disaster revealed that "... certain factors which result in favorable static stability characteristics may actually present greater danger when considered in light of a dynamic analysis."

This is a very important aspect of vessel stability which unfortunately is often overlooked in setting safety requirements. For example, it is often assumed that a vessel's stability is a function of its freeboard with larger freeboard providing greater capsizing resistance. This is correct from a static point of view, but Stephens, et al. showed that the dynamics of the single wave impact capsizing mechanism which dominated the Fastnet '79 casualties (see Figure 2) had the opposite effect. They asserted that,

> ... freeboard, which helps raise the zero-stability crossing in a static case and hence appears as safe, is the source of much overturning energy being impacted to the yacht due to the large area being struck by the breaker and the increased moment arm acting for overturn.

Furthermore, Stephens, et al. concluded that

> ...the beam contribution to static stability is washed out in a capsize by a corresponding moment caused by local wave slope.

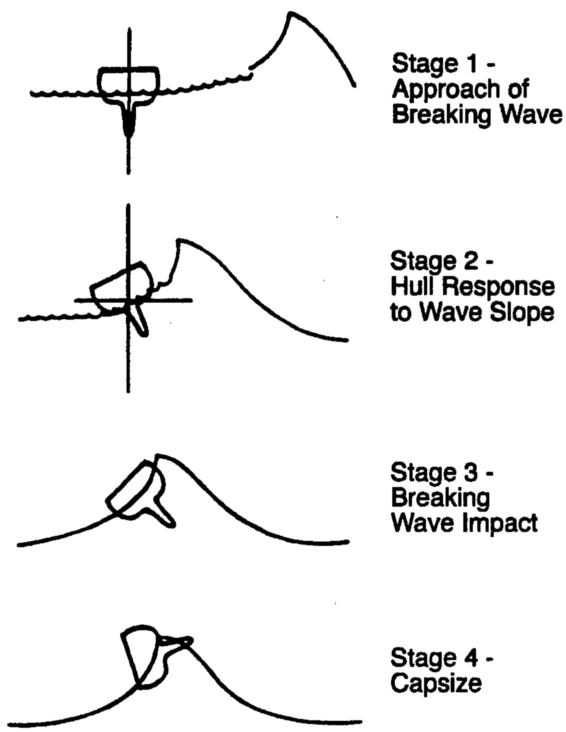


**Figure 2. Schematic of the Single Wave Impact Capsizing which Dominated the Fastnet '79 Casualties (from Stephens, et al., 1981)**

The reason for reviewing the Fastnet Race disaster is to stress the importance of analyzing dynamic stability in extreme sea conditions and to focus attention on the fact that there is a fundamental difference between static and dynamic stability. Vessel safety requirements cannot be established by considering static stability alone and then applying some safety factor to include dynamic aspects. Design trends driven by static stability requirements resulted in a catastrophic disaster in the Fastnet Race of 1979 because they increased freeboard and increased beam, causing dynamic stability problems.

In this report, we address the research, development, and application of advanced computational methods for the assessment of dynamic stability in a seaway. Static stability is assumed to be well understood. It is important to recognize that the advanced computational hydrodynamic tools discussed in this report are equally important for addressing all of the following problems related to vessel safety in waves:

- Dynamic Stability
- Structural and Equipment Damage, and
- Crew Safety.

However, in this report, we focus on the dynamic stability problem. Stability as it is discussed here will include both intact and damaged stability.

The development of computational tools for dynamic vessel stability must be considered as a portion of the much larger topic of seakeeping. Seakeeping assessment, which includes both the wave-induced motions and hydrodynamic loads, may be divided into two classes:

- Linear frequency-domain predictions, and
- Nonlinear time-domain simulations.

Linear frequency-domain prediction methods have been extremely successful in determining sea state operability limitations for weapon systems on naval vessels (Kennel, 1985). Such methods have also been useful in estimating the wave induced loads for large ships (Liu, et al., 1992). However, the linear tools are based upon the assumption that both the motions and the wave amplitudes are small relative to the vessel's dimensions (in particular the draft). This is a serious limitation - the assumption is not valid in general for vessel response in extreme seas.

Nonlinear time-domain simulation is required to determine the vessel's response in extreme seas. Because of this requirement, dynamic stability predictions are an order of magnitude more complex than linear frequency- domain predictions. For example, the wave field description for extreme response prediction must contain much more detailed information than the wave energy-spectrum representation used for linear prediction.

In order to obtain the probabilistic estimates needed for setting safety standards, one has to apply a combination of deterministic and probabilistic calculations. The assessment of a vessel's dynamic stability in waves may be divided into three parts:

**Wave-Event Modeling** extreme wave characterization, selection of potentially dangerous extreme wave events, and detailed numerical modeling of the complex nonlinear hydrodynamics aspects of the selected wave events.

**Vessel-Response Simulations** an accurate time-domain simulation of the vessel's response to the selected wave events,

**Probabilistic Predictions** an estimate of the probability of occurrence of the wave/vessel encounters which will result in catastrophic responses.

The importance and development of these three parts have been addressed by Salvesen and Lin (1993) in their proposed SAFE SEAS System. The first and the third parts will not be discussed further in this report. The current development of the vessel-response simulations will be discussed here.

The current report is a study of the capsizing of a Series 60 merchant ship hull form in beam seas. This is the first U.S. Coast Guard report published on our efforts to develop nonlinear motion simulation capability for marine safety applications as part of the Interactive Design, Evaluation, and Assessment System (IDEAS) project. LAMP is the major development result of the project. Section 2 of this report addresses the need for computational tools which will examine current problems in stability and safety assessment for both conventional and unconventional vessels. Section 3 discusses the methodology of vessel-response simulation in general. The current status of the LAMP system for large-amplitude ship motions and wave loads is given in Section 4. Results of using the LAMP system in studying ship capsizing in beam seas will be presented in Section 5. Both time-domain simulations and some of the mechanism which causes the ship to capsize in beam seas will be given.

## 2 The Need for Computational Tools

The 1966 Load Line Convention (ICLL66) of the International Maritime Organization (IMO) is, along with the International Convention on Safety of Life at Sea (SOLAS), the primary document setting forth international ship safety standards. At the January 1993 IMO meeting of the Sub-Committee on Stability, Load Lines and Fishing Vessel Safety - 37th Session (SLF37), a reexamination of the ICLL66 was placed on the agenda. Considering the proliferation of novel ship designs for which the IMO lacks adequate regulations and the recent advances in analytical seakeeping prediction techniques, it was decided to establish a Working Group to address the "Revision of Technical Regulations of the 1966 Load Line Convention." The Working Group (of which the second author is a member) recognized not only the need for advanced computational tools in the revision of the Load Line Convention, but also that considerable advancement in the state-of-the-art is necessary.

Experimental simulations can be extremely useful in assuring vessel stability, particularly if the test facility can generate realistic extreme wave conditions. A good example of the outstanding experimental stability investigations which have been conducted are the yacht capsizing experiments by Kirkman, et al. (1983) at the U.S. Naval Academy's tank in Annapolis, Maryland. Such well conducted experiments can give the designers both a better understanding of the physics underlying the problem and invaluable design data. However, the utility of experimentation for assessment is severely limited by the cost and time required for model testing. Both routine design evaluation and large scale parametric studies require faster, less expensive results than experiments can provide.

The general assessment of vessel stability and safety requires an advanced accurate numerical simulation system which can be used jointly with experimental investigation. The goal should be a simulation system which can be used to assess the stability of a large class of vessel designs over a large range of sea conditions. Also, the system must be capable of handling broad parametric sensitivity studies for both unconventional and conventional ship designs.

### 2.1 Unconventional Ship Design

Competitive market forces have caused naval architects to employ advanced technology in the greatly accelerated development of new unconventional ship designs which we see today. The worldwide development of high-speed ferries (Holden, Faltinsen, and Moran, 1991) is a good example. Building programs, most of which are ongoing outside the United States, include foilcats, surface effects ships, SWATHs, catamarans, hydrofoils and planing craft. For example, Westamarine A/S in Mandal, Norway is developing a high- speed ferry concept (W-1200) which will carry 1500 passengers and 500 cars. Typical speeds for the new high-speed ferries are in the 35-50 knot range, and the goal for the next generation of ferries is 60 knots. Large, high-speed cargo ships are also under consideration. For example, the Kvaerner-Masa Yards in Finland is presently working on a 600-ft prototype Ro-Ro vessel which will operate at a speed of 40 knots.

It will be a challenging task for the regulatory authorities to ensure that such high-speed ferries and cargo ships have a level of safety equivalent to that possessed by existing conventional ships. The safety issue for these novel concepts is made even more difficult by the fact that safety criteria for existing ships are not that well defined. In addition, the dynamic stability of high-speed craft operating in open sea is far from well understood. Even though model, prototype, and full scale evaluations of point designs will give us invaluable data, an advanced numerical simulation system for systematically assessing the safe operation of a range of high-speed craft designs is clearly desirable.

A large number of open-top container ships (ships without hatch covers and with the cargo holds open to the environment) are presently in operation and under construction. This has resulted in a challenging problem for the regulatory authorities which must ensure their safety in all operating conditions.

The authorities have had to rely on a limited number of model tests for the development of the safety requirements for the open-top container ship class. This is a good recent example where the regulatory authorities would have greatly benefited from an advanced numerical simulation capability.

## 2.2 Conventional Ship Design

Safety assessment is also a challenging problem for conventional ship designs. Between January 1990 and September 1991 thirty-six (36) bulk carriers suffered severe structural damage causing the loss of twenty-one (21) ships and two hundred and fifty (250) lives (Grove et. al, 1992). In most of these cases, structural failure due to hydrodynamic loads imposed by the seaway was the primary cause of casualty, and not capsizing or stability. New development of computational tools should include structural response prediction capabilities for extreme seas and therefore can be very useful for analyzing this class of safety problems.

The large number of safety issues related to vessel stability for fishing and pleasure craft are well known and documented. We have a tendency to give such problems second priority since the individual accidents involve relatively small dollar values and relatively few lives. However, the number of small craft stability accidents is quite substantial and this problem deserves more of our attention. As we shall see, several of the computational hydrodynamic methods included in the SAFE SEA System are uniquely tailored to address both fishing and pleasure craft problems including high-speed planing hulls.

## 2.3 The IMO New Load Line Approach

At the IMO SLF 37 January 1993 meeting it was decided to adopt the approach outlined in the paper on "The International Load Line Convention: Crossroad to the Future" by Alman et al. (1992) as a guide for the development of the new Load Line Convention. Figure 3 from IMO (1993) is a schematic outline of the new approach.

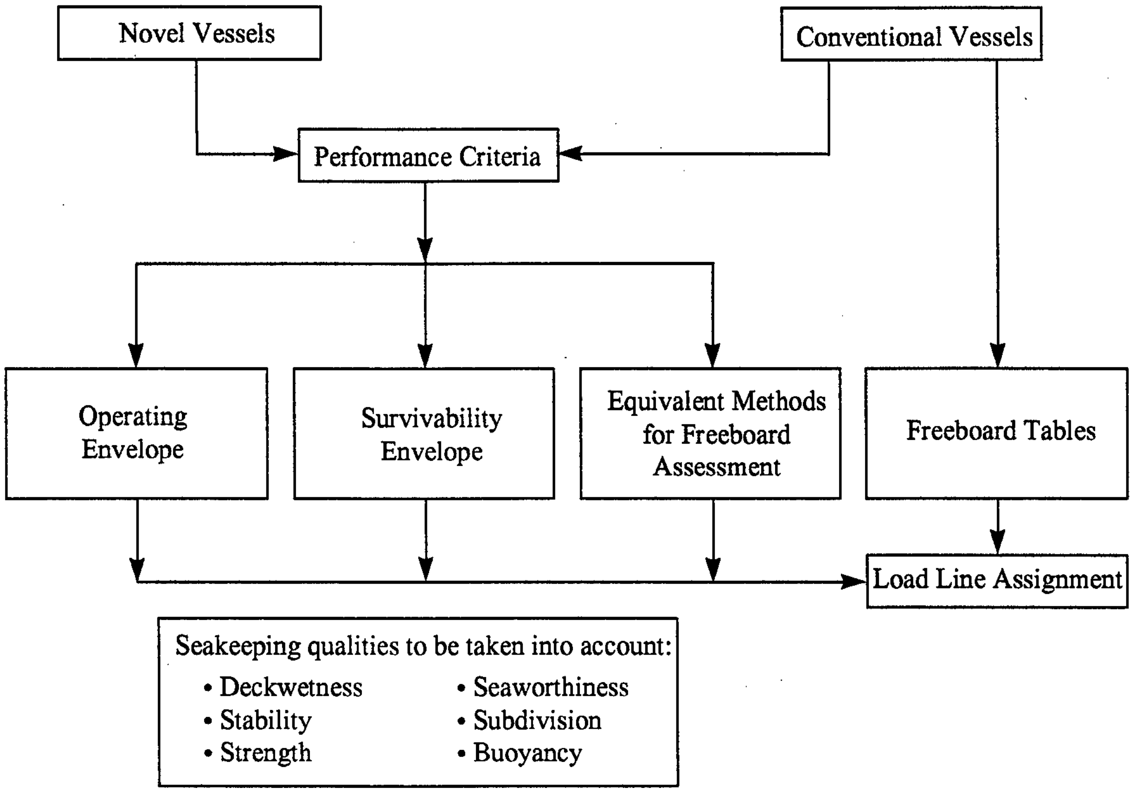


**Figure 3. Proposed Approach for the New Load Line Convention, IMO (1993)**

From Figure 3 it is seen that the proposed new Load Line Convention will have two separate paths. Path one is the well established Freeboard Table approach. Path two is the new Equivalent Level of Safety Assessment I approach based on the development of Safety Performance Criteria and new assessment techniques. The performance criteria will include deck wetness, stability, strength, seaworthiness, subdivision, and buoyancy. Vessels will be evaluated relative to these criteria by their operating envelope and survivability envelope, and by equivalent methods for freeboard assessment. Novel vessels will be required to use the new Equivalent Safety approach, whereas conventional vessels may use either the new approach or the Freeboard Table approach.

The establishment of the safety performance criteria and the development of equivalent assessment methods (experimental and computational) will be a challenging and demanding task. A new advanced numerical stability simulation capability could play a major role in the establishment of the safety criteria and the development of an equivalent method for freeboard assignment. We believe that if we in the United States do not take the lead in supporting the IMO with the needed technology, the development of the new Load Line Convention may not be as successful as would otherwise be the case.

# 3 Predictions of Motion Response

Advances in computational ship hydrodynamics over the past decade have resulted in increasingly capable and accurate computer codes for the prediction of ship motions. The application of these codes has been accelerated over the past few years by the ever-increasing power of modern computers, so that some of these advanced numerical calculations may now be done within design time scales. As a result of these advances, a new level of computational capability is now emerging for the prediction of the nonlinear ship motions in severe seas.

## 3.1 Linear Methods

Traditionally, the ship motion problem is formulated in the frequency domain, and linearized by assuming that the magnitude of the motions and the incident waves are small relative to the draft of the ship. The most commonly used linear tools presently available are based on the strip-theory originated by Korvin-Kroukovsky (1955). These tools were brought to the present state of development by a number of researchers in the United States, Europe and Japan during the mid-1960s. The U.S. Navy standard ship motion program, SMP is a typical code in common use by designers today. The later development of fully three-dimensional linear methods has also resulted in several useful codes. The most promising ones provide solutions using either the transient free-surface Green's function (eg, Beck & Magee, 1991; Lin & Yue, 1990; Bingham, et al., 1993) or the Rankine source methods (eg, Nakos & Sclavounos, 1990).

Linear methods have been very successful in many respects; for example, in determining sea state operability limitations for weapon systems on naval vessels (Kennel, et al., 1985). Such methods have also been useful in estimating the wave induced loads for large ships (Liu, et al., 1992). Unfortunately, computational methods based on the linear formulation have limited applicability to the very nonlinear dynamic stability problem.

## 3.2 Nonlinear Methods

Due to the severe limitations of the linear ship motion theories, several investigators have extended the frequency-domain strip-theory approach to large-amplitude time-domain strip theory approaches. In these large-amplitude approaches, the nonlinear hydrostatic restoring forces and the wave forces are calculated accurately whereas the hydrodynamic restoring and diffraction forces are calculated by some approximate extensions of the strip theories.

In the United States, such an approach has been applied by de Kat and Paulling (1989) to predict capsizing with some success. In particular, for low-frequency following seas their method showed very promising results. Outside the United States, such methods have had notable success in calculating the nonlinear global loads (bending moments and shear forces) (see for example, Fujino and Yoon, 1986). Approximate methods of this type can be very useful if they are applied carefully and with full understanding of their limitations.

The more recent research efforts in the United States have been focused on the development of more advanced 3-D nonlinear methods. These methods may be divided into two categories: fully nonlinear methods and approximate nonlinear methods. Typically, fully nonlinear methods address the exact free surface condition as well as the exact nonlinear body boundary conditions, whereas approximate nonlinear methods apply certain approximations to the nonlinear free surface conditions. Most of the theories in either category are formulated within classical potential flow theory. Good examples of the fully nonlinear approach are the work of Korsmeyer, et al., (1992), Maskew (1991), Cao, et al., (1992), and Yue (1994). The approximate methods of Lin & Yue (1990), and Beck & Magee (1991) solve the body-exact problem in which the free-surface condition is linearized. The work of Pawlowski and Bass (1991) is another example of an approximate nonlinear method. Tulin and Maruo (1992) address the nonlinear deck wetness problem with a promising approximate method based on a 2-1/2-D formulation.

It is believed that the approximate nonlinear approaches will result in practical and validated computational tools which can be run on modern advanced workstations within the near future. The fully nonlinear methods will require advanced supercomputers and for the near future will remain research codes serving as validation tools of the more approximate methods.

# 4 The LAMP System

In 1990, Lin & Yue presented a three-dimensional time-domain method to study the large-amplitude motions and loads of floating bodies in waves. In their so-called "body-exact" approach, the free-surface boundary conditions are linearized and the body boundary condition is satisfied exactly on the portion of the instantaneous wetted surface which lies below the undisturbed free surface. The problem is solved using a transient free-surface Green's function singularity distribution. The validity and practical utility of this method have been demonstrated by several studies including predictions of large-amplitude motion coefficients, motion history of a ship advancing in an irregular seaway, as well as the effect of bow flare on wave loads (see Lin & Yue 1990, 1992; Lin et al., 1991, 1992).

This method was employed for the prediction of motions and loads of a cruiser hull, CG47, in waves (Lin & Meinhold, 1991; Lin & Yue, 1993). The results were satisfactory for moderate seas but difficulties were encountered in severe seas. The difficulties arise from the fact that the body-exact approach models only that portion of the hull below the undisturbed free surface. When the wave amplitude is large compared to the ship draft, this representation becomes inadequate, especially near the transom stern.

## 4.1 New Formulation

In order to improve the Lin & Yue (1990) method and extend its applicability to more severe wave conditions, a new large-amplitude method has been developed in which both the body motions and the incident waves can be large (Lin & Yue, 1993). In this new Large-Amplitude Motion Program (LAMP), the body boundary condition is satisfied on the instantaneous wetted surface below the incident wave profile. The radiation and diffraction waves are part of the overall solution, but they are assumed to be small compared to the incident wave. In addition, the incident wave slopes are assumed to be small. This is typically the case for non-breaking wave which has a limiting slope of 1/7. At each time step, local incident free surface elevations are used to transform the body geometry into a computational domain with a deformed body and a flat free surface. By linearizing the free surface boundary conditions about this incident wave surface, the problem can be solved in the computational domain using linearized free-surface transient Green's functions.

Figure 4 shows a typical master geometry panel distribution. In the physical domain, the geometry will be cut at the incident wave surface, the cut geometry will then be transformed to the computation domain in the vertical direction. The solution procedures used for the problem in the computational domain are very similar to those used in the physical domain (Lin & Yue, 1993). Both the source formulation and potential formulation can be used. The two main features of this new large-amplitude approach are:

**i** true hydrodynamic effects for the wetted portion of the ship under the incident wave surface

**ii** automatic inclusion of the correct hydrostatic and Froude-Krylov forces.

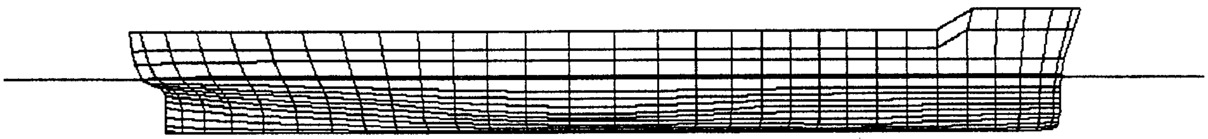

**Figure 4. Master Geometry and Panel Distribution for LAMP Computations**

In oblique or beam seas, forces due to viscous and lift effects will have a significant effect on the motions and loads. LAMP includes an option to approximate these effects in the time-domain. The viscous and lift effects approximated are as shown in Table 1 following the recommendation of Himeno (1981). For each effect, the table presents a reference for the calculation method and whether it is a linear or non-linear effect. These components are determined in a manner very similar to that used in the U.S. Navy's SMP code (Meyers et al., 1981). However, in the SMP code, the forces are calculated in the frequency domain, assuming certain averaged magnitudes of roll displacement and roll velocity.

