Journal of Hazardous Materials, 25 (1990) 61–73
Elsevier Science Publishers B.V., Amsterdam

DYNAMICS AND STRUCTURAL DAMAGE OF TANKER SHIPS RUNNING AGROUND

G.V. ASADI and H. VAUGHAN
Department of Mechanical Engineering, University of British Columbia (Canada)

SUMMARY

A finite element method is described which models a ship impacting against another object. The ship is modeled as a beam which has the same flexibility, mass distribution and floatation properties as the ship. The ship structure may collapse locally during the ongoing collision process as also may the struck object. A number of collision scenarios involving the collision of a large tanker–ship with a submerged rigid rock are investigated. The significance of ship speed on the extent and nature of the damage to the hull is established.

INTRODUCTION

Grounding incidents, by their very nature, occur in shallow waters, often close to a docking facility and sometimes near populated land masses. The consequences of a large crude–oil spill with its disastrous effects on local ecology and environment have been demonstrated recently on the Alaska coast. The parallel situation of a LNG tanker ship being damaged and releasing upwards of one hundred thousand tonnes of liquid natural gas in the form of an unconfined heavy and toxic flammable gas–cloud has fortunately not occurred. Such ships are usually protected at the sides by collision barriers and from below by a "double bottom" which raises the LNG tanks a meter or so above the bottom plate of the hull.

Engineers have responded to the demands for improved safety against collision particularly in the automotive industry and to a lesser degree in the general transportation industry. The literature on "crashworthiness" is fairly recent and limited although several recent conference proceedings and textbooks have been devoted to it, see for example [1], [2], and [3].

Some revealing statistics on marine damage incidents are given by Kinkead [4]. He gives data on one thousand serious tanker casualties by category of casualty, collected by IMCO (International Maritime Consultative Organization) during the period 1968 to 1980. The data shows that slightly more than four hundred of those incidents were caused by collision or grounding (165 collision, 239 grounding). It is the major hazard in the transportation of oils and gases at sea.

The effectiveness of double bottoms in the prevention of oil outflow from grounded tankers has been demonstrated by Card [5]. A double bottom is a passive system and although effective against shallow penetration of the hull, merely by keeping the cargo above the struck object, it gives a very limited protection against deep penetration, as

0304-3894/90/$03.50

could occur when a ship strikes a sharp reef or ice-flow. Then it is necessary to design a stronger hull based on knowledge of estimated impact forces and using design codes well beyond the elastic limit.

In the last decade considerable work has been done to understand the interactive structural processes involved in ship collisions and grounding with a view towards improved design and increased safety. This work is now briefly reviewed.

REVIEW OF TECHNICAL APPROACHES TO SHIP COLLISION AND GROUNDING DAMAGE

In this section we categorize ship damage and ship collisions into some analytic reference frames which have been found to be appropriate for purposes of developing theoretical models for the study of ship collision damage. Firstly we clarify minor and major collisions.

A minor collision is one in which the ship structure sustains only elastic or small plastic strains to the shell plating or stiffeners. No gross distortion of the structure occurs neither is there any fracture or plate tearing caused by external penetration of the hull. This means that all strength members continue to be effective although possibly at some slightly reduced level. Significant contributions towards understanding the criteria under which minor collision damage occurs and the nature of the damage has been developed by Jones [6]. This work has been particularly useful in the preliminary design of protection barriers which are built into the sides of specialty ships which carry hazardous cargo. Clearly the design criterion is to ensure that if collision occurs then the damage is minor or restricted to the barrier. See for example Jones' paper on collision protection [7].

A major collision is one in which there is some gross structural distortion or tearing of plate leading to significant loss in strength and effectiveness of the damaged parts. Such collisions cannot be analyzed using classical theories of elasticity and plasticity. Instead a method due to Minorsky [8] is commonly used which merely relates the volume of structural material destroyed to the energy absorbed in the collision, the latter being related to the loss in kinetic energy of the ship. A range of coefficients to be used in Minorsky's method are available for many types of ships.

A major collision which cannot be analyzed by Minorsky's method occurs when a ship runs aground. The damage is then often not volumetric but consists of torn plating, perhaps only one meter in width but up to one hundred meters in length. A method for examining such incidents was first proposed by Vaughan [9] and applied by him to some LNG tanker groundings [10], [11], and [12]. Some dimensional effects were determined from a series of small scale tests by Vaughan [13] which allowed his method to be applied to general grounding problems. His experiments were recently repeated and considerably extended by Jones and Jouri [14].

In the late 1970's and early 1980's a considerable effort went into the design of arctic ships in the form of ice breakers, ice-strengthened freighters, and particularly ice-breaking supertankers. At that time the high price of oil made the exploitation of the Arctic financially attractive. A design for a supertanker to withstand ice impact forces was proposed by Johansson [15]. The ship was not built, but an ice-strengthened general purpose ship, the M.V. Arctic, was built a decade ago to carry ore from the Hudson Bay to Northern Europe, which is still in operation. Other designs for ice-breakers were also extremely successful, particularly in the range of 7,000 tonne displacement. All of these ships demonstrate how the designer has progressed, through improved understanding of collision and impact effects, towards safer ships in harsher environments. They have effectively created designs for ships in which collision damage is "minor" instead of "major".

The strongest ship yet proposed for the Canadian Coast Guard is a Class Eight ice-breaker, capable of operating in eight feet of level ice at 3 knots. Several designs have been proposed, one of which has been assessed by Glen and Daley [16]. Estimates of the greatest impact forces and bending moments likely to occur during operation of ships of this type can be obtained using simple formulas derived by Vaughan [17].

FINITE ELEMENT ANALYSIS OF SHIP COLLISION AND GROUNDING

We now describe a finite element analysis recently developed by the authors in which some new damage mechanisms for ships have been investigated [18]. We do not include the mathematical details since they are derived in complete detail in [18]. We merely state the equations of motion that have been solved, then concentrate on the results that have been obtained.

The ship is modelled as a floating beam divided into n stations. The nodal displacement matrix is denoted by [U] and the displacement [u] at any point in the body is then given by

$$[u] = [N]\ [U] \tag{1}$$

where [N] is the interpolation function matrix. The ship is assumed to be axially rigid and its axial motion is considered separately. The degrees of freedom of each element is therefore reduced from twelve to ten, as shown in Fig. 1. The (10 x 10) mass matrix of the element is $[m] = [m_1] + [m_2]$, where $[m_1]$ is due to lateral inertia and $[m_2]$ is due to rotary inertia.

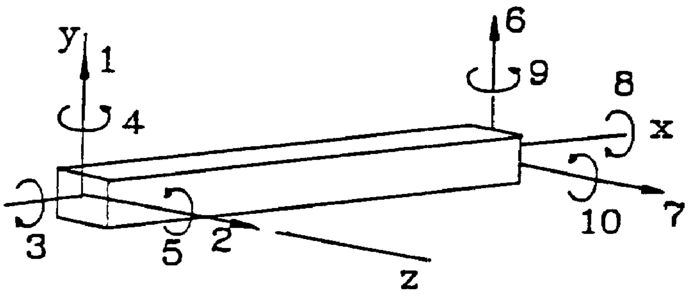

Fig. 1. The beam element with ten degrees of freedom.

In the analysis it is mathematically convenient to represent rigid-body motions as special cases of the dynamic flexural motions. Thus the natural frequency matrix includes as its leading elements the rigid-body periodic motions in heave, pitch and roll, respectively. This was made possible by treating each beam element as a free-free beam which consequently admits rigid-body displacements as well as elastic vibrational motions.

The simultaneous treatment of rigid-body motions and flexural motions enables added mass to be included in [m] in the form of a frequency dependent matrix. The three leading elements of the added mass matrix correspond again to heave, pitch and roll whereas the later elements have to be identified with the appropriate mode of vibration, either a vertical, lateral, or torsional mode, starting with the fundamental modes and working up to the desired number of higher harmonics. A particular example is given later in this paper. Added mass coefficients for transient impulse motions in shallow water are difficult to assess so in the numerical calculations discussed later, deep water values for sinusoidal motions have been used

The stiffness matrix of the element is denoted by [k]. It is composed of three parts: (i) the elastic structural stiffness of the hull, (ii) the hydrodynamic stiffness due to buoyancy, (iii) the hydrodynamic stiffness due to roll. When evaluating the stiffness associated with (ii) and with (iii) the effects of flexural vibration and torsion have been included respectively, as well as the rigid body motions in heave, pitch and roll.

The ship, and hence each element, is influenced by the following forces: weight, buoyancy, restoring buoyancy torque, impact or contact forces. These forces are now briefly discussed. The weight distribution of the ship is obtained from the standard curve in the form of tonne per meter. The weight of each element is thus estimated and the nodal values obtained.

The buoyance force on each element is obtained from the hydrostatic curves of the ship. It is assumed that the immersed cross-sectional area is constant along each element (different constants for each element). The buoyancy force is then adjusted to include vertical motions of the element relative to the water surface - this merely involves adding the relative displacement of the ship onto the static draft. The restoring torque caused by heeling is calculated for each element using the small angle formula with the metacentric height. The angle of heel for each element is easily deduced from the nodal rotations U_3 and U_8 shown in Fig. 1.

The impact forces are treated as surface tractions and may be either distributed or concentrated. They are transferred to the nodal points using the interpolation matrix.

The equation of motion of each element may be expressed in the form

$$[m]\,[\ddot{U}] + [c]\,[\dot{U}] + [k]\,[U] = [q] \tag{2}$$

where [c] is the damping matrix and [q] is the generalized force matrix.