**Table 1. Viscous and Lift Effects**

| Effect | Reference | Linearity |
|---|---|---|
| Hull Lift | Low Aspect Ratio Lifting Theory | Linear |
| Skeg, Bilge Keel and Foil Lift | High Aspect Ratio Lifting Theory | Linear |
| Hull Eddymaking | Tanaka (1960) and Ikeda et al. (1978) | Non-Linear |
| Bilge Keel Eddymaking | Kato (1966) | Non-Linear |
| Skeg and Foil Eddymaking | Hoerner (1958) and Ikeda et al. (1978) | Non-Linear |
| Hull Skin Friction | Kato (1958) | Non-Linear |

Such an averaged roll damping approach is not satisfactory for time domain calculations where a primary objective is the accurate calculation of the extreme response events. The new calculation method uses the formulae from the references in Table 1, but uses the current magnitude of roll displacement and roll velocity rather than an averaged value. At every time step, the time history of roll displacement and roll velocity is examined for a peak value, positive or negative. These peak values generate parameters for the viscous forces until a new peak is found. At any given time step, the actual forces depend on these parameters and the instantaneous value of roll displacement and roll velocity. This approach is very different from the approach used in SMP which uses an iterative process to calculate an "equivalent" or "averaged" roll amplitude for viscous damping. The current approach is a more direct calculation taking advantage of the fact that the roll angle and velocity are known at all time.

In the present version of LAMP the incident wave can be represented by a superposition of any number of harmonic wave components at any direction relative to the ship's heading. Given a wave spectrum, the program will generate an irregular wave representation automatically with random phases and a pre specified spreading function. Irregular wave representations for multiple spectra can also be generated. The wave field may also be represented by higher-order Stokes waves.

For any given wave representation, LAMP will calculate the time-domain six-degree-of-freedom coupled motions and the time-domain wave-induced global loads, i.e. the bending and torsional moments and shear forces at any cross section along the length of the ship. The program also calculates the hydrodynamic pressure distribution over the instantaneous wetted hull surface below the incident wave surface at each time step. Furthermore, the added resistance in waves as well as the wave resistance can be calculated. Typically the program is run with the ship advancing at a given heading angle and constant forward speed; however, any path and/or speed may be specified.

### 4.2 The Multi-Level LAMP System

A complete computational capability for the assessment of ship motions and wave loads must be based on a multi-level approach. Such a system integrates methods which are based not just on one single code or one single level of sophistication, but rather on a system of codes with different levels of sophistication. As a general rule, the physics underlying the ship/wave interactions is best understood using comparisons generated by incremental increases in complexity - a procedure which also moderates computer usage. Analysis tools at the lower levels may employ several approximations to attain a short enough turnaround time for use in early stages of the evaluation process. An examination of the results obtained by the lower level code guides the engineer in choosing areas where more accurate theories must be used. In other words, the lower level codes should be used as a filtering mechanism for the selection of more accurate but more complicated and computationally intensive codes.

A multi-level system can also effectively tie the probabilistic and deterministic approaches together providing the missing ingredient of probabilistic prediction. Statistical data of ship motion in given random seas can be obtained by using lower level evaluation codes to efficiently compute the ships responses to a very wide range of deterministic excitations. The severe ship responses can be selected from these, to be examined with the higher level nonlinear simulations. Conversely, nonlinear dynamic simulations of ships in episodic wave events can be used to understand the actual physical mechanisms underlying the ship responses to these events, such as capsizing, and to identify dominant factors of vessel stability, which can be used in the statistical screening process using the lower level codes.

Recognizing the need for a fully integrated multi-level code system, we have developed the Interactive Design, Evaluation and Analysis System (IDEAS) consisting of a total of four computational methods of different levels of sophistication.

| | |
|---|---|
| LAMP-4: | The large-amplitude 3-D nonlinear method |
| LAMP-2: | The approximate large-amplitude 3-D nonlinear method |
| LAMP-1: | The linearized 3-D time-domain method |
| SMP: | The U.S. Navy linear strip-theory Ship Motion Program |

As shown in Figure 5, the total capability is labeled the IDEAS Ship Motion and Wave Load System. The most advanced code is the Large Amplitude Motion Program, LAMP-4 discussed in the previous section. Three simplified versions of the LAMP-4 code have also been developed. The lowest level code uses the linear strip theory.

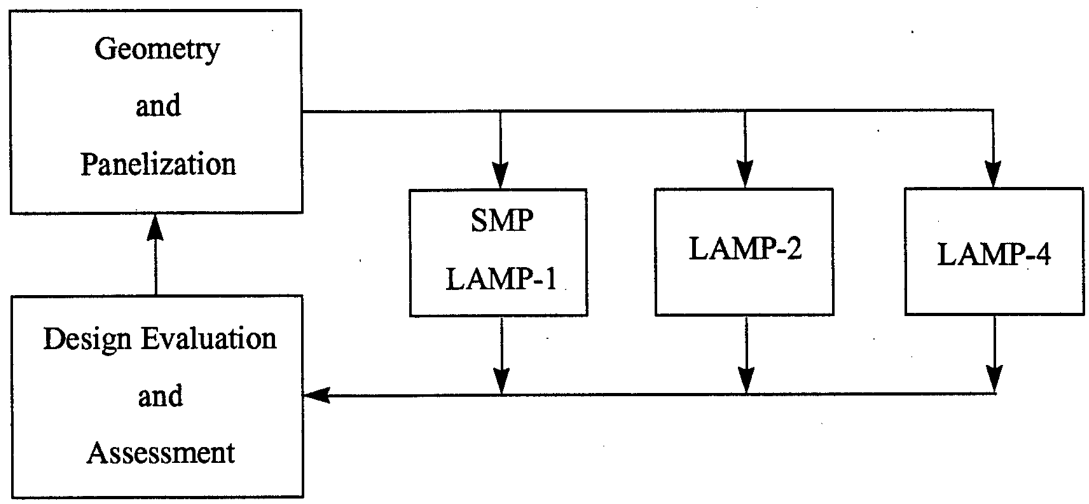


**Figure 5. The Present IDEAS Ship Motions and Wave Load System**

The LAMP-4 method is the complete large-amplitude method where the 3-D potential is computed with the linearized free-surface condition satisfied on the surface of the incident wave. Both the hydrodynamic and hydrostatic pressure are computed over the instantaneous hull surface below the incident wave surface. Large computer resources are required for this method.

In the LAMP-2 method, the linear 3-D approach is used to compute the hydrodynamic part of the pressure forces. However, the hydrostatic restoring and wave forces are calculated with the same accuracy as in LAMP-4. The reason for developing this simplified method is that it drastically reduces the requirements for computer resources.

The LAMP-1 method is the linearized version of the LAMP-4 method. This 3-D time-domain method includes a routine for automatic generation of the frequency domain results.

The SMP is the linear strip-theory code presently used by the U.S. Navy. It is based on the theory developed by Salvesen, Tuck and Faltinsen (1970).

Table 2 shows how the hydrostatic restoring and wave forces and the hydrodynamic (added mass, damping and diffraction) forces are calculated for the four different LAMP methods. The hardware requirements for the four methods are also shown in the Table. Note that all of the nonlinear methods, LAMP-2 and LAMP-4 are based on the approach that both the motions and the waves may have large amplitudes. For all of these three nonlinear methods, the restoring and Froude-Krylov forces are calculated exactly over the instantaneous wetted surface below the incoming wave surface.

**Table 2. Computation Methods and Hardware Requirements for the LAMP Code. ($Z = 0$ and $F$(t) are Still Water Surface and Incident Wave Surface Respectively)**

| Method | Hydrodynamic, Restoring and Wave Forces |
|---|---|
| LAMP-4 | Free Surface Boundary Conditions on $F$(t)<br>3-D Large-Amplitude Hydrodynamics<br>Nonlinear Restoring and Wave Forces |
| LAMP-2 | Free Surface Boundary Conditions on $Z = 0$<br>3-D Linear Hydrodynamics<br>Nonlinear Restoring and Wave Forces |
| LAMP-1 | Free Surface Boundary Conditions on $Z = 0$<br>3-D Linear Hydrodynamics<br>Linear Restoring and Wave Forces |

# 5 Nonlinear Motion Simulation of Series 60, $C_B$=0.7 Ship in Beam Seas

## 5.1 Ship Characteristics

To demonstrate the application of the LAMP System to dynamic stability in waves, a series of computations have been performed using LAMP-2 for a Series 60, $C_B$=0.7 ship in regular beam waves. LAMP-2 is considered a suitable level of computation tool for this particular application since it does not require intensive computation effort and can take into consideration the nonlinear hydrostatic and wave forces. These forces are important from the dynamic stability point of view. The ship's major dimensions and other properties important to the capsizing problem are described in this section. The principle dimensions of the ship are presented in Table 3.

**Table 3. Full Scale Ship Particulars for Study**

| | |
|---|---|
| LBP | 400 ft (121.95 m) |
| Beam | 57.14 ft (17.42 m) |
| Draft | 22.86 ft (6.97 m) |
| $C_B$ | 0.70 |
| Displacement | 10,460 LTSW |
| Freeboard Amidships | 30 ft (9.15 m) |
| VCG | 20.79 ft (6.34 m) above Baseline |

A panel representation of the ship is given in Figure 6. The center of gravity, CG is located amidships and is 2.07 ft (0.63 m) below the design waterline in the current computation. The actual location of the center of gravity depends on the weight distribution of the ship and can be a very important factor in the stability computation.

### 5.1.1 Static Stability by Coast Guard Regulation

As part of the analysis, standard stability checks were made to confirm that the ship was statically stable. The U.S. Coast Guard requires that the Metacentric Height, or GM, have a minimum value in all loading conditions. This requirement is based on the ship's ability to resist expected heeling moments due to high winds. From the Code of Federal Regulations, (46 Shipping), Chapter 1, Subchapter 5, "Subdivision and Stability," part 170, Subpart E, the following equation specifies minimum GM for any large cargo ship:

$$GM_{min} = \frac{PAH}{W\tan(T)} \qquad (1)$$

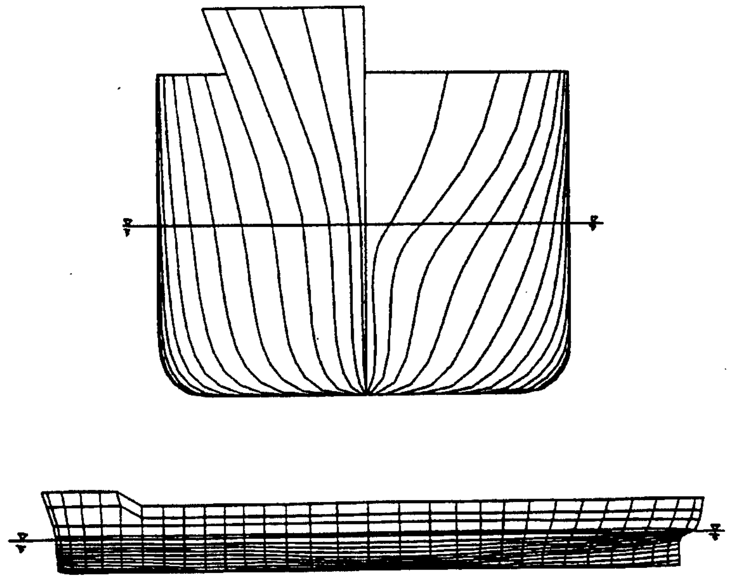

**Figure 6. Geometry and Panel Representation of the Series 60, $C_B = 0.7$ Cargo Ship Hull Form.**

Where :

$GM_{min}$ = minimum metacentric height

$$P = 0.005 + \left(\frac{L}{14200}\right)^2 \frac{\text{tons}}{\text{ft}^2} \text{ for ocean service}$$

$L$ = Length Between Perpendiculars, ft

$A$ = Projected Lateral Area above the waterline,

$H$ = Vertical distance from the centroid of A to one-half the draft

$W$ = Displacement in Long Tons

$T$ = 14 degrees or heel angle at which half the freeboard to the deck edge is submerged, whichever is smaller

The formula is suitable for the English units specified above. Metric equivalent units can be obtained after GM is calculated. We determined the following values, assuming the ship is a typical break-bulk cargo ship, with machinery aft, and a length of 400 feet (121.95 m). For a model of the above-deck profile, we used a profile of an actual cargo ship, taken from (Dillon et. al., 1962), and shown in Figure 7.

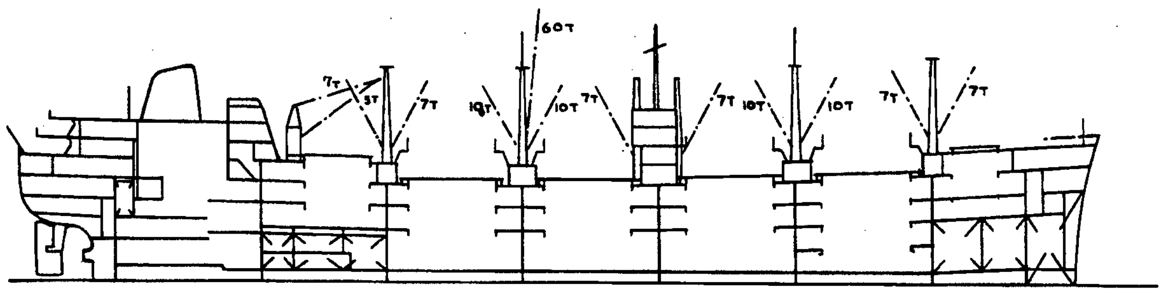

Fig. 41 Profile of new American Export Lines' cargo ship with machinery aft

**Figure 7. Profile of American Export Lines Cargo Ship With Machinery Aft, from Dillon et. al., 1962**

| | | |
|---|---|---|
| $L$ | = | 400. Ft (121.95 m) |
| $P$ | = | $0.005793\left(\dfrac{\text{ton}}{\text{ft}^2}\right)$ |
| $A$ | = | 10,790 $\text{ft}^2$ (1002.94 m) Determined by integrating areas from Figure 7 scaled to an LBP of 400 ft (121.95 m). |
| $H$ | = | 20 ft (6.10 m), Estimated based on Figure 7 |
| $W$ | = | 10,457 Long Tons, Salt Water |
| $T$ | = | 14 Degrees (See Figure 8) |

Therefore, for the minimum GM is

$$GM_{\min} = \frac{(0.005793)(10790)(20)}{(10457)\tan(14°)} ft = 0.479 ft = 0.146 m \qquad (2)$$

This is a small value, but it is not surprising for this ship in the full load condition, with relatively little windage. The value used in the LAMP calculations was GM = 2.204 feet (0.617 m). If we were to model the ship in a ballast condition (as would be required for Coast Guard approval), we would expect to find a GM closer to the minimum value. In ballast condition, the freeboard and therefore windage would increase, increasing the required minimum GM. The displacement would go down and the center of gravity would likely rise, decreasing the actual GM. The conclusion is that the GM we used is not excessive, even though it exceeds the requirement for this condition.

### 5.1.2 Navy Wind Heeling Standards

A more careful look at stability requires us to generate the ship's intact righting arm (or GZ) curve. This was generated using LAMP-2 and is shown in Figure 8. Superimposed on the figure is a body plan view of the inclined ship geometry at a heel angle of 38 degrees, showing that the deck begins to be immersed at a this angle. For larger heel angles, the GZ values are questionable, since deck openings would need to be considered for a real ship. Above this point, the GZ curve is shown as a dashed line. The curve also indicates that stability vanishes at a heel angle of 121 degrees.

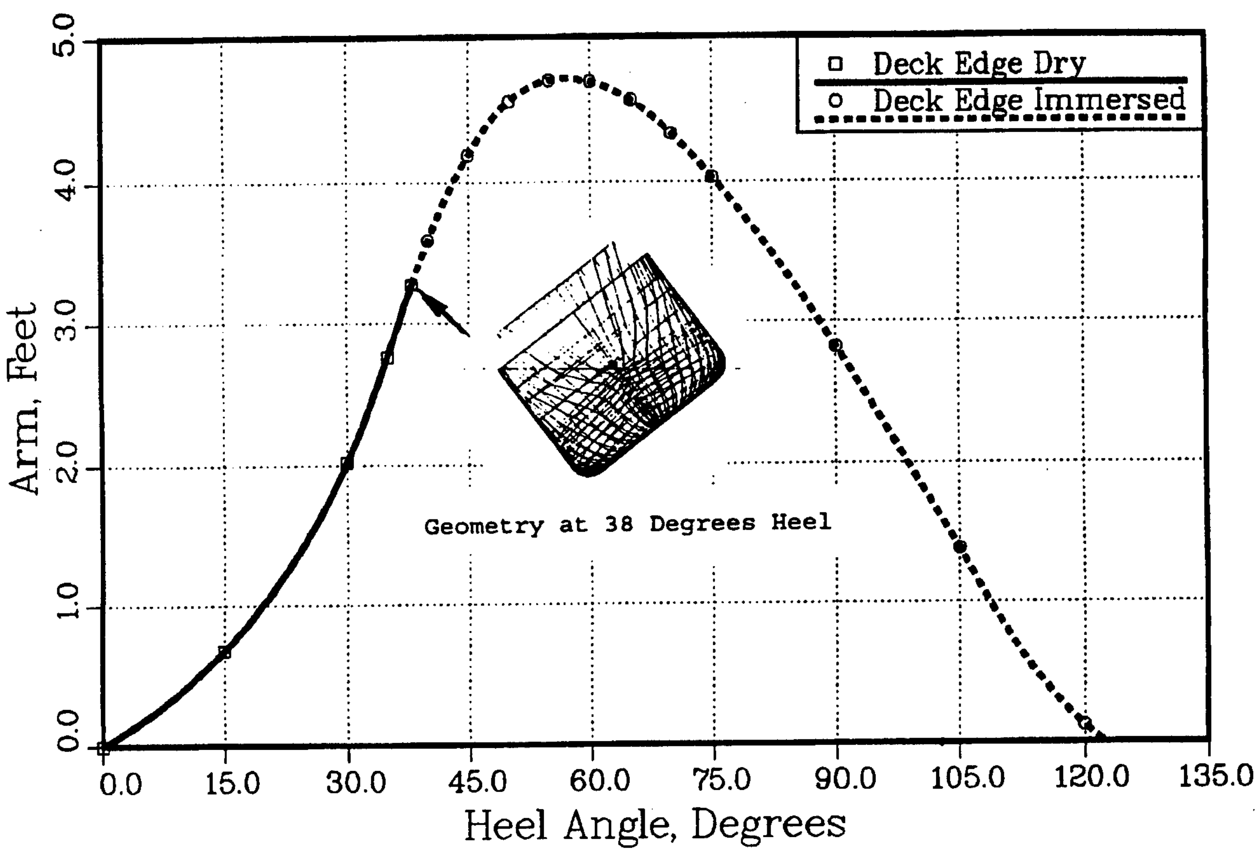


**Figure 8. Righting Arm and Flooding Point for Series 60, VCG=20.79 ft (6.34 m)**

A more detailed static stability evaluation can be performed using U.S. Navy Standards described by Sarchin and Goldberg (1962). These criteria are also based on expected wind moments. In this analysis, a wind-heeling arm corresponding to the righting arm is calculated using the following formula,

$$\text{Wind Heeling Arm, in ft} = \frac{0.004V^2 Al\cos^2\theta}{2240\Delta} \qquad (3)$$

Where :

| | | |
|---|---|---|
| V | = | nominal wind speed, knots |
| $A$ | = | Projected Lateral Area above the waterline |
| $l$ | = | Vertical distance from the center of A to one-half the draft |
| $\Delta$ | = | Displacement in Long Tons |
| $\theta$ | = | heel angle |

Using the following values :

| | | |
|---|---|---|
| V | = | 100. knots |
| $A$ | = | 10,790 $ft^2$ (1003 $m^2$) |
| $l$ | = | 20 ft (6.10 m) |
| $\Delta$ | = | 10,457 Long Tons |
| $\theta$ | = | degrees |

$$\text{Wind Heel Arm} = \frac{(0.004)(100^2)(10,790)(20)\cos^2\theta}{(2240)(10,457)} \text{ ft} \tag{4}$$

$$\text{Wind Heel Arm} = 0.369\cos^2\theta$$

This equation is plotted along with the righting arm in Figure 9. The stability criteria are as follows:

a. The heeling moment at the intersection of the righting arm curve and the wind heeling arm curve is not greater than six-tenths of the maximum righting arm.

b. The area $A_1$ is less than $1.4A_2$, where $A_2$ extends 25 degrees to windward of the intersection.

The figure clearly indicates that both these criteria are met.