Physical characteristics of each station of the ship are taken as input data. These data are: cross-sectional area; breadth, depth, two principal inertias, mass and length of each element. Mass and stiffness matrices for each element are generated and external forces (excluding contact forces) are included. The global equation of motion for the ship is then obtained with respect to a fixed external frame. Transference between local and global frames is accomplished by a series of transfer matrices. Assemblage is based on the requirement that the displacement at a node shared by two elements must be the same for both of those elements.

The global matrices of the ship are of the form

$$
\begin{aligned}
[M] &= \sum_{s=1}^{n} [T]'_s \, [m]_s \, [T]_s \\
[K] &= \sum_{s=1}^{n} [T]'_s \, [k]_s \, [T]_s \\
[C] &= \sum_{s=1}^{N} [T]'_s \, [c]_s \, [T]_s
\end{aligned}
\tag{3}
$$

where $[T]_s$ is the transfer matrix for element s, $[m]_s$, $[k]_s$ and $[c]_s$ are the mass, stiffness and damping matrices for element 's'. The global equation of motion for the ship is then

$$[M]\,[\ddot{U}] + [C]\,[\dot{U}] + [K]\,[U] = [Q] \tag{4}$$

where $[Q] = \sum_{s=1}^{n} [T]'_s \, [q]_s$

When assembling the global equation of motion (4) there is a matrix overlap of size 5 x 5 between adjacent elements due to the assemblage requirement on the displacements at the common node. Consequently the matrices in (4) are of size 5(n+1), where n is the number of elements. Clearly [M], [C] and [K] are square, [U] and [Q] are columnar.

Equations (4) are a set of simultaneous coupled equations. They are uncoupled in the usual way by determining the eigenvalues, constructing the modal matrix, and then transforming to principal coordinates. This necessitates the assumption that the damping matrix is a linear combination of the mass and stiffness matrices, usually referred to as proportional damping. Accordingly we write

$$[C] = \alpha[M] + \beta[K] \tag{5}$$

The main cause of damping is due to hydrodynamic dissipation of energy. Damping coefficients α and β can then be found from the logarithmic decrement of the ship vibrational response records in heave and pitch. Such records are available but in limited supply.

The uncoupled equations are integrated incrementally using a second order finite difference method. The grounding and collision forces are interactive and are dealt with in the following way. The position of initial contact must be specified, involving one or more nodal positions. The nature of the contact force relative to ship position must be specified, either explicitly or implicitly. Examples considered are : the ship and object are rigid; the ship is rigid but the object crushes with a specified constitutive behaviour; the object is rigid but the ship becomes crushed or torn. In the latter case, certain strength coefficients of the Minorsky type involving crushing strength both laterally and longitudinally must be used together with tearing strengths obtained from [13]. During the integration process, the position of the ship must be checked and the contact force updated to its new value. In the case of separation or rebound the force must be cancelled. The extent of the contact force is allowed to move as the motion proceeds so that new nodal positions may become involved as time increases.

Some typical grounding scenarios which have been examined using this method are now presented.

SIMULATION RESULTS OF TANKER SHIP GROUNDINGS

We consider a very large tanker ship of length 330 meters and displacement 260 thousand tonne. The mass and buoyancy curves are shown in Fig. 2. Other required geometry is given in Table 1. This ship is referred to as the 'Standard Tanker Ship' about which various parameters are changed. The yield strength of the hull is taken as 280 MPa. Added mass coefficients in heave, sway and surge are taken as 0.75, 0.6 and 0.1 respectively. The natural frequencies (first six) were calculated and are identified in Table 2.

The ship is assumed to be running ahead when it strikes a smooth rock, 15 meters below water level, 17 meters to the port side. It is assumed that the smooth rock does not tear the hull plating, but volumetric damage of the Minorsky type does occur.

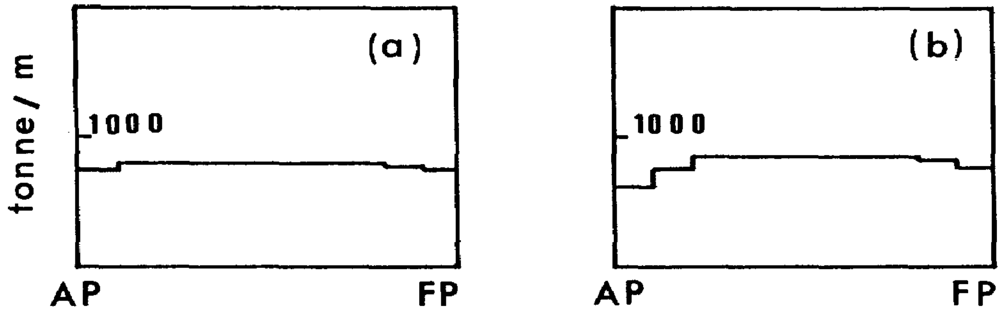

Fig. 2. (a) Massdistribution and (b) Buoyancy distribution for standard ship

TABLE 1 – Specification of the 'Standard Tanker Ship' at the Stations

STATION No.	STATION LENGTH meter	STATION BREADTH meter	STATION DRAFT meter	STATION SHAPE FCT.	MOMENT OF INERTIA I_Y m^4	I_z m^4
1	33	32	22	5	500	360
2	33	38	22	7	600	400
3	33	42	22	8	800	600
4	33	42	22	8	800	600
5	33	42	22	8	800	600
6	33	42	22	8	800	600
7	33	42	22	8	800	600
8	33	42	22	8	800	600
9	33	41	22	7	600	400
10	33	40	22	5	600	400