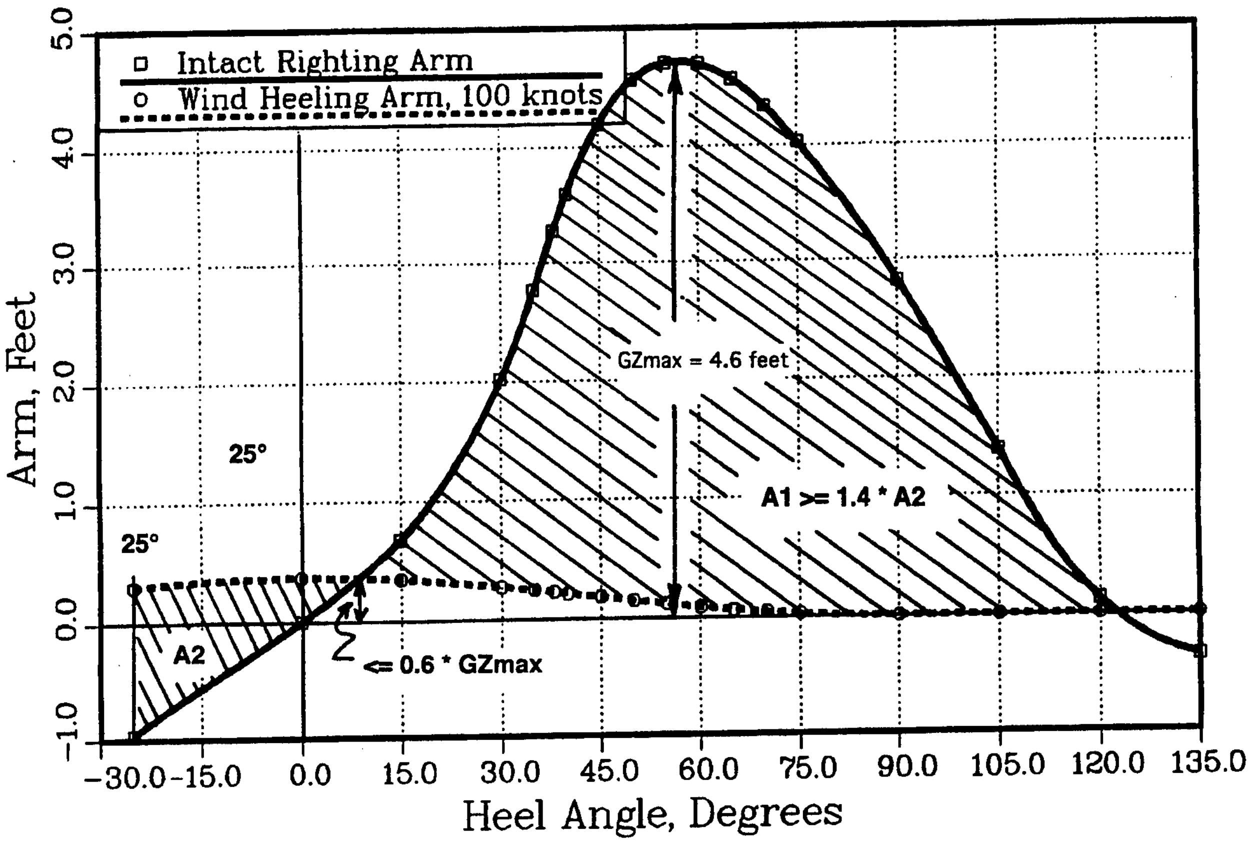


**Figure 9. Righting Arm and 100 Knot Wind Heeling Arm for Series 60, VCG=20.79 ft (6.34 m)**

### 5.1.3 Natural Roll Period

The roll period for the ship was determined using several methods. The purpose was to provide both a guide for choosing the test conditions, and as a check on the LAMP results. In *Principles of Naval Architecture*, the following empirical formula for estimating the roll period of ships is provided:

$$T_n = \frac{1.108k}{\sqrt{GM}} \qquad (5)$$

Where:

$T_n$ = roll period in sec

$k$ = roll gyradius in ft

$GM$ = metacentric height in ft

We assume the roll gyradius to be 35% of the beam, then

$$k = (0.35)(57.14) = 20.00 \text{ ft} = 6.10 \text{ m}.$$

GM = 2.204 ft from previous calculations, so,

$$T_n = 14.93 \text{ sec}$$

Using LAMP-2, the ship was given an initial heel angle of 11.4 degrees and allowed to roll freely. Figure 10 shows the time history of the free rolling. From this result, the natural roll period calculated by LAMP was found to be 15.09 seconds. This is very close to the approximated period supplied by the formula.

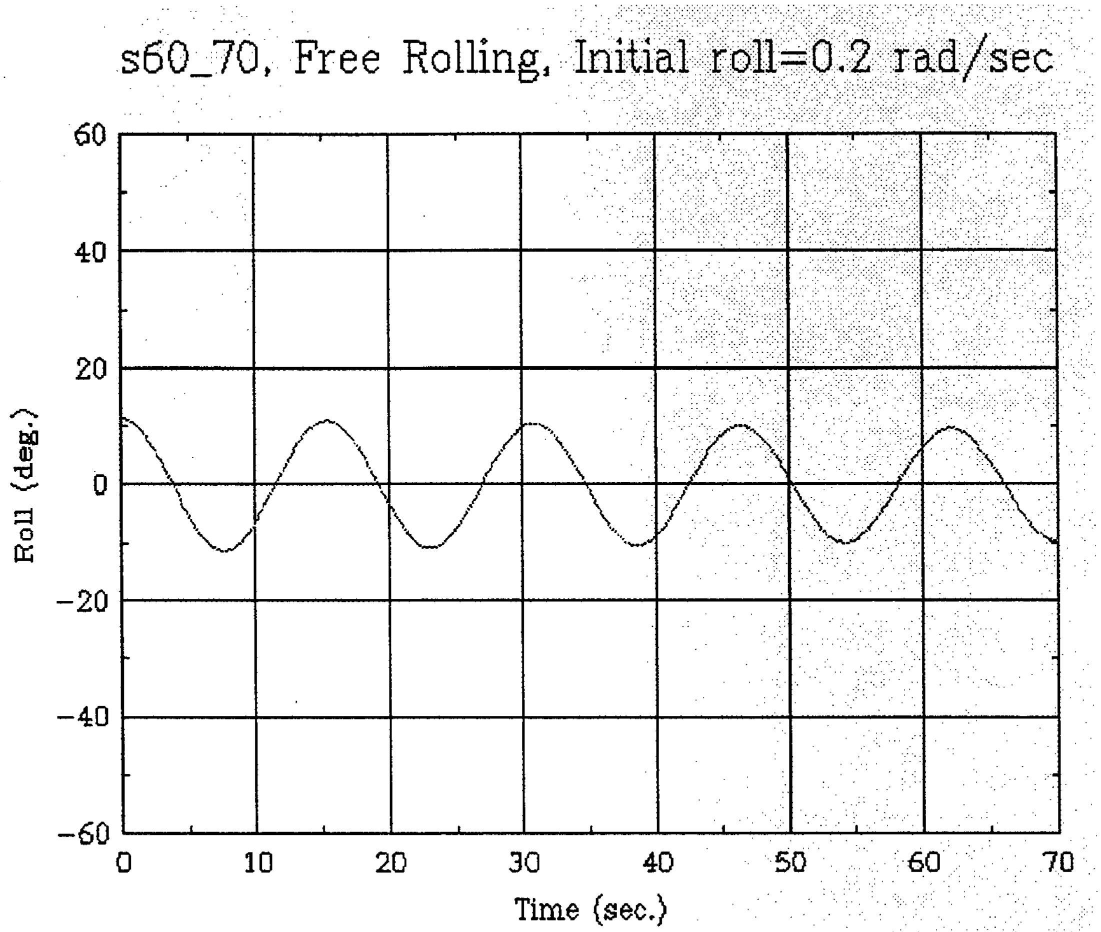


**Figure 10. Free Rolling Time History for Series 60.**

## 5.2 Nonlinear Motion Simulations

To demonstrate the use of the LAMP System for assessing the dynamic stability of ships in waves, a series of computations was performed using LAMP-2 for a Series 60, $C_B$=0.7 hull in regular beam waves. Only intact stability is studied. The center of gravity, CG, of the ship is located amidship and is about 2.07 ft (0.63 m) below the design waterline in the current study. The actual location of the center of gravity depends on the weight distribution of the ship and is a very important factor in the roll stability. The CG used in the current calculations is a reasonable one for such a cargo ship in full load condition. The importance of the CG location on ship stability and capsizing is demonstrated and discussed in the next section.

In principle, the LAMP code is applicable to general large-amplitude motion simulation, including the capsizing of ships, in a wide range of sea conditions. For the current study, linear regular incident waves are selected. The wave amplitude and wave frequency are varied to see the effects of these parameters on capsizing. The forward speed effect is considered to be of little importance for beam sea capsizing. Therefore, a two-dimensional (heave and roll) zero speed condition was specified in all LAMP runs. The incident wave conditions for these runs are listed in Table 4. Note that w is the wave frequency, l is the wave length, h is the wave height, and L is the ship length. All waves selected are bounded by the "steepest wave" limit (wave height / wave length < 1 / 7 = 0.1429 ).

**Table 4. Cases of LAMP Runs**

| w (rad/sec) | l (ft / m) | h (ft / m) | h / L | h / l |
|---|---|---|---|---|
| 0.4256 | 1117 / 340 | 16 / 4.88 | 0.04 | 0.0143 |
| 0.4256 | 1117 / 340 | 32 / 9.76 | 0.08 | 0.0286 |
| 0.4256 | 1117 / 340 | 48 / 14.63 | 0.12 | 0.0430 |
| 0.4256 | 1117 / 340 | 64 / 19.51 | 0.16 | 0.0573 |
| 0.7093 | 402 / 123 | 8 / 2.44 | 0.02 | 0.0199 |
| 0.7093 | 402 / 123 | 20 / 6.10 | 0.05 | 0.0498 |
| 0.7093 | 402 / 123 | 32 / 9.76 | 0.08 | 0.0796 |
| 0.7093 | 402 / 123 | 44 / 13.41 | 0.11 | 0.1095 |
| 0.7093 | 402 / 123 | 56 / 17.07 | 0.14 | 0.1393 |
| 0.9079 | 245 / 75 | 8 / 2.44 | 0.02 | 0.0327 |
| 0.9079 | 245 / 75 | 16 / 4.88 | 0.04 | 0.0653 |
| 0.9079 | 245 / 75 | 24 / 7.32 | 0.06 | 0.0980 |
| 0.9079 | 245 / 75 | 32 / 9.76 | 0.08 | 0.1306 |
| 1.1349 | 157 / 48 | 8 / 2.44 | 0.02 | 0.0510 |
| 1.1349 | 157 / 48 | 20 / 6.10 | 0.05 | 0.1274 |

Heave and roll motion time histories of these motion simulations are given from Figure 11 to Figure 14. It can be seen in these figures that the heave motions are approximately linear in the frequency range studied (although there are initial transient responses and obvious coupling between heave and roll motions). On the other hand, the roll motions are very nonlinear and depend strongly on frequency and height of the incident wave. A strong modulation effect between the roll natural frequency and the incident wave frequency can also be seen.

Figure 11 shows that the roll angle is not increasing as a function of h (wave height) at this low frequency (long wave length). The frequency of the roll responses is the same as the incident wave frequency for the h=16 ft (4.88 m) and 32 ft (9.76 m) cases. The incident wave frequency (w = 0.4256 rad/sec) in this case is very close to the natural roll frequency (w = 0.41 rad/sec) of the ship. However, the resonance effect does not seem to be very strong. For the h = 16 ft (4.88 m) case, the roll response increases slightly with time. However, this is not true for the h=32 ft (9.76 m) case. For the h=48 ft (14.63 m) case, ship starts to respond at a higher frequency in roll. For the h=64 ft (19.51 m) case, the response frequency is almost twice the frequency of the incident wave. It is believed that this increase in response frequency is due to a nonlinear roll restoring moment.

The maximum roll response of the ship at this incident wave frequency is close to 80 degrees. Since deck edge immersion occurs at 38 degrees, downflooding is likely to occur on a real ship. This indicates that capsizing is likely to occur even at such a low frequency. However, a typical sea condition has very little energy in the low frequency range. Therefore, this is not the region to be concerned about. It should be noted that from the static stability consideration, the freeboard will submerge when the roll angle is large than 38 degrees. Water on deck or down-flooding of compartments may be a concern for ships with large roll motions.

For the w = 0.7093 rad/sec (l = 402 ft = 122.56 m) case, the ship rolls over almost immediately when the wave height reaches 56 ft (17.07 m). When the wave height is 44 ft (13.41 m), the ship rolls back and forth several times and eventually capsizes. For the h = 32 ft (9.76 m) case, the roll angle is growing rapidly and capsizes after 6 cycles. This is an interesting case of ship capsizing with roll angle growing with each wave cycle. The process of capsizing can be understood better from the time history of ship roll motion given in Figure 15. From the actual animation, it can be seen that the ship is rolling in the counter clockwise direction while the wave crest is approaching from the right. The starboard side hit the wave and the ship starts to roll back toward the port side. Eventually, the ship capsizes near the crest of the wave. This is a dangerous situation and dynamic stability is definitely an important consideration.

For the w = 0.9079 rad/sec and 1.1349 rad/sec cases (Figures 13 and 14), the ship experiences large roll motions while the wave is approaching the "steepest wave" limit. Note that in the current study, linear regular incident waves are used. Nonlinear or random incident waves may introduce additional dynamic effects on the roll motion.

The results of the simulation runs are summarized in Figure 16. The maximum roll angle of each run is represented by a symbol in the figure to indicate safe, water on deck (roll angle <

38 degrees), or capsize. As can be seen, this ship is very stable in the current loading condition and it will capsize at only one combination of frequency and wave height. However, only intact stability is studied here. Water on deck should be taken into consideration for the frequency and wave height combination in the shaded area below the "steepest wave" limit. For ships in high frequency incident waves, capsizing or water on deck will not occur from the roll motion point of view. However, breaking waves are likely to exist in this region. Nonlinear wave effects and spray created by breaking waves may be important.

### 5.3 Effect of the Location of the Center of Gravity

As discussed in the previous section, this ship is very stable at the full load condition. For real ship operation, it is necessary to consider not only the full load situation but also the ballast condition. Motion simulations were done for this ship with reduced GM in one wave condition. The righting arm curves for this ship with three different GM are given in Figure 17. All three conditions satisfy the minimum GM requirement. A GM = 2.2 ft (0.67 m) is associated with the original loading condition.

Heave and roll motion time histories of this ship with three different values of GM are given in Figure 18. The incident wave frequency is 0.9079 rad/sec and the wave height is 32 ft (9.76 m). The ship capsizes when the GM is reduced to 0.73 ft (0.22 m). This is consistent with our experience and is a clear indication that loading condition is critical to the stability of the ship. As pointed out previously, the Coast Guard requirement for this vessel is a GM = 0.48 ft (0.15 m).

# 6 Summary

A new numerical simulation method, LAMP, has been developed for studying extreme motion, including capsizing of ships in beam seas. A sample study of the static and dynamic stability of a typical cargo ship (Series 60, $C_B$=0.7 hull) is presented in this report. LAMP-2 motion simulations of the ship in various incident waves were performed. LAMP-2 is a weakly nonlinear method with linear hydrodynamic force and nonlinear hydrostatic restoring and wave forces. It was found that the ship was stable at the full load condition in most cases, but that the ship may capsize due to nonlinear restoring and extreme wave conditions.

Only intact stability was considered in this study. From the numerical results, it is found that water on deck may be important for ships in several different wave conditions. Loads due to water on deck or possible down flooding of compartments should be taken into consideration in a future study.

The loading condition directly affects the location of the center of gravity, therefore GM, of the ship. The effect of GM on roll stability is demonstrated. It is shown that although the ship satisfies the minimum GM requirement, it will still capsize especially for those ships with small GM. A similar study using a less stable ship which nevertheless satisfies all stability regulations would be interesting.

The current LAMP-2 simulations were limited to two degrees of freedom (heave and roll) and is at the zero speed condition. Several other important factors such as wind effect, bilge keel effect, and nonlinear waveeffects were not modeled and should be included in a future study.

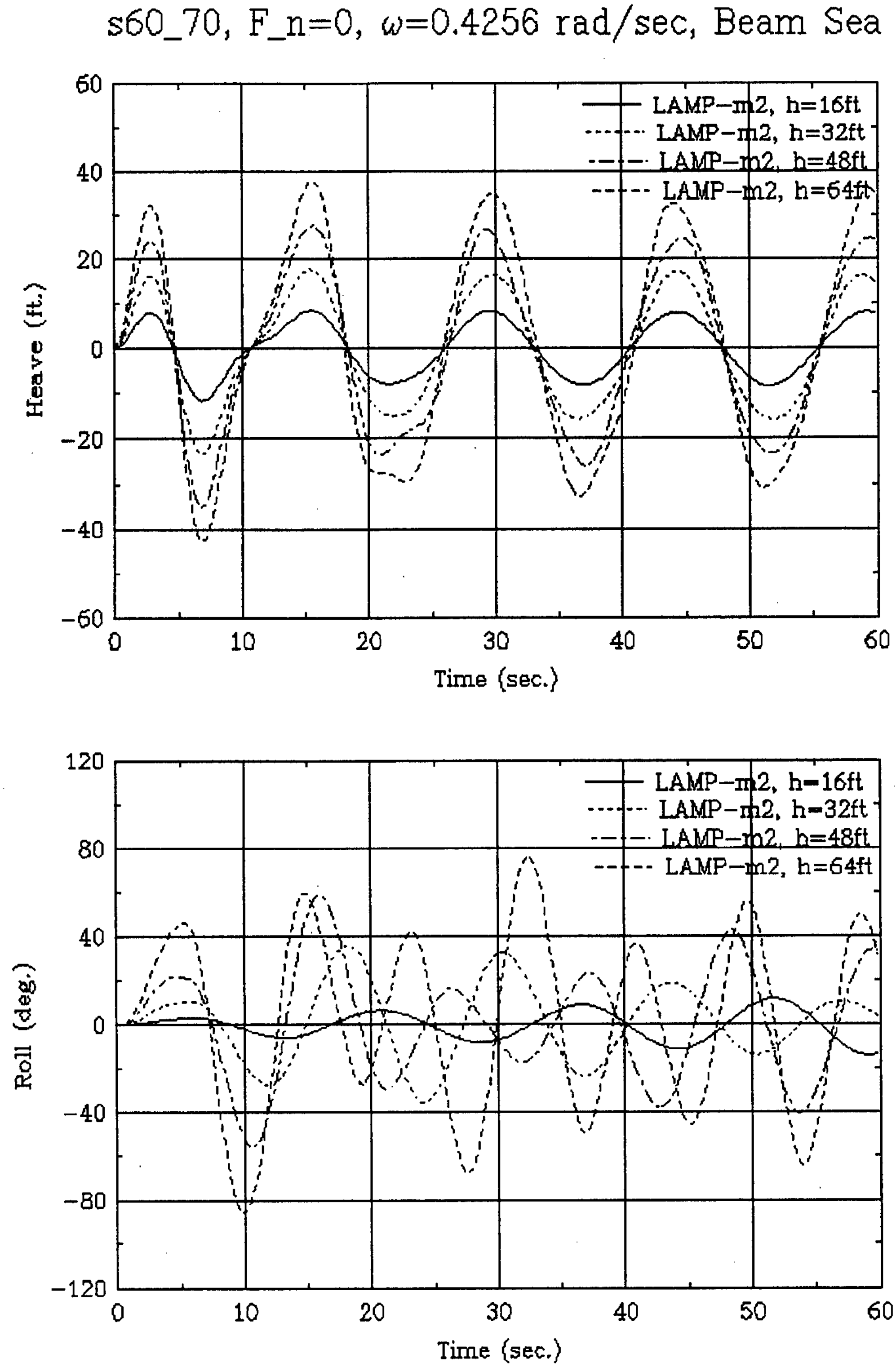


**Figure 11. Time History of Heave and Roll Motions of the Series 60, $C_B$=0.7 Hull in Linear Regular Beam Waves with w = 0.4256 rad/sec and l = 1117 ft (340 m) . GM = 2.204 ft, at full load displacement.**