TABLE 2 – Natural Frequencies of the Ship

Mode Number	Frequency (Hz)	Mode Shape
1	0.154	Heave
2	0.186	Pitch
3	0.426	Roll
4	1.638	1st Vertical
5	1.751	1st Lateral
6	4.307	1st torsional

Fig. 3 shows the extent of damage sustained by the hull for forward velocities of 1 m/sec, 3 m/sec, 5 m/sec and 7 m/sec. For the low speeds the hull receives a damaging impulse near the bow which changes the angular momentum of the ship. It then clears the rock but the combined forward motion and angular motions cause a secondary collision near midships which bring it to rest. For high speeds the damage estimates are rather catastrophic and separation from the rock does not occur. The figures clearly show the importance of collision speed on the nature and extent of the damage. Collision force and contact times have also been determined and are shown in Fig. 4 for the higher speeds. It is evident that the initial kinetic energy of such a large ship can be absorbed only by inflicting considerable structural damage to the ship, a process which can last for up to one minute and damage half the length of the ship.

The effect of increased structural strength has been examined. Fig. 5 shows the damage estimates for an identical scenario except that the ship structural steel has been doubled. The reduced damage is evident in each case.

Next we consider what happens when the ship plating is ruptured, as is likely if the ship runs over a sharp rock. The damage inflicted to the standard tanker for the range of