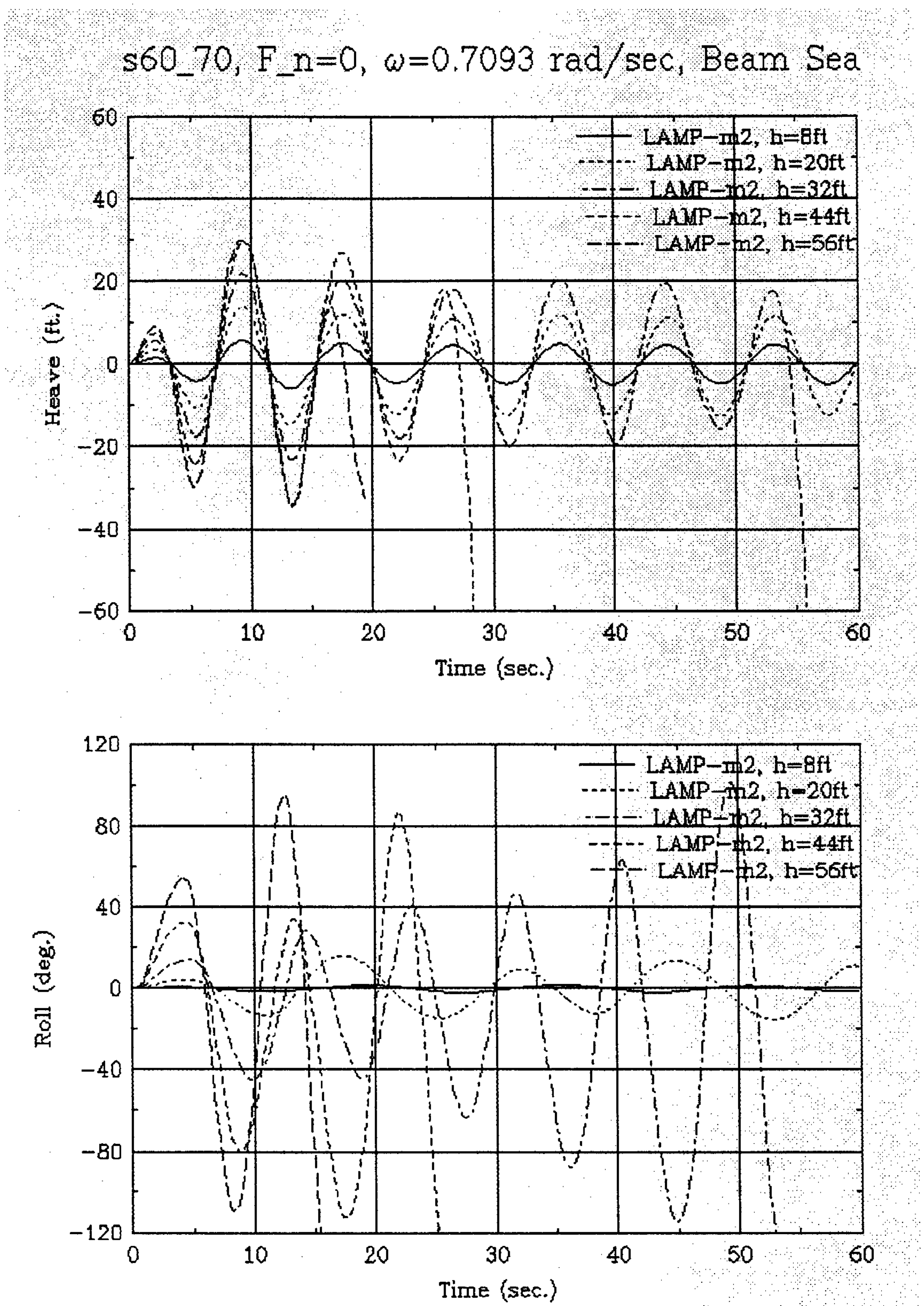


**Figure 12. Time History of Heave and Roll Motions of the Series 60, $C_B$ = 0.7 Hull in Linear Regular Beam waves with w = 0.7093 rad/sec and l = 402 ft. GM = 2.204 ft, at full load displacement.**

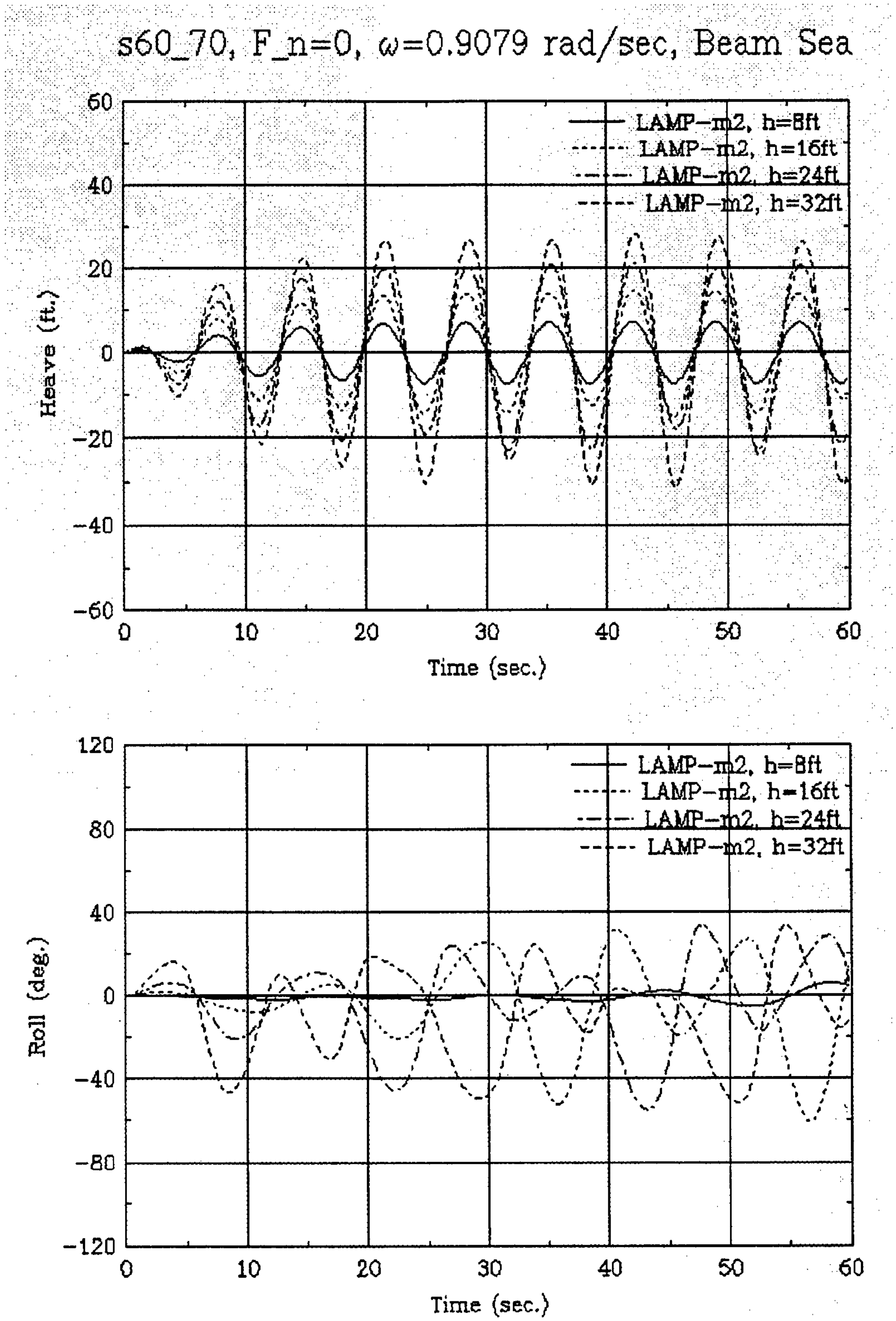


**Figure 13. Time History of Heave and Roll Motions of the Series 60, $C_B$ = 0.7 Hull in Linear Regular Beam Waves with w = 0.9079 rad/sec and l = 245 ft. GM = 2.204 ft. at full load displacement.**

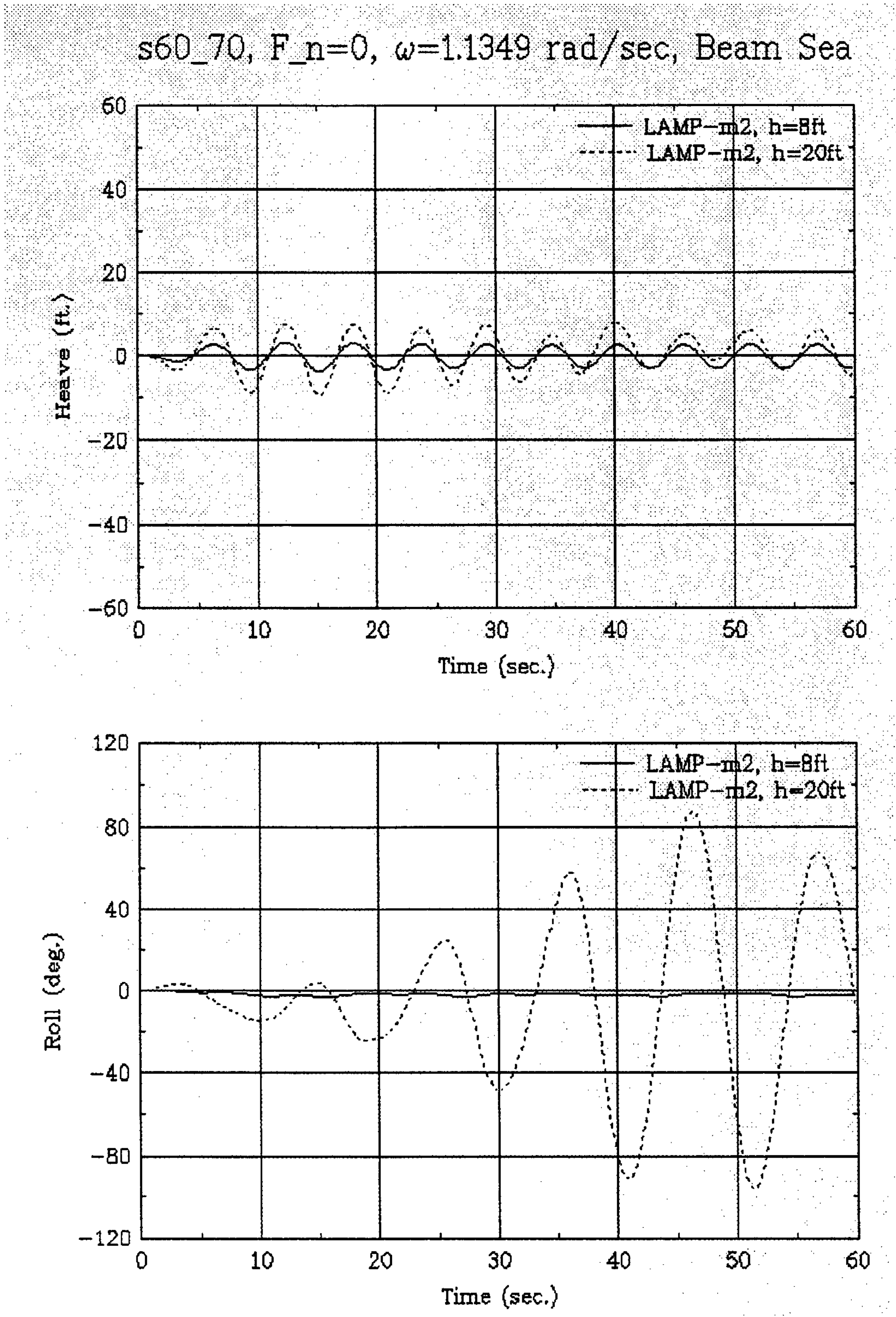


**Figure 14. Time History of Heave and Roll Motions of the Series 60, $C_B$ = 0.7 Hull in Linear Regular Beam Waves with w = 1.1349 rad/sec and l = 157 ft. GM = 2.204 ft. at full load displacement.**

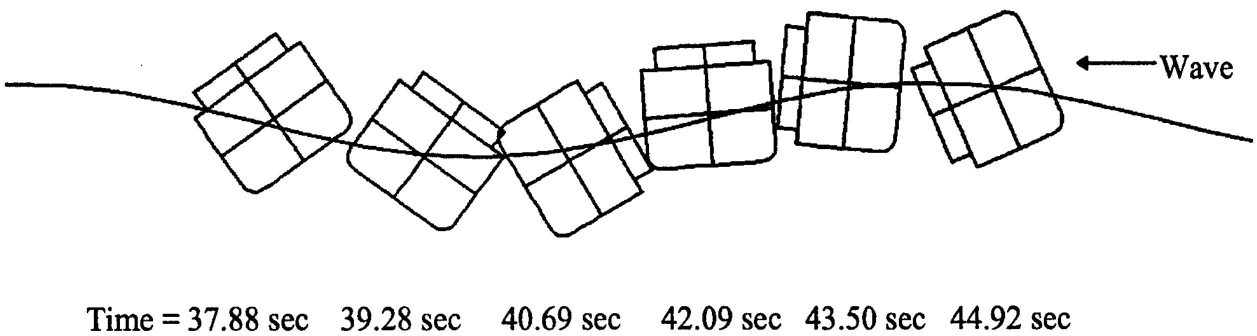


**Figure 15. Time Sequence of Roll Motions of the Series 60, $C_B = 0.7$ Hull in Linear Regular Beam Waves with w = 0.7093 rad/sec, l = 402 ft, and h = 32 ft. GM=2.204 ft. at full load displacement.**

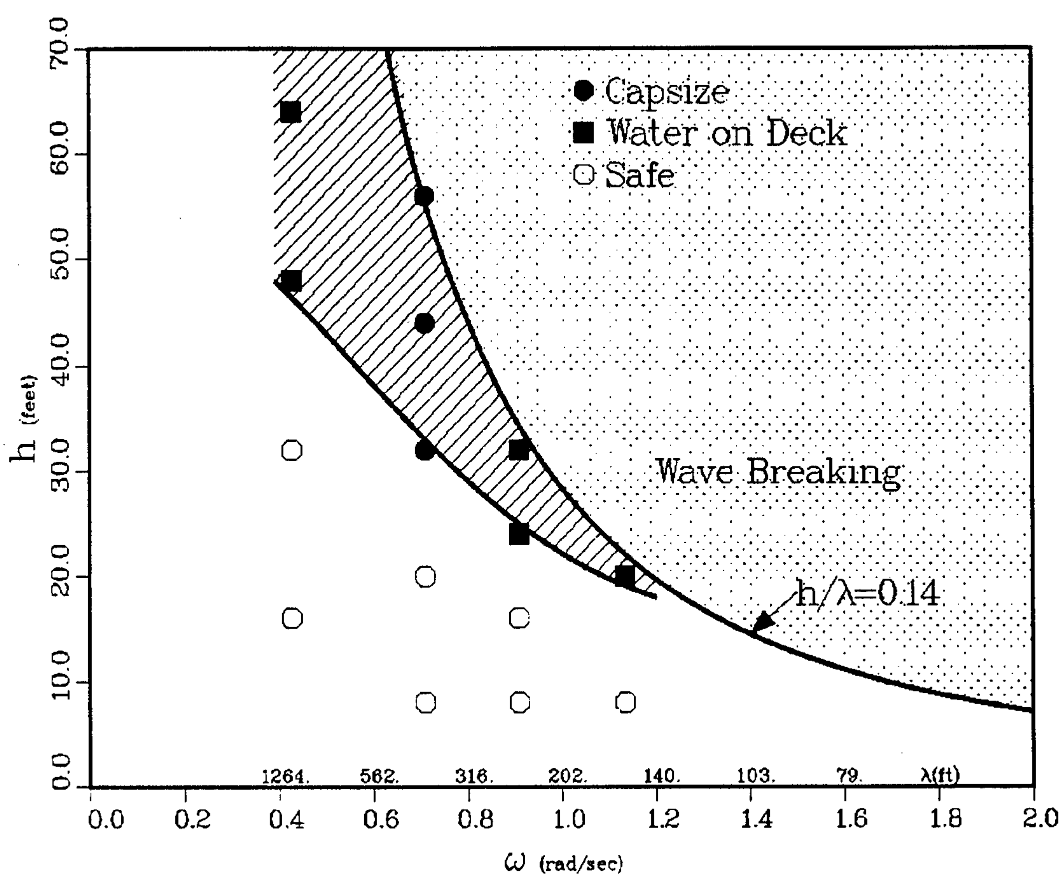


**Figure 16. Capsizing Characteristics for Series 60 $C_B = 0.7$ Hull in Linear Regular Beam Waves**

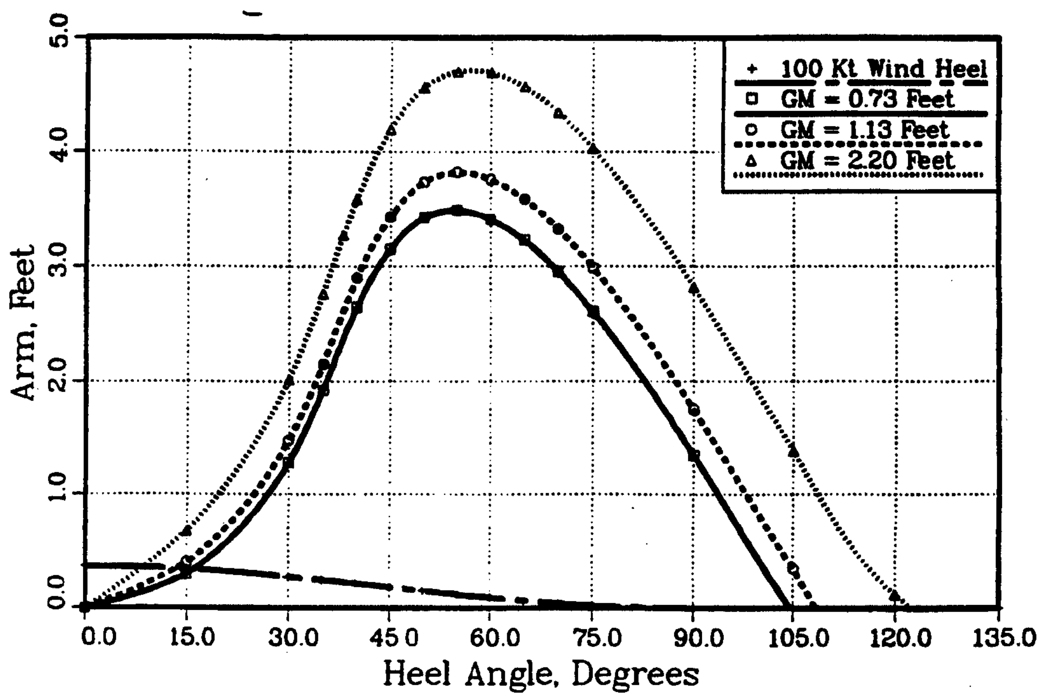


**Figure 17. Righting Arm Curves of the Series 60, CB = 0.7 Hull with Three Different GM's**

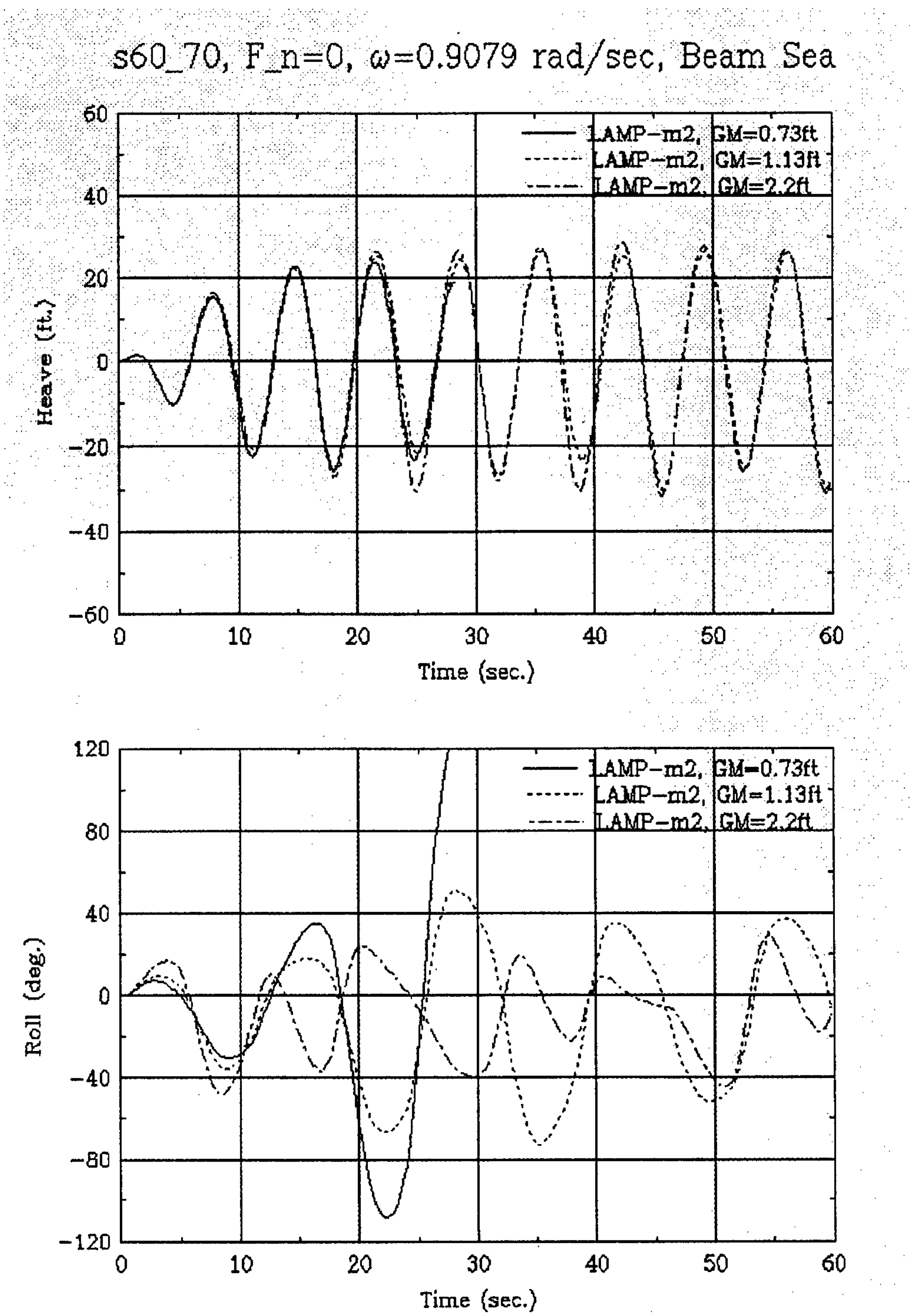


**Figure 18. Time History of Heave and Roll Motions of the Series 60, $C_B$ = 0.7 Hull with Three Different GM's in Linear Regular Beam Waves with w = 0.9097 rad/sec, l = 245 ft., and h = 32 ft. Ballast displacement.**

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# Appendix A

## Glossary of Terms

2-D: Two-dimensional

3-D: Three-dimensional

Body boundary condition: Body velocity is the same as the flow velocity in the normal direction

Body-exact: Body boundary condition is satisfied on the exact position not the linear position.