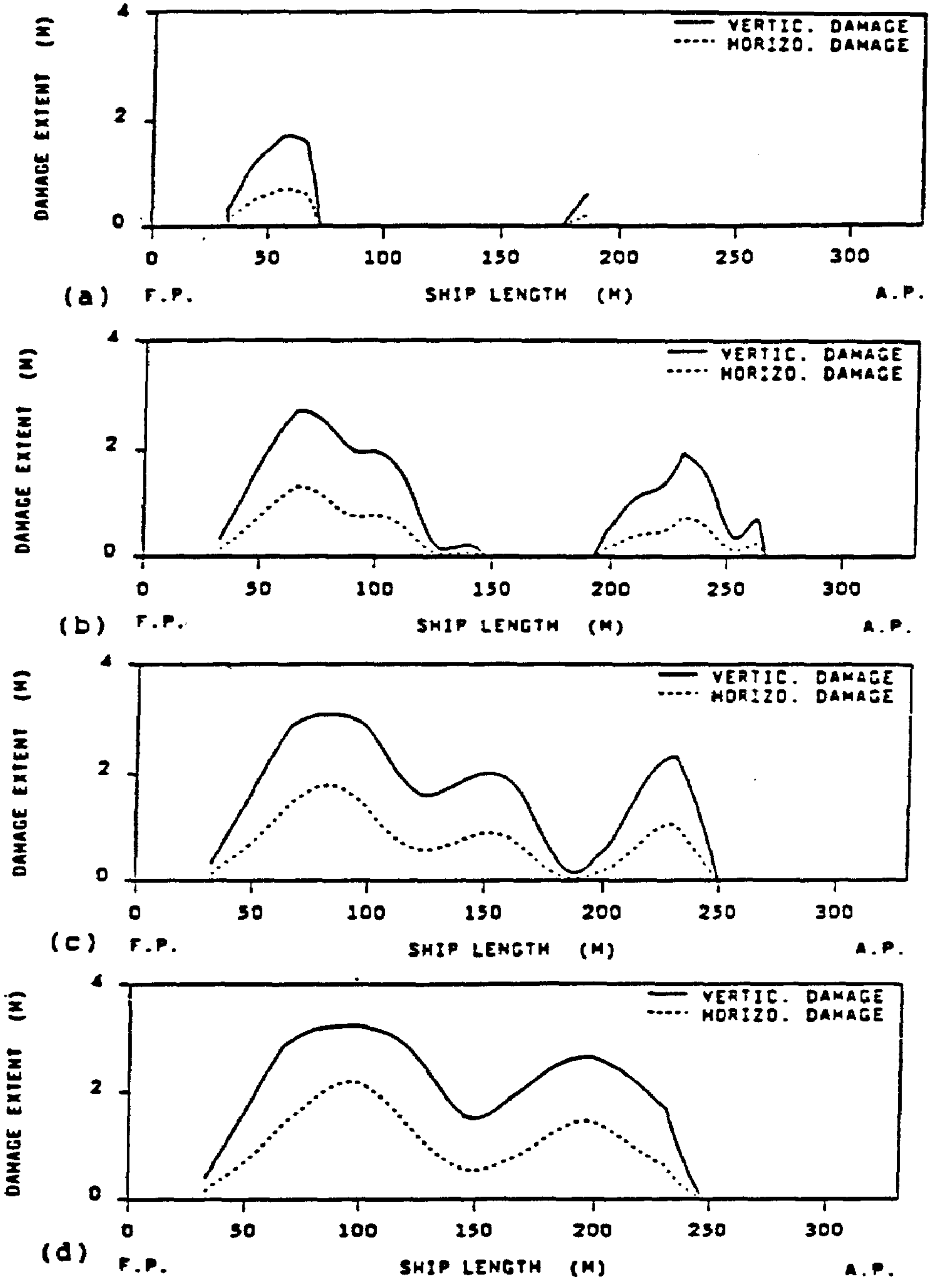

Fig. 3. Extent of damage to standard hull without rupture
(a) V = 1m/sec, (b) v=3 m/sec, (c) v = 5 m/sec, (d) v = 7 m/sec.

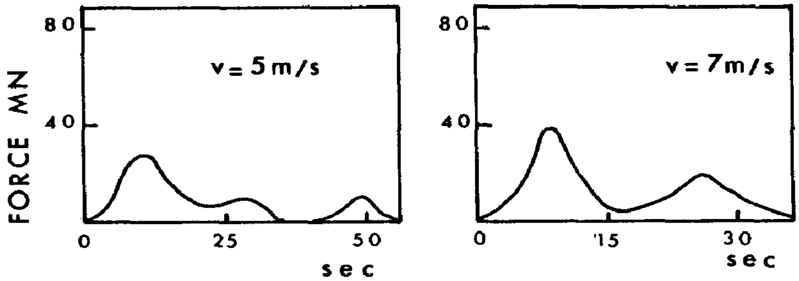

Fig. 4. Collision force on the standard hull versus time.

velocities considered previously, is shown in Fig. 6. It exceeds that shown in Fig. 4 for the simple reason that a torn plate absorbs less energy than a dented one, primarily because the torn plate has little membrane strength.

Finally we illustrate what happens to a ship when it grounds on a smooth rock during maneuvering. We take a forward speed of 2 m/sec and fore and aft rotational speeds of 2 m/sec and 0.5 m/sec. The rock is 15 meters below the surface, extends over 20 meters of the hull, with centre of collision 50 meters from the bow. The force and damage calculations are shown in Fig. 7. As a results of the initial impact force, the ship moves away from the rock and the damage depth subsequently decreases with time. This example has been included to show that the interaction and response due to a rotational collision is fundamentally different from a normal head-on collision.

The number of variables and scenarios which may be examined are obviously very great. In this paper we have merely demonstrated that a working model has been developed and that it can be used to investigate many real ship-grounding situations. A more comprehensive investigation is included in the thesis [18] but ongoing case studies are also being made.

CONCLUDING REMARKS

The consequences of what happens when a tanker ship runs aground and spills oil or releases liquid natural gas is a subject of much current interest. By making mathematical models of a grounding ship and determining the interactive forces that the ship receives, design criteria can be established and subsequently incorporated. By such processes safety standards can be increased and ships that might otherwise have been breached can be shown to be structurally sound under specified operating conditions.

A finite element model developed at the University of British Columbia which models ship grounding incidents has been described and a number of realistic situations have been illustrated. Such illustrations clarify the complex interaction which exists between ship motion and grounding force. This clarification can assist in the improved structural design of LNG and tanker ships so that spill incidents will be less likely to occur, even if grounding does occur.