Free surface conditions: Boundary conditions on the free surface - zero pressure.

Frequency domain: Results obtained are function of frequency.

Transient Free Surface Green's function: Velocity potential of a singularity moving under a free surface.

Linear: Responses will double if input is doubled.

Nonlinear: The ratio of response of input is not a straight line (linear function).

Strip-theory: Hydrodynamic theory based on the solution of 2-D strips (ship cross sections)

Wave energy spectrum: Wave energy defined as a function of frequency.

# Large-Amplitude Motions and Wave Loads for Ship Design

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## ABSTRACT

A new three-dimensional time domain approach for the assessment of the large-amplitude motions and wave loads of a ship in a seaway is presented. In this approach, the body boundary condition is satisfied on the instantaneous wetted surface of the moving body below the *incident* wave surface, while the free surface boundary conditions are linearized about the incident wave.

Results for four ships with different geometry features are presented using the new approach, as well as simplified versions derived from the new approach. The results include linear and nonlinear motion and load responses of ships advancing in regular and irregular seas. The results clearly demonstrate the importance and the magnitude of nonlinear effects in ship motions and wave loads.

The ongoing development of a fully integrated computational system for design assessment of not only the ship motions and wave loads but also the structural responses is discussed. The necessary steps to fulfill the difficult requirements for a practical and complete design support system are also presented.

## INTRODUCTION

The accurate prediction of large amplitude nonlinear wave-induced motions, hydrodynamic loads, and resulting structural responses is of crucial importance in ship design. In addition to compromising efficiency and comfort, severe motions can limit operability and affect safety, while extreme loads may lead to structural failure. Furthermore, the importance of accurate predictions of ship motions, loads, and responses in design and safety assessment is increasing with the advent of new ship types, innovative designs and more demanding operational requirements.

### Naval Ship Needs

The urgent need for an improved motion and load assessment capability within the U.S. Navy is demonstrated by the following examples.

- *Innovative Ship Design* – When the SWATH was first proposed it represented a radically new idea in ship design. At that time its benefits in terms of improved seakeeping were known (Salvesen, 1973). Yet the SWATH has taken more than twenty years to work its way through the design cycle, in large part due to an inability to accurately predict hydrodynamic loads.

- *Extrapolations from Current Designs* – Current ship design is performed primarily through interpolation from known hull forms. Extrapolation beyond the historical database is risky. The design of the catamaran U.S.S. Hayes was an example of an extrapolation from known hulls. She experienced severe damage from slamming in rough seas in the North Sea.

- *Modifications to Existing Ships* – The modifications to the carrier U.S.S. Midway included the addition of blisters to gain buoyancy. This resulted in such a severe degradation in seakeeping that aircraft landings had to be curtailed.

- *Safety* – Although during the serviceable lifetime of a ship only a few encounters with highly nonlinear extreme seas are experienced, it is precisely those encounters that dictate safety margins. Recent structural damage on several ships of the CG47 and CG52 classes have demonstrated the lack of an adequate design capability.

### Commercial Ship Needs

The same need for predictive capability exists outside the naval community.

- *Catastrophic Structural Damages* – Between January 1990 and September 1991 thirty-six bulk carriers suffered severe structural damages causing the loss of twenty-one ships and two hundred and fifty lives (Grove *et al*, 1992). In most of these cases, structural failure due to hydrodynamic loads imposed by the seaway was the primary cause of casualty.
- *Capsizing of Fishing and Pleasure Craft* – In the Fastnet Race of 1979 seventy-seven boats were capsized and fifteen sailors died in what is considered to be the greatest disaster in the history of the sport of yachting. Stephens, *et al*, (1981) pointed out that the current design practice measured stability by static criteria and compensated for dynamic effects through safety margins. Their investigation revealed that because of the fundamental difference between static and dynamic stability "... certain factors which result in favorable static stability characteristics may actually present greater danger when considered in light of a dynamic analysis". The same considerations apply to powered pleasure craft and fishing vessels.
- *Load Line Assessment* – At the January 1993 International Maritime Organization (IMO) Meeting of the Sub-Committee on Stability, Load Lines and Fishing Vessel Safety - 37th Session (SLF37) an entirely new approach for the future Load Line Convention was adopted. Two separate paths will be allowed in determining the load line. Path one utilizes the well established Freeboard Table. Path two is a new Equivalent Level of Safety Assessment using accurate computational methods for predicting ship responses in extreme sea conditions to assess conformance to safety performance criteria.

### System Requirements

As can be seen from the previous examples, there is a pressing need to expand current ship design assessment capabilities for both naval and commercial ships. Additional expertise is needed in a wide range of disciplines - for example, in structures, hydrodynamics, computational methods, and electronic database management. It also seems clear that to be effective in design practice this new capability must evolve into a tightly integrated system so that design assessments and optimizations can include all factors which significantly effect any aspect of the ship's performance.

This paper is organized into two main sections. The first addresses the core requirement for accurate predictions of ship motions and loads in moderate and severe seas. A general overview of computational methods for motion and loads predictions is immediately followed by an introduction to the Large Amplitude Motion Program (LAMP) system of codes which have been developed by the authors for calculations of motion and loads in large amplitude waves. A detailed discussion of both the accuracy and efficiency of the existing codes follows, together with plans for code improvements.

The second section addresses the development of an integrated design assessment system and its use in practical design environments and applications. The first part of this discussion focuses on the progress made thus far in developing the Interactive Design Evaluation and Analysis System (IDEAS) and presents examples of design assessments which are currently performed with it. The second part of the discussion centers on extensions to the system both in terms of added capability for structural responses to impulse loads such as slamming as well as improvements in utility for design. The concluding remarks of the paper provide the broad overview of future requirements for such systems if they are to truly impact the practice of ship design.

## MOTIONS AND LOADS PREDICTIONS

Fortunately, advances in computational ship hydrodynamics over the past decade have resulted in increasingly capable and accurate computer codes for the prediction of ship motions and loads. The application of these codes has been accelerated over the past few years by the ever-increasing power of modern computers, so that some of these advanced numerical calculations may now be done within design time scales. As a result of these advances, a new level of computational capability is now emerging for the prediction of the nonlinear ship motions and wave loads for severe sea conditions.

### Linear Methods

Traditionally, the ship motion problem is formulated in the frequency domain, and lin-

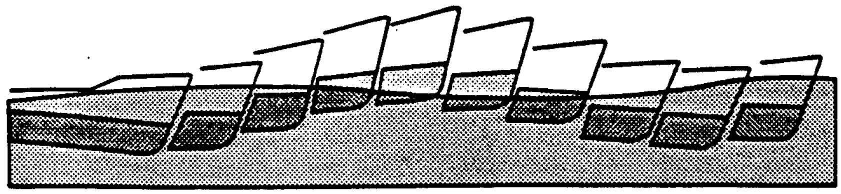


Figure 1: Destroyer Hull in Sinusoidal Wave, $\lambda = 1.20L$ and $H/\lambda = 0.013$. (From Salvesen, 1978)

earized by assuming that the magnitude of the motions and the incident waves are small relative to the draft of the ship. The most commonly used linear tools presently available are based on the strip-theory originated by Korvin-Kroukovsky (1955). These tools were brought to the present state of development by a number of researchers in the United States, Europe and Japan during the mid-1960s. The U.S. Navy standard ship motion program, SMP is a typical code in common use by designers today. The later development of fully three-dimensional linear methods has also resulted in several useful codes. The most promising ones provide solutions using either the transient free-surface Green function (*e.g.*: Beck & Magee (1991), Lin & Yue (1990), Bingham, *et al*, (1993)) or the Rankine source methods (*e.g.*, Nakos & Sclavounos (1990)).

Linear frequency-domain methods have been very successful in many respects; for example, in determining sea state operability limitations for weapon systems on naval vessels (Kennel, *et al*, 1985). Such methods have also been useful in estimating the wave induced loads for large ships (Liu, *et al*, 1992). However, the linearity assumption of small motions relative to the draft is violated by the bow motions of most ships even in moderate head waves. Consider, for example, a typical destroyer hull. The relative bow displacement can be as much as four times the wave amplitude at $Fn = 0.35$ (Frank & Salvesen, 1970). Therefore the bow will exit the surface in moderate waves. Figure 1 illustrates the bow motions for a destroyer hull in sinusoidal wave with $\lambda = 1.20L$ and $H/\lambda = 0.013$.

Since it can be expected that a large percentage of the waves are much steeper than $H/\lambda = 0.013$, the assumption of small displacements at the bow will often be violated. Much steeper waves can occur when waves receive energy from currents or reflections. For example, Smith (1976) points out that off the southeast coast of South Africa the rapid Agulhas Current can result in waves with $H/\lambda = 0.10$. Buckley (1994) has reported that "the most nonlinear waves" in the hurricane Camille wave data had "a height to length ratio of about 1/7" ($H/\lambda = 0.14$). Note that for non-breaking waves the maximum theoretical value of $H/\lambda$ is 0.14.

## Nonlinear Methods

Due to the severe limitations of the linear ship motion theories, several investigators have extended the frequency-domain strip-theory approach to large-amplitude time-domain strip-theory approaches. In these large-amplitude approaches, the nonlinear hydrostatic restoring forces and the Froude Krylov forces are calculated accurately whereas the hydrodynamic restoring and diffraction forces are calculated by some approximate extensions of the strip-theories.

In the United States, such an approach has been applied by de Kat and Paulling (1989) to predict capsizing with quite some success. In particular, for low-frequency following seas their method showed very promising results. Outside the United States, such methods have had noticeable success in calculating the nonlinear global loads (bending moments and shear forces) (see for example, Fujino and Yoon, 1986). Approximate methods of this type can be very useful if they are applied carefully and with full understanding of their limitations.

The more recent research efforts in the United States have been focused on the development of nonlinear methods. These methods may be divided into two categories: fully nonlinear methods and approximate nonlinear methods. Typically, fully nonlinear methods address the exact free surface condition as well as the exact nonlinear body boundary conditions, whereas approximate nonlinear methods apply certain approximations to the nonlinear free surface conditions. Most of the theories in either category are formulated within classical potential flow theory. Good examples of the fully nonlinear approach are the work of Korsmeyer, *et al*, (1992), Maskew (1991), Cao, *et al*, (1992), and Yue (1994). The

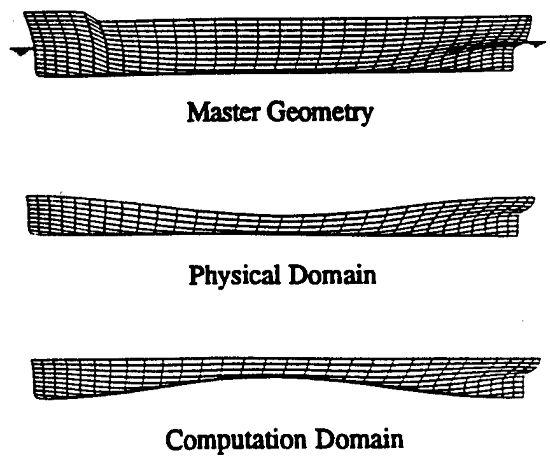



Figure 2: Master Geometry and Panel Distribution in both Physical and Computation Domains.

approximate method of Lin & Yue (1990), and Beck & Magee (1991) solve the body-exact problem in which the free-surface condition is linearized. The work of Pawlowski and Bass (1991) is another example of an approximate nonlinear method. Tulin and Maruo (1992) address the nonlinear deck wetness problem with promising approximate method based on a 2-1/2-D formulation.

It is believed that the approximate nonlinear approaches will result in practical and validated computational tools which can be run on modern advanced workstations within the near future. The fully nonlinear methods will require advanced supercomputers and for the near future will remain research codes serving as validation tools of the more approximate methods.

## LARGE AMPLITUDE METHOD

In 1990, Lin & Yue presented a three-dimensional time-domain method to study large-amplitude motions and loads of floating bodies in waves. In their so-called "body-exact" approach, the free-surface boundary conditions are linearized and the body boundary condition is satisfied exactly on the portion of the instantaneous wetted surface which lies below the undisturbed free surface. The problem is solved using a transient free-surface Green function singularity distribution. The validity and practical utility of this method have been demonstrated by several studies including predictions of large-amplitude motion coefficients, motion history of a ship advancing in an irregular seaway, as well as the effect of bow flare on wave loads (see Lin & Yue 1990, 1992; Lin *et al*, 1991, 1992).

This method was employed for the prediction of motions and loads of a cruiser hull, CG47, in waves (Lin & Meinhold, 1991; Lin & Yue, 1993). The results were satisfactory for moderate seas but difficulties were encountered in severe seas. The difficulties arise from the fact that the body-exact approach models only that portion of the hull below the *undisturbed* free surface. When the wave amplitude is large compared to the ship draft, this representation becomes inadequate, especially near the transom stern.

### New Formulation

In order to improve the Lin & Yue (1990) method and extend its applicability to more severe wave conditions, a new large-amplitude method has been developed in which both the ***body motions*** and the ***incident waves*** can be large (Lin & Yue, 1993). In this new Large-Amplitude Motion Program, LAMP, the body boundary condition is satisfied on the instantaneous wetted surface below the *incident* wave profile with the assumption that the diffracted waves are small compared to the incident wave and that the incident wave slopes are small. At each time step, local incident free surface elevations are used to transform the body geometry into a computational domain with a deformed body and a flat free surface. By linearizing the free surface boundary conditions about this incident wave surface, the problem can be solved in the computational domain using linearized free-surface transient Green functions.

Figure 2 shows a typical master geometry and panel distributions in both physical and computation domains. The solution procedures used for the problem in the computational domain are very similar to those used in the physical domain (Lin & Yue, 1993). Both the source formulation and potential formulation can be used. The two main features of this new large-amplitude approach are: (i) true hydrodynamic effects for the wetted portion of the ship under the incident wave surface; and (ii) automatic inclusion of the correct hydrostatic and Froude-Krylov forces.

In oblique or beam seas, forces due to viscous and lift effects will have a significant effect on the motions and loads. LAMP includes an option to approximate these effects in the time-domain. The viscous and lift effects approximated are as shown in Table 1. For each effect, the table

Table 1: Viscous and Lift Effects

| Effect | Reference | Linearity |
|---|---|---|
| Hull Lift | Low Aspect Ratio Lifting Theory | Linear |
| Skeg, Bilge Keel and Foil Lift | High Aspect Ratio Lifting Theory | Linear |
| Hull Eddymaking | Tanaka (1960) and Ikeda et al. (1978) | Non-Linear |
| Bilge Keel Eddymaking | Kato (1966) | Non-Linear |
| Skeg and Foil Eddymaking | Hoerner (1958) and Ikeda et al. (1978) | Non-Linear |
| Hull Skin Friction | Kato (1958) | Non-Linear |

presents a reference for the calculation method and whether it is a linear or non-linear effect. These components are determined in a manner very similar to that used in the U.S. Navy's SMP code (Meyers *et al*, 1981). However, in the SMP code, the forces are calculated in the frequency domain, assuming certain ***averaged*** magnitudes of roll displacement and roll velocity.

Such an ***averaged*** roll damping approach is not satisfactory for time domain calculations where a primary objective is the accurate calculations of the extreme response events. The new calculation method uses the formulae from the references in Table 1, but uses the current magnitude of roll displacement and roll velocity rather than an averaged value. At every time step, the time history of roll displacement and roll velocity is examined for a peak value, positive or negative. These peak values generate parameters for the viscous forces until a new peak is found. At any given time step, the actual forces depend on these parameters and the instantaneous value of roll displacement and roll velocity.

In the present version of LAMP the incident wave can be represented by a superposition of any number of harmonic wave components at any direction relative to the ships heading. Given a wave spectrum, the program will generate automatically an irregular wave representation with random phases and a pre-specified spreading function. Irregular wave representations for multiple spectra can also be generated. The wave field may also be represented by higher-order Stokes waves.

For any given wave representation, LAMP will calculate the time-domain six-degree-of-freedom coupled motions and the time-domain wave-induced global loads, that is the bending and torsional moments and shear forces at any cross-section along the length of the ship. The program also calculates at each time step the hydrodynamic pressure distribution over the instantaneous wetted hull surface below the incident wave surface. Furthermore, the added resistance in waves as well as the wave resistance can be calculated. Typically the program is run with the ship advancing at a given heading angle and constant forward speed; however, any path and/or speed may be specified.

### The Multi-Level Code System

A complete computational capability for the assessment of ship motions and wave loads must be based on a multi-level approach. Such a system integrates methods which are based not just on one single code or one single level of sophistication, but rather on a system of codes with different levels of sophistication. As a general rule, the physics underlying the ship/wave interactions is best understood using comparisons generated by incremental increases in complexity - a procedure which also moderates computer usage. Analysis tools at the lower levels may employ several approximations to attain a short enough turnaround time for use in early stages of the evaluation process. Examination of results obtained by the lower level code guides the engineer in choosing areas where more accurate theories must be used. In other words, the lower level codes should be used as a filtering mechanism for the selection of more accurate but more complicated and computationally intensive codes.

A multi-level system can also effectively tie the probabilistic and deterministic approaches together providing the missing ingredient of probabilistic prediction. Statistical data of ship motion in given random seas can be obtained by using lower level evaluation codes to efficiently compute the ships responses to a very wide range of deterministic excitations. The severe ship responses can be selected from these, to be examined with the higher level nonlinear simulations. Conversely, nonlinear dynamic simulations of ships in episodic wave events can be used to

Table 2: Computation Methods and Hardware Requirements for the LAMP Code. ($Z = 0$ and $\mathcal{F}(t)$ are Still Water Surface and Incident Wave Surface Respectively)

| Method | Hydrodynamic, Restoring and Froude-Krylov Forces | Hardware |
|---|---|---|
| LAMP-4 | Free Surface Boundary Conditions on $\mathcal{F}(t)$<br>3-D Large-Amplitude Hydrodynamics<br>Nonlinear Restoring and Froude-Krylov Forces | Supercomputer |
| LAMP-3 | Free Surface Boundary Conditions on $\mathcal{F}(t)$<br>2-1/2-D Large-Amplitude Hydrodynamics<br>Nonlinear Restoring and Froude-Krylov Forces | Workstation |
| LAMP-2 | Free Surface Boundary Conditions on $Z = 0$<br>3-D Linear Hydrodynamics<br>Nonlinear Restoring and Froude-Krylov Forces | Workstation |
| LAMP-1 | Free Surface Boundary Conditions on $Z = 0$<br>3-D Linear Hydrodynamics<br>Linear Restoring and Froude-Krylov Forces | Workstation |

understand the actual physical mechanisms underlying the ship responses to these events, such as capsizing, and to identify dominant factors of vessel stability, which can be used in the statistical screening process using the lower level codes.

Recognizing the need for a fully integrated multi-level code system, we have developed the Interactive Design, Evaluation and Analysis System (IDEAS) consisting of a total of five computational methods of different levels of sophistication.

- LAMP-4: The large-amplitude 3-D nonlinear method
- LAMP-3: The large-amplitude 2-1/2-D nonlinear method
- LAMP-2: The approximate large-amplitude 3-D nonlinear method
- LAMP-1: The linearized 3-D time-domain method
- SMP: The U.S. Navy linear strip-theory Ship Motion Program

The total capability is labeled the IDEAS Ship Motion and Wave Load System. The most advanced code is the Large Amplitude Motion Program, LAMP-4 discussed in the previous section. Three simplified versions of the LAMP-4 code have also been developed. The lowest level code uses the linear strip theory.

The LAMP-4 method is the complete large-amplitude method where the 3-D potential is computed with the linearized free-surface condition satisfied on the surface of the incident wave. Both the hydrodynamic and hydrostatic pressure are computed over the instantaneous hull surface below the incident wave surface. Large computer resources are required for this method.

The LAMP-3 method is presently under development as part of the Cooperative Norwegian/USA High-Speed Craft Project. The method, which includes impact forces, is intended originally for planing craft, but is being extended to include displacement hull forms. The hydrodynamic forces are computed by a 2-1/2-D slender-body approximation which includes all of the most important nonlinear hydrodynamic effects for moderate and high-speed displacement hulls.

In the LAMP-2 method, the linear 3-D approach is used to compute the hydrodynamic part of the pressure forces. An option is available to approximate large-amplitude effects by stretching the hydrodynamic pressure. However, the hydrostatic restoring and Froude-Krylov forces are calculated with the same accuracy as in LAMP-4. The reason for developing this simplified method is that it drastically reduces the requirements for computer resources.

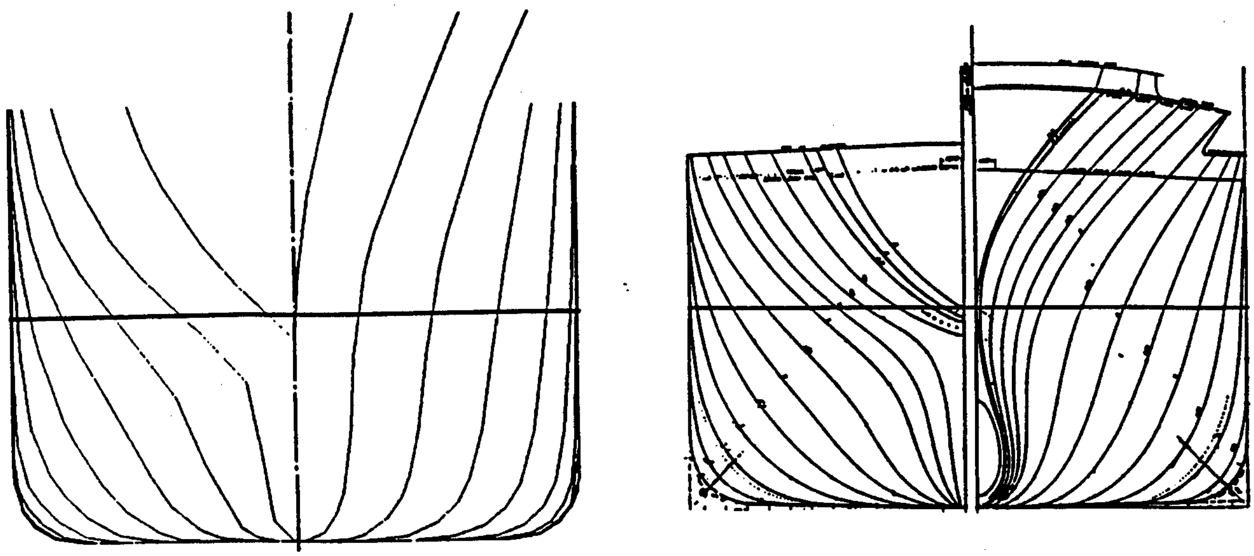
The LAMP-1 method is the linearized version of the LAMP-4 method. This 3-D time-domain method includes a routine for automatic generation of the frequency domain results.

The SMP is the linear strip-theory code presently used by the U.S. Navy. It is based on the theory developed by Salvesen, Tuck and Faltinsen (1970).

Table 2 shows how the hydrostatic restoring and Froude-Krylov forces and the hydrodynamic (added mass, damping and diffraction) forces are calculated for the four different LAMP methods. The hardware requirements for the four

Table 3: General Dimensions

| | LBP | B/L | D/L | $C_B$ | $F_n$ | $R_y/L$ |
|---|---|---|---|---|---|---|
| Series 60 | N/A | 0.143 | 0.057 | 0.700 | 0.200 | 0.25 |
| S175 | 175.0 | 0.146 | 0.054 | 0.572 | 0.275 | 0.24 |



Series 60, $C_B = 0.70$ S175 Containership

Figure 3: Body Plans for Series 60, $C_B = 0.70$ Hull Form and S175 Containership.

methods are also shown in the Table. Note that all of the nonlinear methods, LAMP-2, LAMP-3 and LAMP-4 are based on the approach that both the motions and the waves may have large amplitudes. For all of these three nonlinear methods, the restoring and Froude-Krylov forces are calculated exactly over the instantaneous wetted surface below the incoming wave surface.

## VALIDATION

An extensive validation study of the LAMP code system is presently ongoing. We will here present some sample results for two ship cases in order to demonstrate that the results obtained by the new nonlinear motion and load capability are generally in good agreement with experimental and other theoretical data. The result should also serve to demonstrate the importance of the nonlinear effects. In particular, it is hoped that they will assist in forming a better understanding of the relationship between ship hull geometry and the nonlinearity of the responses.

Result are presented in this section for two hull forms, the Series 60, $C_B = 0.7$ parent hull and the S175 Containership. The general dimensions and body plans for these two ships are given in Table 3 and Figure 3, respectively. The Series 60 hull has mostly wall-sided bow sections with small bow flare and a typical old fashioned cruiser stern. This ship has very small nonlinear geometry features. The S175 Containership has a moderate U/V-shaped bow with considerable flare and a small bulb. The stern is a typical cruiser stern quite similar to the Series 60.

### The Series 60, $C_B = 0.7$ Parent Hull

All of the results presented for the Series 60 hull are for regular head waves at $F_n = 0.20$. Figure 4 shows the comparisons between linear theories (SMP and LAMP-1) and experimental results (Vossers, *et al*, 1961) for pitch and heave displacements and phases. For this particular case a reasonably good agreement between strip theory (SMP) and experimental results was established more than twenty years ago (Frank and Salvesen, 1970). The SMP results are included here so that comparisons with a well established existing design tool can be made. It is seen in

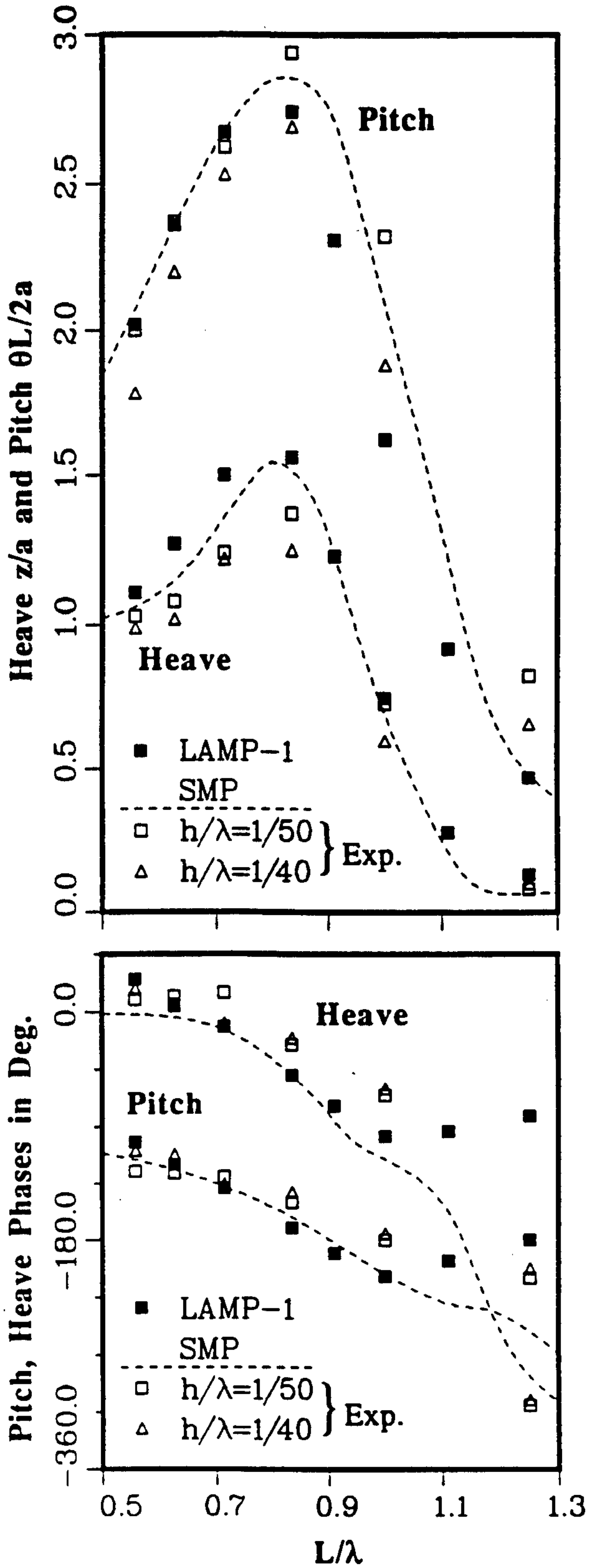



Figure 4: Heave and Pitch Displacements and Phases in Regular Waves for Series 60, $C_B = 0.7$ Hull at $F_n = 0.20$. Comparison of Linear Theories and Experiments.

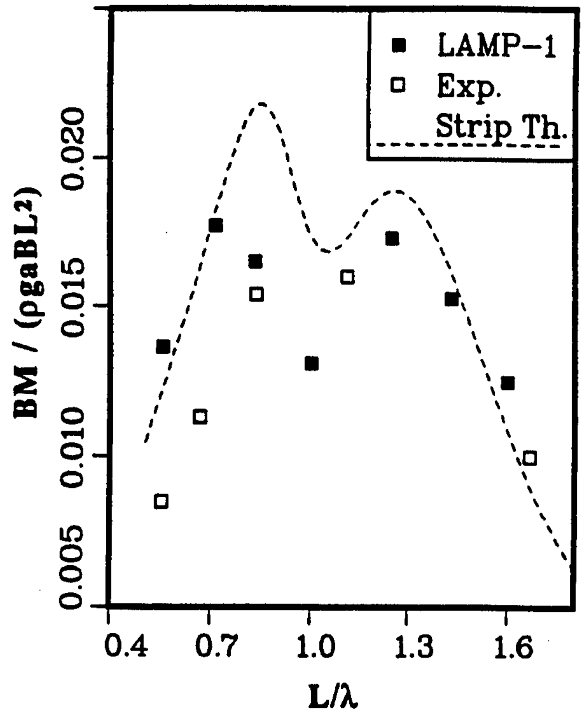



Figure 5: Vertical Midship Bending Moment in Regular Waves for Series 60, $C_B = 0.7$ Hull at $F_n = 0.20$. Comparison of Linear Theories and Experiments.

Figure 4 that the three-dimensional linear theory (LAMP-1) also agrees well with the experiments in general. Other comparisons between LAMP-1 and strip theory have also shown close agreement for slender hull forms at moderate speeds (Lin & Meinhold, 1991).

Figure 5 shows a similar comparison between linear theories and experimental results for the vertical midship bending moments. There are some noticeable differences between the strip-theory results and the LAMP-1 results. However, both predict two peaks which in this case do not seems to be present in the experimental results. Experimental results for other ship cases have demonstrated the existence of such double peaks (Wahab, 1967). The first peak occurs near the frequency where the heave and pitch motions are close to maximum. At this frequency, the bending moments are dominated by inertia loads. The second peak occurs when the motions are very small and therefore seems to be dominated by the hydrodynamic wave excitations.

A sample comparison between bending

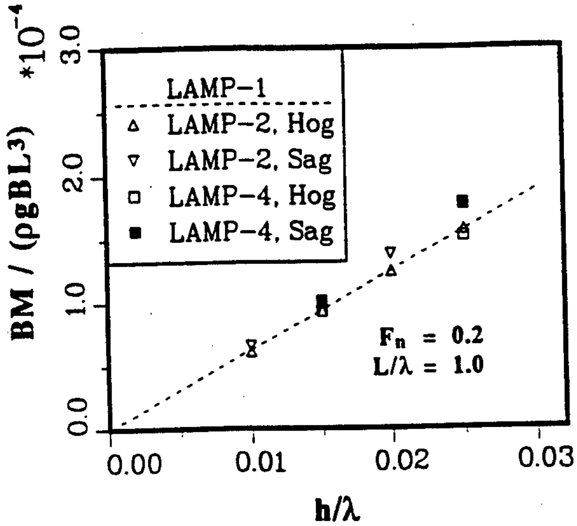



Figure 6: Vertical Midship Bending Moment in Regular Waves with $L/\lambda = 1.00$ as s Function of Wave Height for Series 60, $C_B = 0.7$ Hull at $F_n = 0.20$. Comparison of Linear Theories (LAMP-1) and Nonlinear Theories (LAMP-2 and LAMP-4).

moment results from linear and nonlinear theories are shown in Figure 6 for $\lambda/L = 1.0$. The results show small nonlinear effects as would have been expected for a hull form with relatively small nonlinear geometry features. For the steepest wave case, $h/\lambda = 0.025$, the nonlinear calculations (both LAMP-2 and LAMP-4) show about 14% increase in the sagging bending moment relative to the LAMP-1 calculations. It is seen in the figure that the hogging bending moments predicted by linear and nonlinear theories are in good agreement in this case. It is important to note here that for this ship LAMP-2 and LAMP-4 results are very close.

## The S175 Containership

The S175 is one of the few hull forms for which there exists substantial experimental information about the nonlinear effects. In particular, the experimental data include heave and pitch data in regular head waves with increasing wave steepness (O'Dea, *et al*, 1992). Figure 7 shows comparison between nonlinear calculations (LAMP-2 and LAMP-4) and the experimental heave and pitch data for three wavelengths, $\lambda/L = 1.0, 1.2$, and 1.4. It is encouraging to note that the LAMP-4 results shows very much the same nonlinear trend as found in the experiments. However, the pitch predictions seem to be lower than the experimental values. Similar underpredictions were observed near this wave

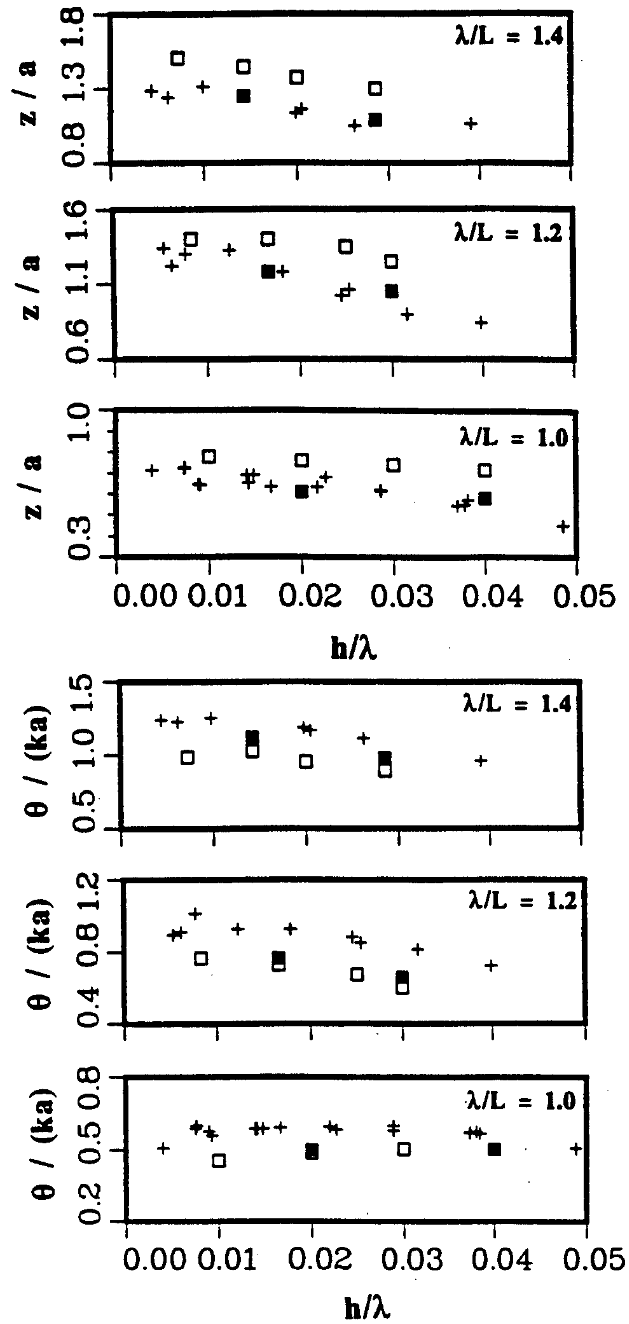



Figure 7: Heave (upper half) and Pitch (lower half) Displacements as a Function of Wave Height for S175 Containership at $F_n = 0.275$ in Three Regular Wave Conditions. Comparison of Nonlinear Theories(LAMP-2. □, and LAMP-4, ■) and Experiments (+).

length range for the Series 60, $C_B = 0.7$ hull form. Further investigation is required on this aspect.

Results predicted by LAMP-2 and LAMP-4 seem to agree well for pitch motion, whereas the

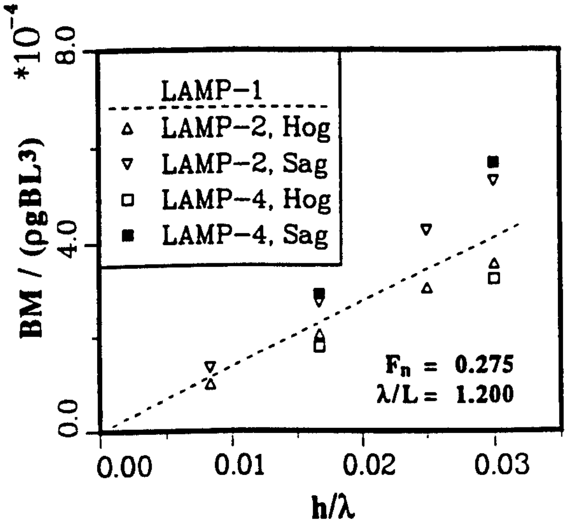



Figure 8: Vertical Midship Bending Moment in Regular Waves with $\lambda/L = 1.20$ as s Function of Wave Height for S175 Containership at $F_n = 0.275$. Comparison of Linear Theories (LAMP-1) and Nonlinear Theories (LAMP-2 and LAMP-4).

heave motions are overpredicted by LAMP-2. A careful assessment has shown that the heave results are sensitive to the implementation of the so called "m-term" effects associated with forward speed (Ogilvie & Tuck, 1969). In LAMP-4, the body boundary condition is satisfied on the instantaneous location of the wetted hull boundary under the incident wave surface. Therefore, the "m-term" effects are automatically and exactly included. In LAMP-2, only simple forward speed terms are included. We intend to introduce an improved m-term approximation method in the LAMP-1 and LAMP-2 code.

Linear and nonlinear vertical midship bending moment results as a function of wave height for $\lambda/L = 1.2$ are presented in Figure 8. The calculations show relatively large nonlinear effects both for the sagging and hogging moments. For the steepest wave case with $h/\lambda = 0.03$, the sagging and hogging moment predicted by LAMP-4 are 35% larger and 21% smaller, respectively, than that predicted by LAMP-1. This clearly shows that the nonlinear wave-load effects are substantial and that they must be included in design assessment. Furthermore, the results in Figure 8 show that the nonlinear effect predicted by LAMP-2 is somewhat smaller than those predicted by LAMP-4, but the trend is very much the same.

**Table 4: CPU Time Requirement for Different Motion and Load Methods on a Workstation and a Supercomputer (LAMP-3 is under development)**

| | IBM RS6000/550 Workstation | CRAY-YMP Supercomputer |
|---|---|---|
| SMP | 2.5 seconds | 0.5 seconds |
| LAMP-1 | 5.0 minutes | 1.0 minutes |
| LAMP-2 | 6.0 minute | 1.2 minute |
| LAMP-3 | - | - |
| LAMP-4 | 4.0 hours | 0.8 hours |

## LAMP EFFICIENCY

Effective use of computer tools such as LAMP code depends a great deal on computation speed. Table 4 shows the CPU time required for a typical one minute real-time ship motion simulation on a high-end workstation and on a supercomputer. With ever increasing simulation capabilities and demands on hydrodynamic codes such as LAMP, the overall efficiency and robustness become crucial factors in the overall success and impact of these codes. Recent research has made appreciable strides in these respects. We highlight two of the more significant developments which can both be incorporated into the LAMP system.