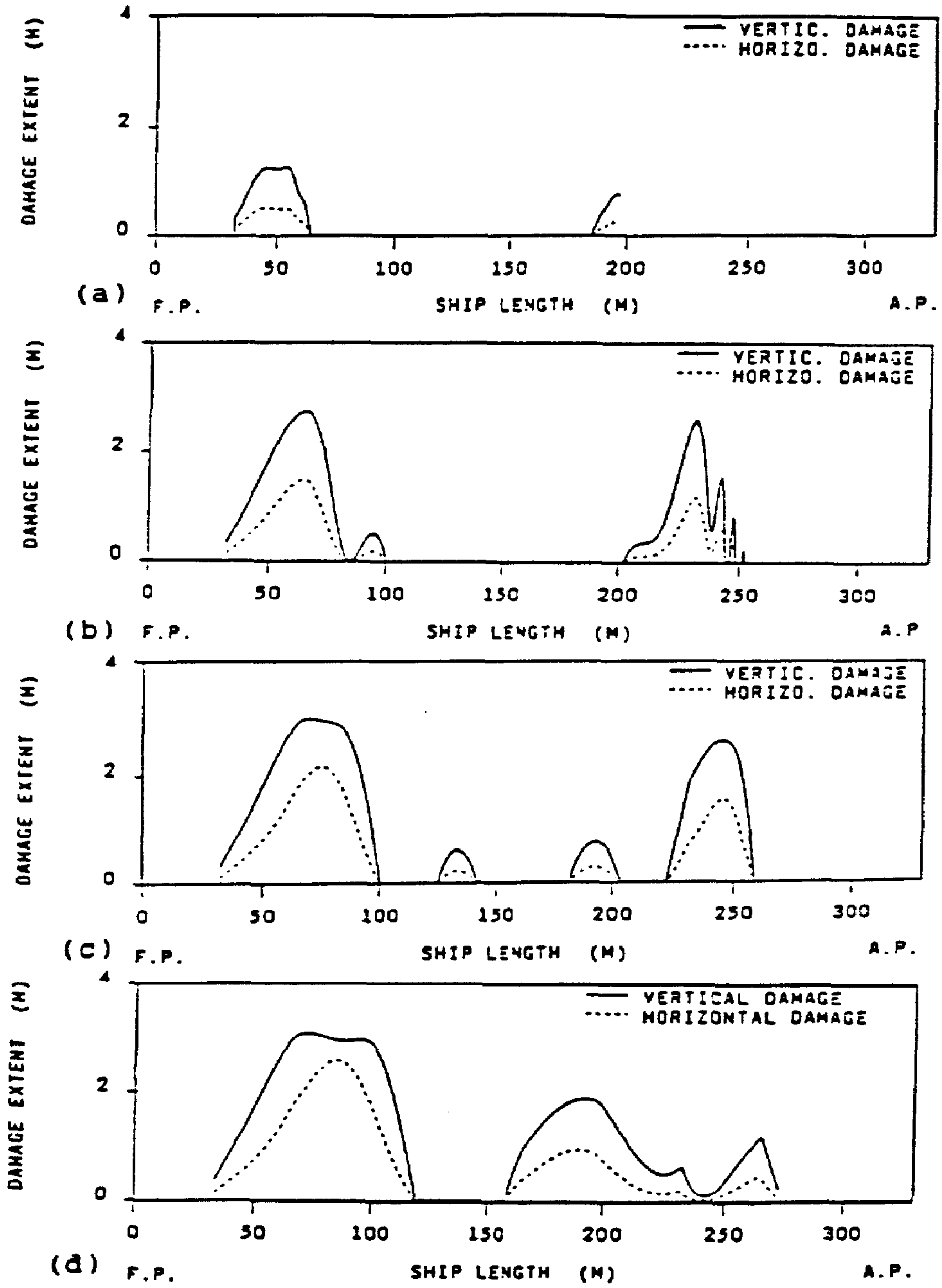

Fig. 5. Extent of damage due to forward collision without rupture of the reinforced hull (a) v = 1 m/sec, (b) v = 3 m/sec, (c) v = 5 m/sec, (d) v = 7 m/sec.