### High-Order Boundary-Element Method

Programs such as LAMP use the traditional constant-panel method (CPM) approximation wherein the boundary geometry is discretized into piecewise linear elements within which singularity strengths are assumed to be constant. CPM is in some sense the simplest boundary-element discretization possible and leads to simplifications in terms of geometry, analyses and code structure. It is now known, however, that CPM is not computationally optimal and often require far too many panels for a given accuracy than is suggested from geometric considerations. Figure 9 shows typical performance of CPM as compared to the quadratic (both geometry and singularity distribution) boundary-element method (QBM) of Xü & Yue (1992). The problem considered is the time-domain calculation of the (linear) impulse-response function of a heaving sphere using transient free-surface Green functions. The number of unknowns, $N$, required for a given ac-

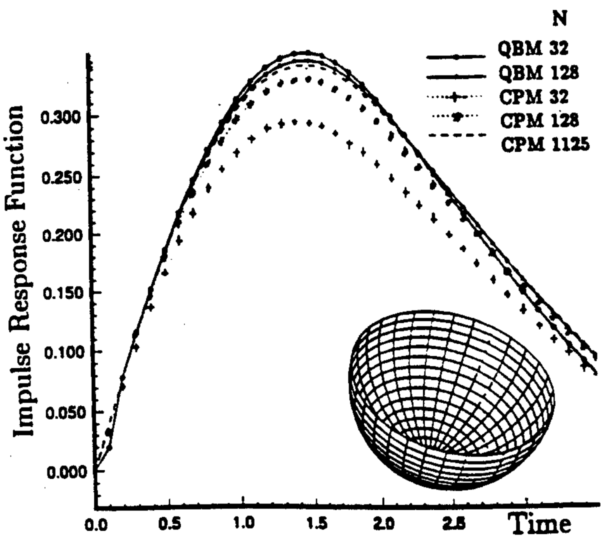
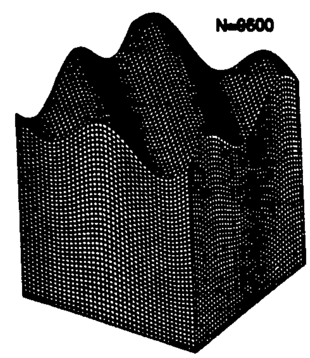



Figure 9: QBM vs. CPM for Computing the Impulse Response Function of a Heaving Hemisphere

Table 5: CPU Time Comparison between an $O(N)$ Scheme and an $O(N^2)$ Method for Tank Sloshing Problem. CPU Times are in Seconds and CPU* is Normalized CPU.




| | O(N) Scheme | | Direct Scheme | |
|---|---|---|---|---|
| $N$ | CPU | (CPU*)/$N$ | CPU | (CPU*)/$N^2$ |
| 600 | 7.23 | 1.0 | 10.8 | 1.0 |
| 1350 | 16.3 | 1.0 | 62.3 | 1.1 |
| 2400 | 31.7 | 1.1 | 321.4 | 1.8 |
| 3750 | 55.4 | 1.2 | - | - |
| 5400 | 71.9 | 1.1 | - | - |
| 7350 | 103.7 | 1.2 | - | - |
| 9600 | 143.8 | 1.2 | - | - |

curacy is $O(10 \sim 20)$ times greater for the CPM. Even factoring in the increased operation count per $N$ for QBM, the overall savings in computational time for comparably optimized code can be a factor of $O(10^2)$ or more, depending on the required accuracy.

Another important and possibly paramount consideration is the loss of robustness of CPM at edges and corners of the boundary, for example near the body waterline. In this case, it is known that CPM may in fact fail to converge in terms of the maximum local error (Xü & Yue 1992). Such non-uniform convergence is eliminated when QBM is used. The incorporation of high-order capabilities such as QBM into LAMP is now under way and is expected to significantly enhance its utility in routine design and analysis simulations.

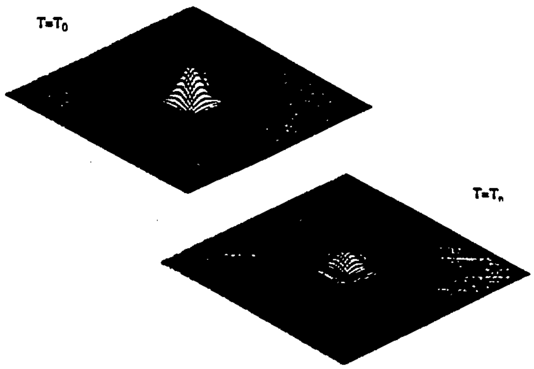
### Fast ($O(N)$) Multipole-Expansion Methods

Even with optimal boundary elements and efficient preconditioning and iterative solution of the resulting equations, the ultimate feasibility of boundary element methods for increasingly larger and more complex problems is limited by the $O(N^2)$ operation count where $N$ is the total number of (spatial) unknowns. Recent development of fast multiple-expansion techniques for boundary-integral methods requiring only $O(N)$ computational effort (see, e.g., Korsmeyer, Yue, Nabors & White 1993) has removed such limitations for all practical purpose. Tables 5 and 6 illustrate the typical efficiency of an $O(N)$ scheme (using CPM) as compared to a direct ($O(N^2)$) method for two 3-D problems – wave sloshing in a tank and the Cauchy-Poisson development of an initial disturbance, respectively. The cross-over value for the number of unknowns is $N_C \sim O(10^2)$. The computational savings (by a factor of $N/N_C$) for large $N$ (say up to $N \sim O(10^4 - 10^5)$) is profound. Another feature of these $O(N)$ schemes is their special suitability for massively-parallel computers. Our recent experience with such a code on the CM-5 suggests that an $O(1)$ time count for moderately large $N$ may be feasible.

## SYSTEM DEVELOPMENT

As discussed in the Introduction, the ship design community is now at the threshold of acquiring a new generation of hydrodynamic design assessment tools of unparalleled accuracy and utility. The IDEAS for ship motions and wave loads is just one example of such emerging capabilities. The motivation for the IDEAS

Table 6: CPU Time Comparison between an $O(N)$ Scheme and an $O(N^2)$ Method for the Cauchy Poisson Problem. CPU Times are in Seconds and CPU* is Normalized CPU.




| $N$ | O(N) Scheme CPU | O(N) Scheme (CPU*)/$N$ | Direct Scheme CPU | Direct Scheme (CPU*)/$N^2$ |
|---|---|---|---|---|
| 400 | 3.2 | 1.0 | 7.3 | 1.0 |
| 3600 | 25.6 | 0.9 | 632.1 | 1.0 |
| 10000 | 79.7 | 1.0 | - | - |
| 14400 | 120.9 | 1.0 | - | - |

concept came from a need for rapid assessment of new designs and design changes aided by analysis of hydrodynamic performance characteristics. Although the computational capability for flow analysis had been markedly improving for decades in both the hydrodynamic and aerodynamic communities, no means existed for a rapid assessment of the effects of configuration changes on mission effectiveness in either area.

The term hydro-numeric design was coined at SAIC to characterize a new discipline which now integrates geometry manipulation, numerical hydrodynamic computation and design performance assessment (Salvesen, *et al*, 1985). The iterative nature of the design process would permit the use of complex, general purpose codes in a systematic manner. The proximate goal in motivating the IDEAS system happened to come from the need to support yacht design for the Americas Cup. However, once the methodology was formalized, it was obvious that the existence of such a system could have a large impact throughout the maritime industry. The design of the 12-meter yacht Stars & Stripes for the 1987 Americas Cup races was a textbook example of the successful use of direct analysis methods coupled to explicit performance criteria, in this case, the probability of winning the Cup (Oliver, *et al*, 1987). The primary lesson learned in that contest was that rapid analysis using state-of-the-art CFD codes was absolutely critical to support design decisions both in the building phase and for last minute design modifications.

The total IDEAS Ship Motion and Wave Load System has been constructed by integrating the hydrodynamic codes with geometry modeling, panelization, visualization and design criteria evaluation codes running on graphics engineering workstations. The primary objective with IDEAS has been to develop a fully integrated *hydro-numeric design* system with sufficient performance, accuracy, and ease of use to impact conceptual and preliminary ship design problems.

The IDEAS Motion and Load System as now configured is shown in Figure 10. This multilevel system allows us to build computational capability progressively. As new computational codes are developed, they can be integrated into the system and validated against existing ones. It must be kept in mind that codes used at each level are limited by different approximations. The confidence level at each level needs to be established through model testing and extensive computations.

The Geometry and Panelization capability within the motion and load system consists of several codes (see Figure 10). Presently the two primary codes are the FASTSHIP geometry generation code by Design Systems & Services, Inc. and the I3G interactive panelization code developed the U.S. Air Force. The two example codes included as part of the Design Evaluation and Assessment capability are the SEP Seakeeping Evaluation Program developed by the U.S. Navy and the ANIMATE code for visualization of time-domain ship motions, including the free-surface elevations.

As seen in Figure 10, the hydrodynamic codes are grouped within three levels of sophistication:

Level I *Linear Methods*
SMP (strip-theory)
LAMP-1 (3-D theory)

Level II *Approximate Large-Ampl. Methods*
LAMP-2 (3-D approximate)
LAMP-3 (2-1/2-D theory)

Level III *Complete 3-D Large-Ampl. Method*
LAMP-4

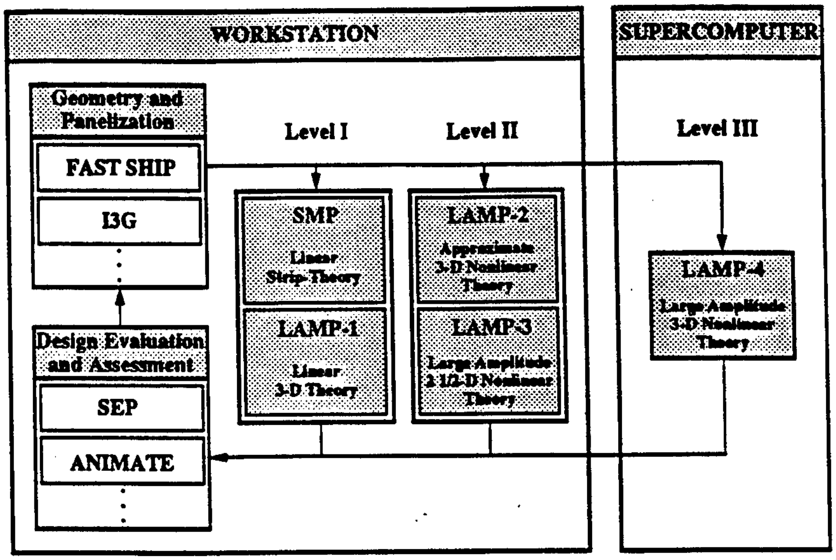



Figure 10: The Present IDEAS Ship Motion and Wave Load System

Consider the application of the Motion and Load System for tanker hull design. In this case, the SMP strip-theory code may be used as a Level I code and the approximate 3-D LAMP-2 code as a Level II code. On the other hand, for higher-speed naval ships where the inclusion of accurate prediction of trim and sinkage is important the 3-D LAMP-1 code which includes the wave-resistance potential would be used as a Level I code. The 2-1/2 D LAMP-3 code, which includes most of the important higher forward speed effects, would be recommended in this case as a Level II code.

Effective use of a computation system such as IDEAS for design depends a great deal on computation speed of different codes and how to use these codes. Table 4 shown previously clearly illustrates how computer resource limitations may affect the use of various methods in the IDEAS system. These numbers also show the need and advantages of using lower level codes as filters to determine the events for which the higher level code can most efficiently be used.

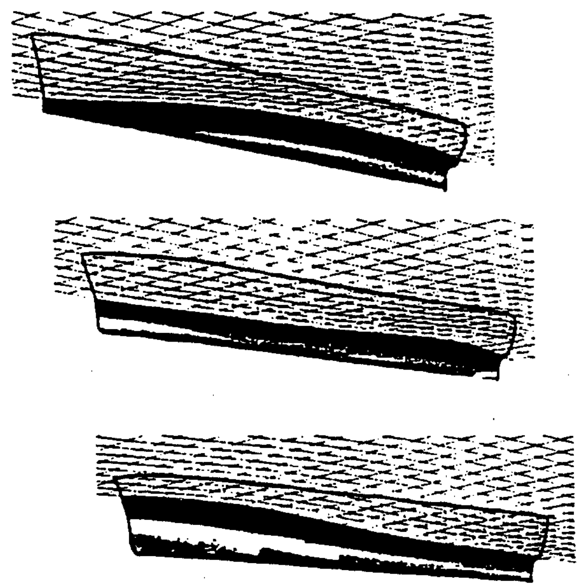


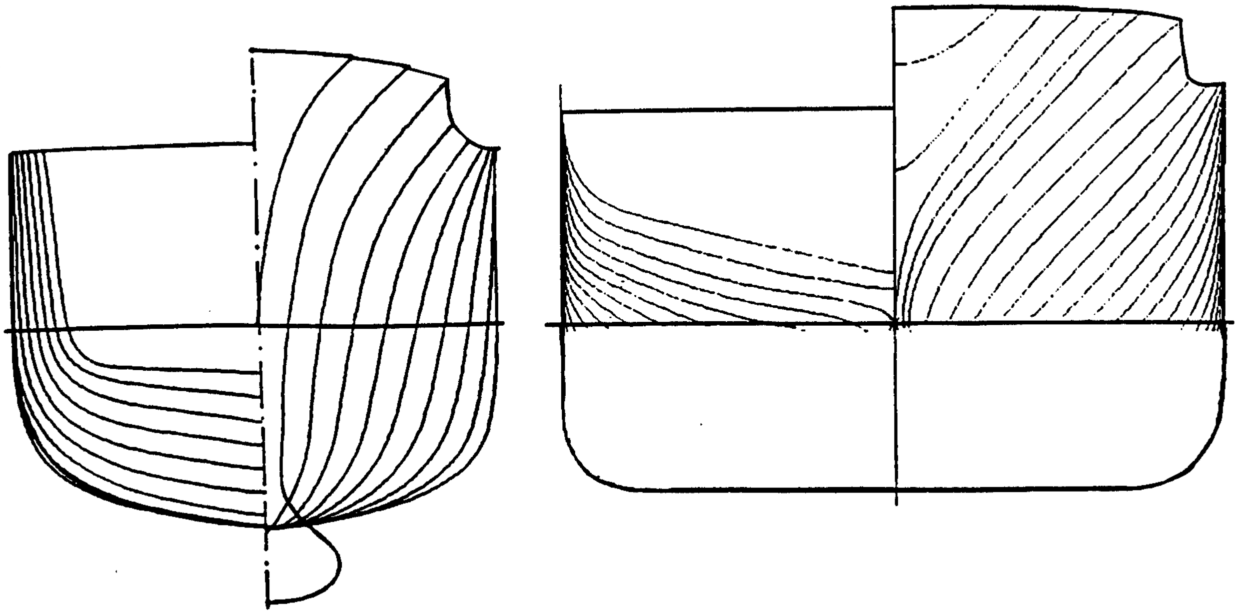
Figure 11: Instantaneous Pressure Distributions for Series 60, $C_B = 0.7$ Hull Form Advancing in Regular Waves

### Coupling to Structural Models

Both the approximate large-amplitude code LAMP-2 and the complete large-amplitude code LAMP-4 calculate the hydrodynamic pressure distributions over the instantaneous wetted hull surface below the incident wave surface. Sample LAMP-2 pressure calculations for the Series 60, $C_B = 0.70$ hull advancing in regular head waves are shown in Figure 11. Interfaces between the hydrodynamic pressure data and three different finite-element codes have been made: the MAESTRO code (in collaboration with its developer Prof. O. Hughes of Va. Tech. (Frankline & Hughes, 1992)), the NASTRAN code (with the American Bureau of Shipping), and the STAGS

Table 7: General Dimensions

| | LBP | B/L | D/L | $C_B$ | $F_n$ | $R_y/L$ |
|---|---|---|---|---|---|---|
| CG47 | 162.3 | 0.103 | 0.042 | 0.510 | 0.260 | 0.25 |
| APL | 260.8 | 0.151 | 0.042 | 0.557 | 0.244 | 0.25 |



CG47 AEGIS Cruiser | APL Containership

Figure 12: Body Plans for CG47 AEGIS Cruiser and APL Containership

code (with the Lockheed Missiles and Space Company, Inc).

The integrated LAMP/STAGS system has been made an integral part of the ARPA Simulation Based Design demonstration system now under development by Lockheed, Newport News Shipbuilding and SAIC. This hydrodynamic and structural code system will serve as a demonstration of the use of multi-level, multi-disciplinary physics-based code systems within the Simulation Based Design approach.

## Design Applications

To illustrate the application of the IDEAS Motion and Load System, two ships were selected. These two ships, CG47 AEGIS cruiser and a APL Containership, are typical modern hull forms in the existing naval and commericial fleets. The general dimensions and body plans for the two ships are given in Table 7 and Figure 12, respectively. It is seen in the figure that the CG47 cruiser has a very fine U-shaped bow with a sonar dome and considerable flare. It has a wide submerged transom stern. The APL Containership is an example of a modern containership. The hull shape below the 10 meter waterline is proprietary and is not shown here. Note that the design waterline is at 11 meters. The bow has large V-shaped flare. The stern has a wide transom which is above the calm water level. This ship has large nonlinear geometry features both at the bow and at the stern which are *not* captured in any linear ship motion theory. Also note that this modern containership has quite different bow and stern shapes than the much older S175 Containership shown in Figure 3.

### *The CG47 AEGIS Cruiser*

We shall briefly discuss an earlier application of the motion and load system to the prediction of the responses of the U.S. Navy AEGIS Cruiser advancing at 10 knots in head seas (Lin and Meinhold, 1991). A thirty-minute linear wave record was generated using a sea spectrum representing Sea State 5.

To run the nonlinear LAMP-4 code for the

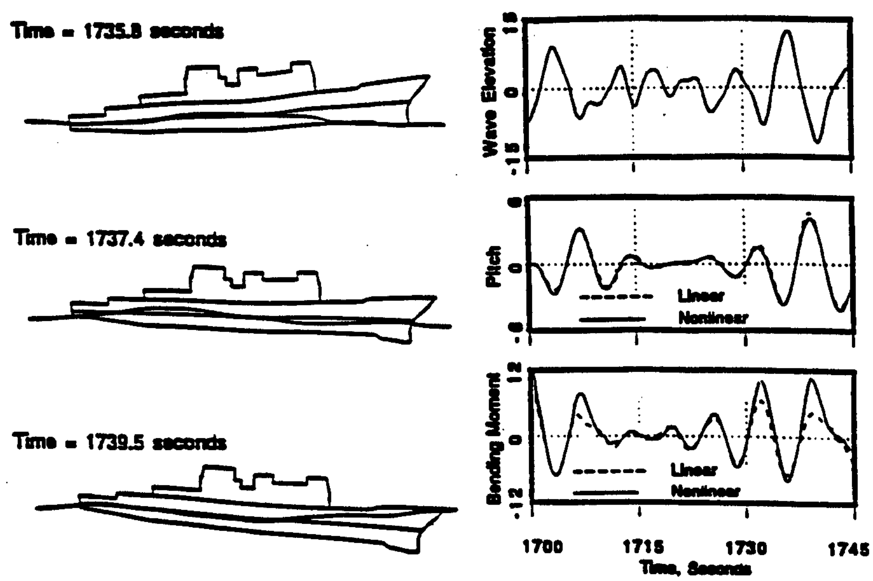



Figure 13: Profile of CG47 at 10 knots in Large-Amplitude Unidirectional Irregular Head Seas and Time Records of Wave Elevations (Ft), and Linear (LAMP-1) and nonlinear (LAMP-4) Predictions of Pitch (Deg.) and Vertical Midship Bending Moments (Ton*Ft*$10^4$)

entire thirty minute wave record was not practical. (Note that the 1991 version of the LAMP-4 code is somewhat different from the LAMP-4 code presented here.) The linear LAMP-1 code was run for the entire wave record to identify three short-term wave events where the midship bending moments were the largest. The nonlinear LAMP-4 code was then used to predict the nonlinear response for these three wave events.

Figure 13 shows the ship advancing in this wave field at three closely spaced time steps as predicted by LAMP-4. Also presented in the figure are the time records of the wave elevations for one of the three wave events as well as the linear and nonlinear pitch and bending moment responses as predicted by LAMP-1 and LAMP-4. The maximum wave height in this wave event is about 25 ft. As shown in the figure, the maximum bending moment predicted by the LAMP-4 code is substantially larger than that predicted by the LAMP-1 code. This difference is believe to be mainly due to the very large bow flare of this hull form which is reflected in the LAMP-4 calculations but cannot be included in the LAMP-1 calculations.

Furthermore, it is seen in Figure 13 that there is practically no difference in the pitch motions predicted by the linear and nonlinear codes. This is important, since it is usually believed that the good agreement between linear-theory heave and pitch motions and experiments is an indica-

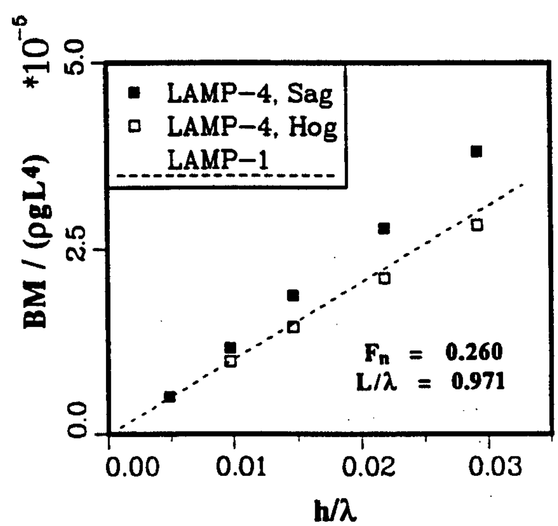



Figure 14: Vertical Midship Bending Moment in Regular Waves with $L/\lambda = 0.971$ as s Function of Wave Height for CG47 at $F_n = 0.260$. Comparison of Linear (LAMP-1) and Nonlinear (LAMP-4) Predictions.

tion that the bending moments are also quite linear. The example above indicates that the bending moment can be very nonlinear even though the motion seems to be linear.

The bending moment for CG47 in regular waves with $L/\lambda = 0.971$ as a function of wave steepness is presented in Figure 14. Both lin-

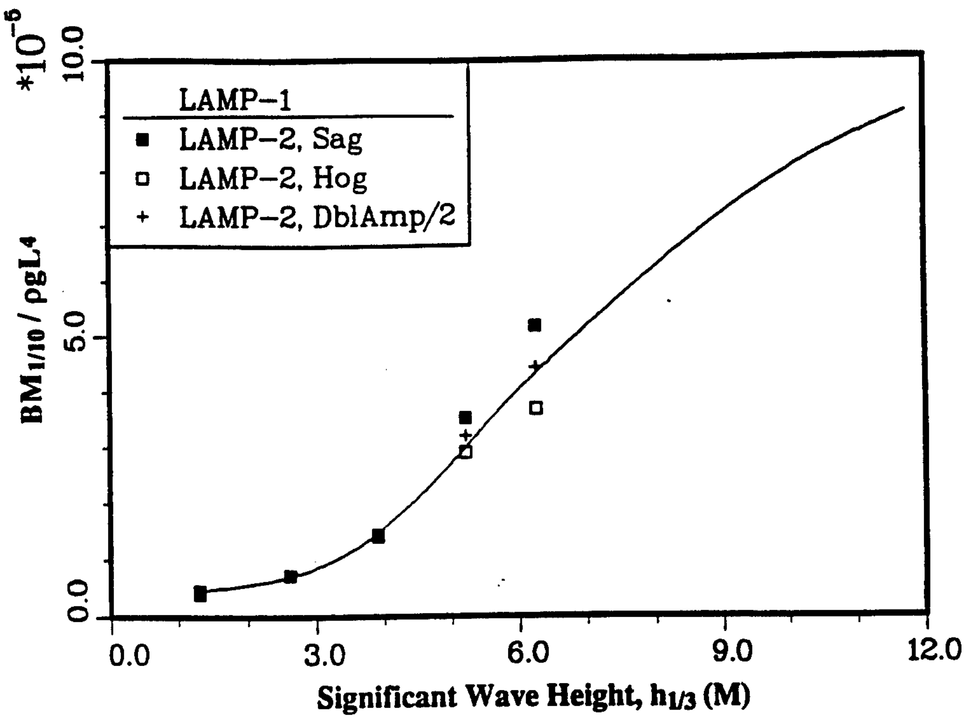



Figure 15: Average One-Tenth Highest Vertical Midship Bending Moments as a Function of Significant Wave Height for APL Containership at $F_n = 0.244$ in Unidirectional Irregular Head Seas. Comparison of Linear Theories (LAMP-1) and Nonlinear Theories (LAMP-2).

ear (LAMP-1) and nonlinear (LAMP-4) results are presented. It is seen that for the steepest wave condition ($h/\lambda = 0.029$), the sagging moment predicted by LAMP-4 is about 28% higher than that predicted by LAMP-1, whereas the nonlinear hogging moment is only about 6% less than the linear prediction. These computations seem to demonstrate that for such naval hull forms, the accuracy of extreme bending moment predictions based on linear superposition methods may be substantially less than required for design applications.

*The APL Containership*

We are presently in the initial phase of an investigation of the nonlinear aspects of the motions and loads for the APL Containership shown in Figure 12. We are primarily interested in an estimate of the magnitude of nonlinear contributions to the bending moment for operations in realistic irregular seas. The averaged one-tenth highest vertical midship bending moment ($BM_{1/10}$) as a function of significant wave height ($h_{1/3}$) are presented in Figure 15. The results are for unidirectional head seas generated from ITTC one-parameter spectrum. For this initial investigation, LAMP-2 has been used for the nonlinear predictions. However, it is recognized that the LAMP-4 code is required for more accurate predictions.

The results presented in Figure 15 show that for significant wave height, $h_{1/3} = 6.26$ meter (corresponding to the high range of Sea State 6), the sagging moment predicted by LAMP-2 is about 20% higher than that predicted by linear theory. The nonlinear hogging moment is about 15% lower than that predicted by linear theory. Also shown as a reference are the "double-amplitude" bending moment divided by 2 as predicted by LAMP-2. It is most convenient in model test to measure the "double amplitude" bending moment values; however, the results presented here clearly demonstrate that the doublamplitude approach has some severe limitations.

The actual time record of the wave elevations and the linear and nonlinear heave, pitch, and vertical midship bending moment predictions are shown in Figure 16, for the $h_{1/3} = 6.26$ meter case. Note that the length of the time record presented in the figure corresponds to 7.5 minutes in full scale. A substantially larger time sequence may be required for a more accurate estimation of the $BM_{1/10}$ values. This again demonstrates the importance of the multi-level approach. LAMP-2 may be used for the long time sequences and then LAMP-4 may only be required for short wave records. This procedure may be used to deter-

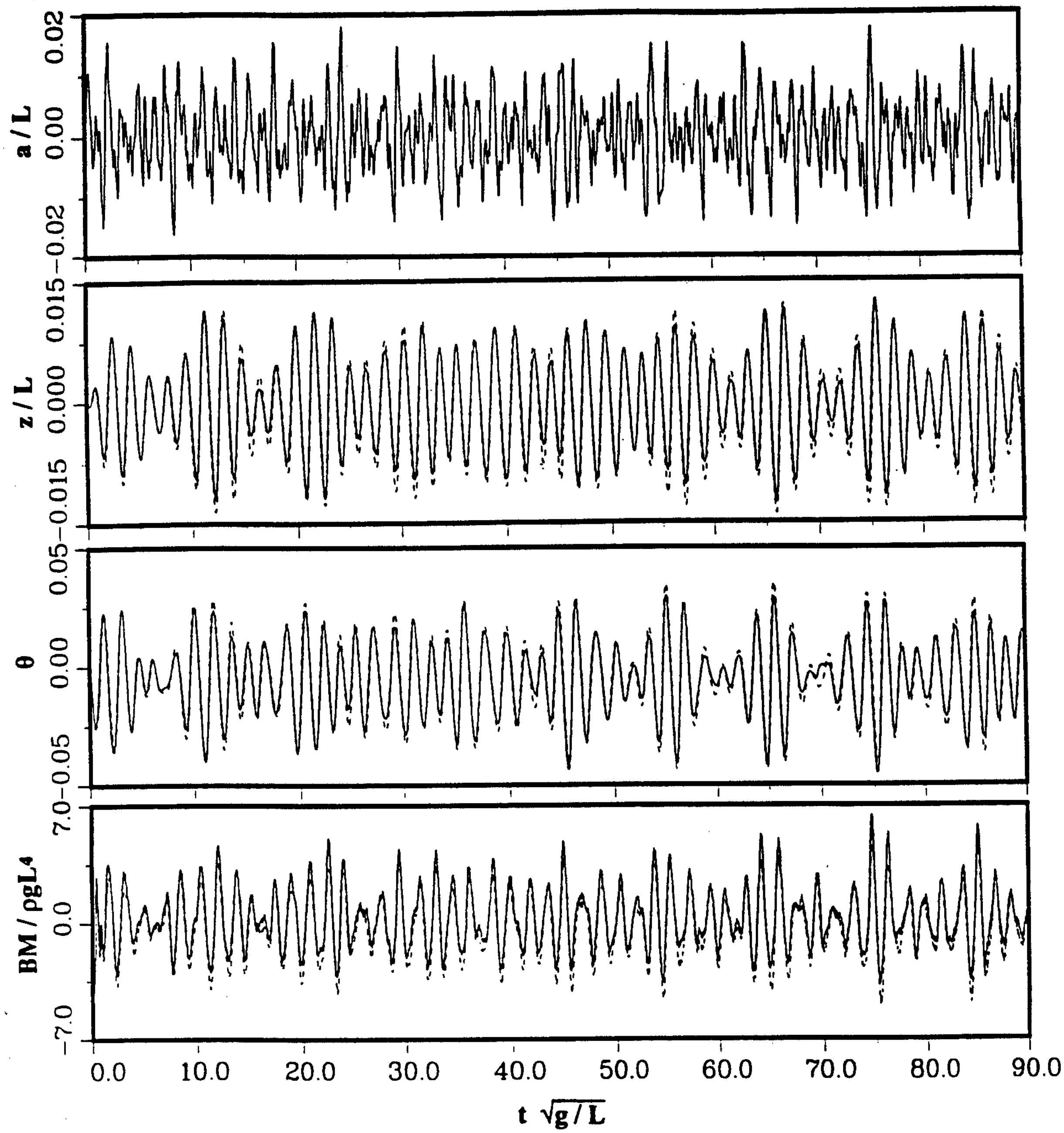



Figure 16: Time Record of wave Elevation and Linear (LAMP-1, - - - -) and Nonlinear (——) Predictions of heave, Pitch, and Bending Moment for APL Containership at $F_n = 0.244$ in Unidirectional Irregular Head Seas with $h_{1/3} = 6.261$ meter.

mine a correction factor which can be used with the statistical values obtained by the LAMP-2 code.

We have barely begun the investigation of the nonlinear aspects of this APL Containership responses. In addition to the wave induced loads, we intend to investigate the occurence of slamming and the nonlinear parametric roll excitation problem. In particular, a better understanding of the parametric roll problem is of critical importance to the shipping companies.

## SYSTEM EXTENSIONS

The results presented here demonstrate that the IDEAS Ship Motion and Wave Load System has the potential to become a new revolu-

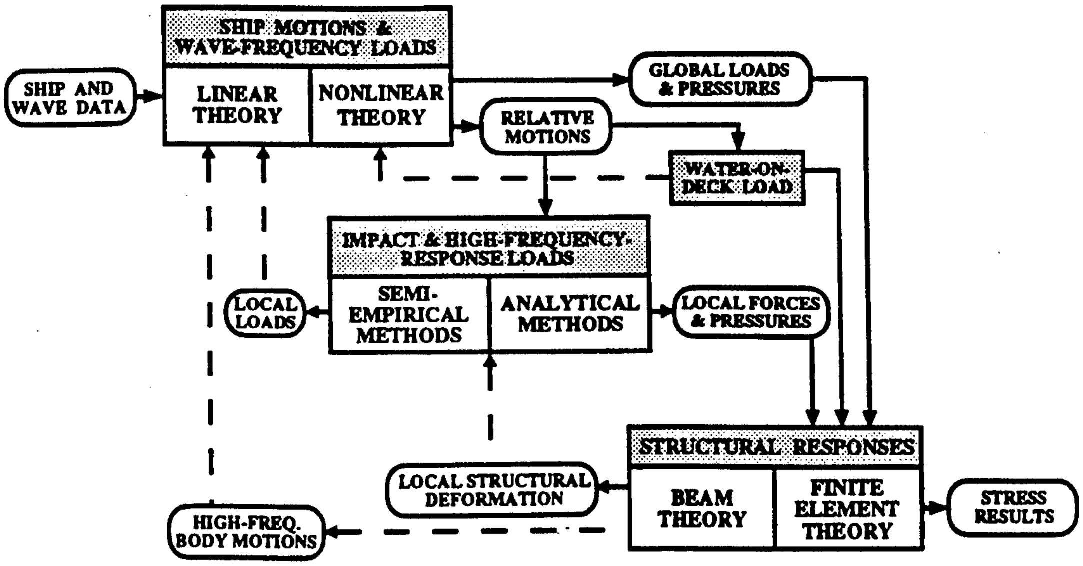



Figure 17: The Complete System for Ship Motion, Wave Load and Structural Response Predictions

tionary tool which will have a major impact on ship design; however, substantial additional work in several important areas is required before the system can meet all the expectations. We shall discuss here some of the most critical areas.

## Wave Impact Loads

The inclusion of wave impact loads or slamming is by far the most important extension. Most traditional methods for analyzing slamming rely on semi-empirical force estimates rather than on accurate prediction of the actual slamming pressure distribution. Furthermore, the traditional methods address only head-sea cases with symmetric slamming. However, the CG47 Cruiser problem and several commercial ship problems have clearly demonstrated that some of the most severe structural failures have been caused by asymmetric slamming loads in oblique seas. It is important, therefore, that any attempt to resolve the total slamming problem must include not only the accurate time-domain simulation of the highly nonlinear motions in oblique seas, but also the prediction of the asymmetric slamming pressures.

The objective will be to develop advanced robust slamming prediction methods and to integrate these methods with both the present hydrodynamics and structural codes to produce a complete capability which includes all of the important components. Figure 17 shows the major components of the total system: (i) ship motions and wave-frequency loads; (ii) impact and high-frequency response loads; and (iii) structural responses. Each of these components will consist of a multi-level code system. Water-on-deck loads are also to be included. The capability will be used as a testbed to determine areas where improved physics modeling is most critically required for improving overall accuracy.

It is expected that the development of a new multi-level computational impact load capability will follow the following steps.

First the experience gained in the cooperative U.S./Norwegian High-Speed Craft project will be incorporated into the LAMP code system. Under the cooperative project, some aspects of the fully-nonlinear slamming load prediction method developed by Zhao and Faltinsen (1992) has been incorporated by Lin (1992) into a new method for motion and load SIMulation of PLANing hulls (SIMPLAN).

A nonlinear 2-D slamming simulation capability which provides the pressure distribution will be further developed into a robust code applicable to general naval and commercial ship shapes including asymmetric cases. This slamming pressure code will then be incorporated into the LAMP system and will become the first complete capability for assessing slamming problems for ships advancing in realistic head and oblique sea conditions. Even though the 2-D approach has its limitations, it is believed that it may be quite accurate for a large class of naval and commercial ship problems and at least far superior to the existing semi-empirical force methods.

The next step will be to integrate into the LAMP system the more advanced 2-D and 3-D slamming pressure methods presently under development. This integration may require further developments to produce robust slamming codes applicable to general ship geometries. Some of the research issues that need to be addressed include the treatment of trapped air, hydroelasticity effects and water compressibility in certain cases. Again the total capability will be used as a testbed to determine the accuracy and applicability of the different slamming methods for ships operating in real sea environments.

We envisage the following multiple-level capability in the near future: (i) simple 2-D phenomenological/empirical models and database for global slamming loads; (ii) extended databases for local pressure distribution for geometrical and operational parameter regimes; (iii) a fully-nonlinear 2-D slamming simulation capability coupled directly into the 2-1/2-D (LAMP-3) and 3-D (LAMP-4) body-nonlinear time-domain computations; (iv) a limited database of 3-D fully-nonlinear slamming simulation for global and local loads; (v) incorporation of fully-nonlinear 3-D capabilities in the LAMP-4 code. This proposed system is based on the following key considerations: (a) the need for a multiple-level capability involving a full range of accuracy/reliability and accompanying computational demands applicable from preliminary to final design and prototyping; (b) the usefulness of simple models for a wide range of applications which however are limited in validity in specific situations; (c) slams are often temporally and spatially very much confined in terms of the entire simulation.

## Other Important Improvements

There are other extensions to the LAMP system which are all important; however, here they will only be addressed briefly.

### *Improved Oblique Seas Calculations*

Accurate oblique- and beam-sea calculations are essential for the prediction of, for example, the torsional moments, slamming, violent quartering sea motions and capsizing. It is believed that the first step is the development of an improved time-domain viscous roll damping approach. The present method relies on mostly 2-D frequency domain empirical data. Our intention is to develop an entirely new method based on unsteady 3-D RANS calculations.

### *Wave Environment Modeling*

The application of the motion and load prediction system to design assessment will require a well defined approach for specifying the wave environment. Different wave-modeling approaches may be used for the estimation of the different responses. For example, in the estimation of comfort level, weapon operability and fatigue loads, the wave environment can in most cases be specified by a sea energy spectrum from which long term time-domain wave events can be generated by assuming linear super-position of the individual wave components. However, predictions of the extreme motion, as for example, capsizing and the most extreme structural loads will require a much better modeling and definition of extreme wave events. Also, we will need a method for estimating the probability of occurrence of the particular wave/vessel encounters which results in the most critical responses.

### *Accuracy and Uncertainty*

The estimation of the accuracy and uncertainty is probably one of the most critical elements in the application of any prediction method in engineering. In particular, in the development of new design approaches which are to be based on advanced physics codes (e.g. Simulation Based Design), a knowledge of the accuracy and uncertainty is absolutely essential. All aspects of accuracy and uncertainty must be accounted for in estimating the total risk associated with building and operating a new construction.

This is a research topic which goes far beyond the more conventional approach to code validation. It will require the development of a new methodology for tracking all of the uncertainties throughout all of the stages in the design. Most importantly, the designer needs to know the sensitivity of the individual errors on the final risk factor for the overall design.

## Fully Nonlinear 3-D Capabilities

Fully-nonlinear 3-D wave-body simulations are, in some sense, the ultimate capability in motion and load predictions in the context of free-surface potential flow. Such capabilities are now becoming available (see Xü & Yue, 1992, Yue 1994) at least for basic research applications. Figure 18 shows a typical simulation of

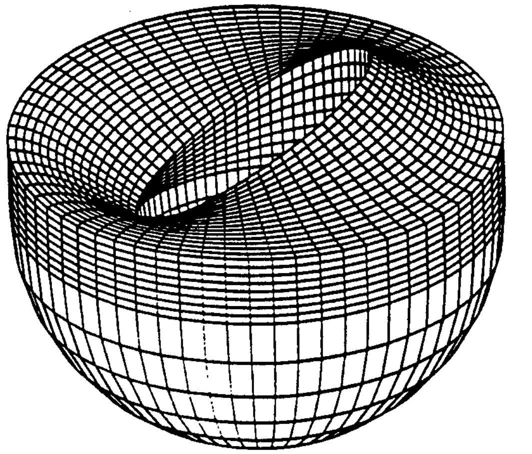


Figure 18: Numerical Simulation of Fully-Nonlinear Wave Diffraction by Floating Body

fully-nonlinear wave interactions with a ship-like floating body. The nonlinear Lagrangian inner computational domain (shown here with $O(2000)$ QBM nodes) is matched to a body-nonlinear outer wavefield via a QBM matching boundary (with $O(1000)$ nodes). Such a matching capability leads to a significant reduction in the total number of surface unknowns. More importantly, since "full" nonlinearity are typically well confined spatially and temporally, a combined approach involving LAMP-like body-nonlinear domains matched to limited dynamic/moving fully-nonlinear regions appears quite feasible. Thus, LAMP is a natural platform for these more advanced 3-D nonlinear tools as they become practically useful.

### System Availability

The effort required to maintain the total Ship Motion and Wave Load and Structural Response System and to serve the entire design and regulatory community is quite similar to that required to operate an experimental seakeeping facility. It is our belief that in order to provide a successful national assessment capability, the final total system must be installed at a national center. A central location for this system is not only of primary interest to the U.S. Navy, but also to the U.S. Coast Guard and ABS as well as the research, the design, and the shipbuilding communities.

Presently, there does not exist a center which has a common focus for academia, industry, and government agencies within the U.S. maritime industry. The U.S. Navy Ship Technology Center located at the David Taylor Model Basin is a start in this direction. This Center was initiated by ARPA, but currently operating under the direction of NAVSEA. The complete Ship Motion and Wave Load System presented in this paper is to be installed at the Tech Center under an ONR contract. It is expected that the Tech Center will soon establish a procedure for serving the entire naval and maritime community.

## CONCLUDING REMARKS

From the discussions presented in this paper it is becoming increasingly clear that the development of a motions, loads, and structural response prediction capability for ships is at a crossroads. On the one hand, hydrodynamics codes are now emerging which are capable of providing an unparalleled level of efficiency and accuracy in calculations for practical design assessment. On the other hand, the integration of these codes into robust, multidisciplinary tools to aid in design decisions is just beginning.

To be most effective, future development of the design tools must be tied to specific design application areas. As in IDEAS, current empirical and computational methods must be available as low level routines which will continue to quickly provide trusted results whenever the required level of accuracy is appropriate. Unless current design expertise is captured in a manner that permits flexible and user-friendly access, there will not exist a matrix of design knowledge within which we can embed the new computational capability. By far the best technique for ensuring that the relevant body of expertise has been included is to develop the system within a design environment, tied to a specific design development.

Once a fairly complete prototype design system exists, the system developers and designers will be in a position to evaluate collaboratively the true impact of additional improvements to the capability. It is anticipated that a design decision support system will permit error estimation and risk propagation as well as cost and performance. This information is exactly what is required to make rational decisions concerning expenditure of resources for additional fidelity, speed, or robustness in the computational codes.

The effort required to develop these systems should not be underestimated. The development of design tools has already begun, but it is still in its infancy. A much higher level of robustness, efficiency, and integration will be necessary to begin to capture existing expertise in a system which can be extended to ongoing research results. The effort in code validation and accuracy estimation alone is daunting. In an era of reduced defense budgets, dissipating design expertise, and increasingly complex design requirements, there may not be a choice. Without such systems, the archiving of naval ship design knowledge and the competitive re-entry of the U.S. into commercial shipbuilding may be extremely difficult.

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