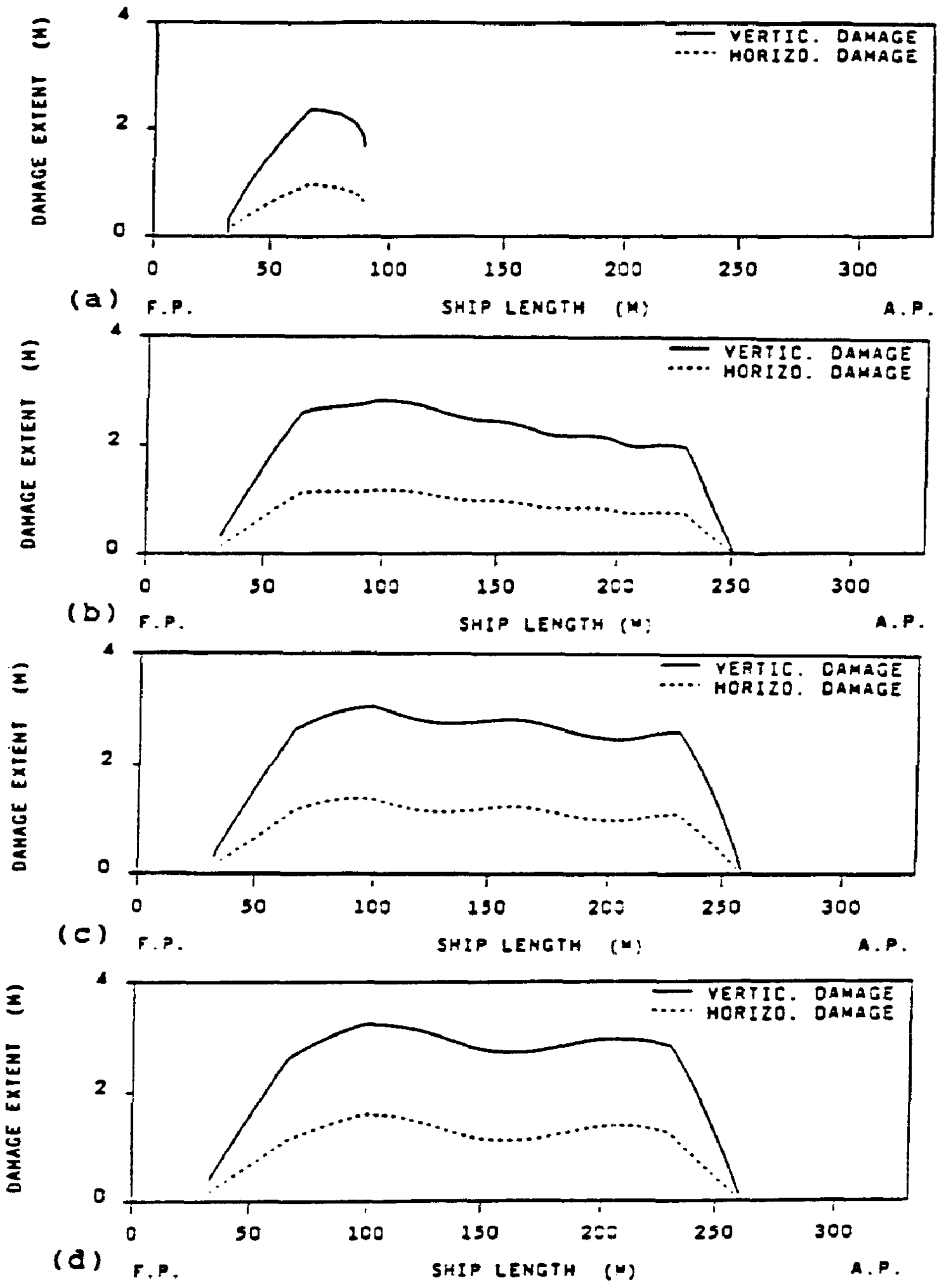

Fig. 6. Extent of damage due to forward collision with rupture of the standard hull. (a) v = 1 m/sec, (b) v = 3 m/sec, (c) v = 5 m/sec, (d) v = 7 m/sec.

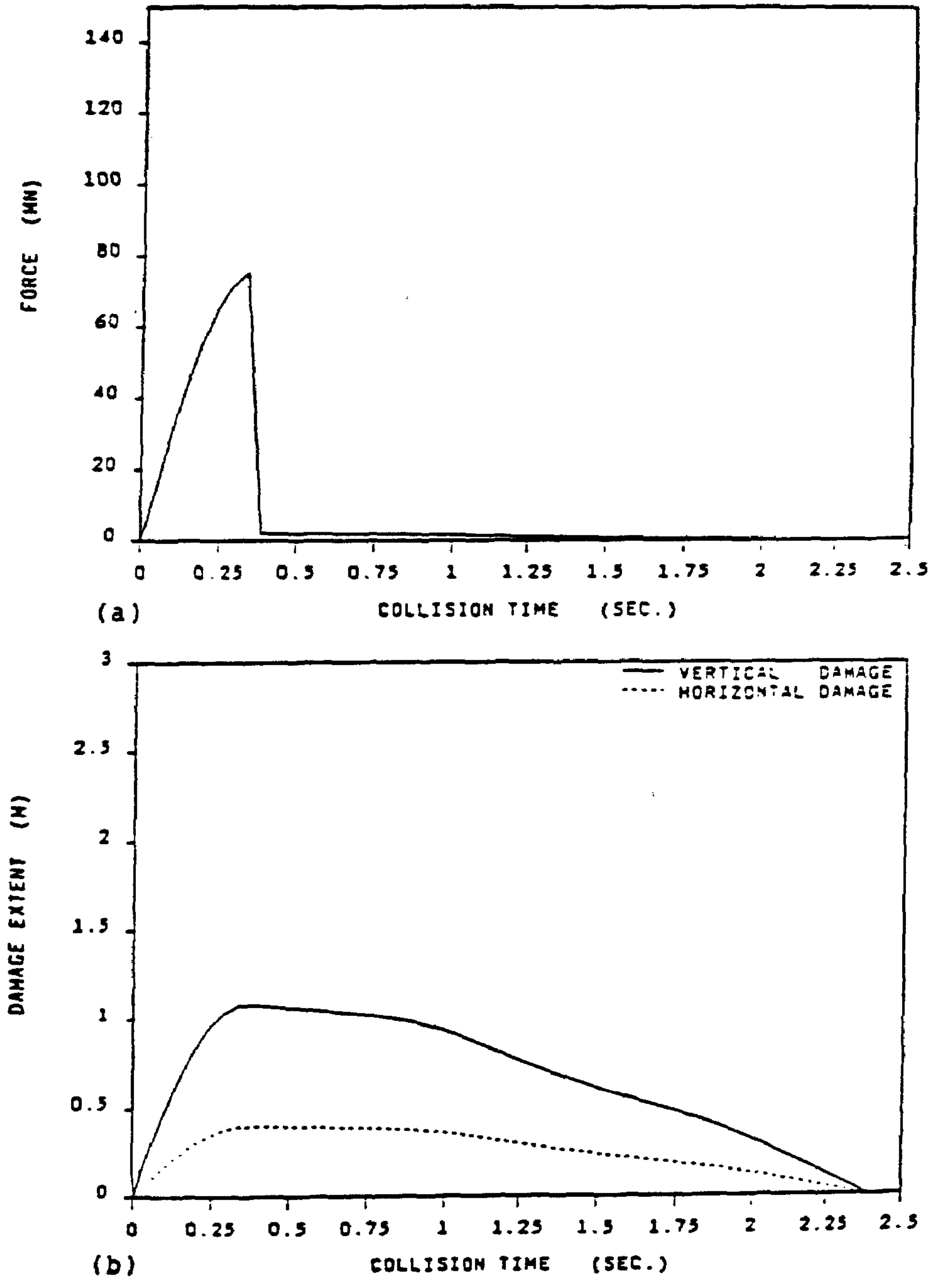

Fig. 7. (a) Generated force and (b) extent of damage due to the side collision of the standard hull during maneuver.

REFERENCES

1. N. Jones, "Structural Aspects of the Ship Collisions", Structural Crashworthiness, Jones and Wierzbicki, Butterworths, 1983, pp. 309–332
2. P.A. Frieze, R.C. McGregor and I.E. Winkle, "Marine and Offshore Safety", Elsevier, 1984.
3. W. Johnson and A.G. Mamalis, "Crashworthiness of Vehicles", Mechanical Engineering Publications Ltd., London, 1978.
4. A.N. Kinkead, "Some Collision and Grounding Considerations for Refrigerated Gas Carriers", Marine and Offshore Safety, Frieze, McGregor and Winkle, Elsevier, 1984, pp. 159–182.
5. J.C. Card, "Effectiveness of Double Bottoms in Preventing Oil Outflow from Tanker Bottom Damage Incidents", Marine Technology, 1975, pp. 60–64.
6. N. Jones, "Plastic Behaviour of Ship Structures", Transactions of the Society of Naval Architects and Marine Engineers, Vol. 84, 1976, pp. 115–145.
7. N. Jones, "A Literature Survey on the Collision and Grounding Protection of Ships", Report SSC–283, 1979, U.S. Coast Guard, Washington, D.C.
8. V.U. Minorsky, "An Analysis of Ship Collisions with Reference to Protection of Nuclear Power Plants", Journal of Ship Research, Vol. 3, 1959, pp. 1–4.
9. H. Vaughan, "Bending and Tearing of Plate with Application to Ship–Bottom Damage", 'The Naval Architect', Journal of the Royal Institution of Naval Architects, May 1978, pp. 97–99.
10. H. Vaughan, "Damage to Ships Due to Collision and Grounding", Det norske Veritas, Oslo, Technical Report 77–345, August 1977.
11. K.R. Johnsen and H. Vaughan, "Estimation of Structural Damage on Kockums 133000 m^3 Liquid Natural Gas Carrier Due to Collision and Grounding", Det norske Veritas, Oslo, Technical Report 77–275, July 1977.
12. J. Poudret, M. Huther, P. Jean and H. Vaughan, "Grounding of a Membrane Tanker, Correlation Between Damage Predictions and Observations", Symposium on Extreme Loads, Arlington, pp. 125–132, Society of Naval Architects and Marine Engineers, 1981.
13. H. Vaughan, "The Tearing Strength of Mild Steel Plate", Journal of Ship Research, Vol. 24, June 1980, pp. 96–100.
14. N. Jones and W.S. Jouri, "A Study of Plate Tearing for Ship Collision and Grounding Damage" Journal of Ship Research, Vol. 31, 1987, pp. 253–268.
15. B.M. Johansson, A.J. Keinonen and B. Mercer, "Technical Development of an Environmentally Safe Tanker", SNAME, Proceedings of the STAR Symposium, June 1981.
16. I.F. Glen, C.G. Daley and G. Tam, "Analysis of the Structure of the Proposed CCG Polar Class 8 Ice Breaker Under Extreme Ice Loads", Transactions SNAME, 1985.
17. H. Vaughan, "Flexural Response of Ships to Impact Loads", Transactions of the Royal Institution of Naval Architects, Vol. 128, 1986, pp. 259–267.
18. G.V. Asadi, "Dynamic Response of Ship Structures to Impact Loads", Ph.D. Thesis, University of British Columbia, May 1989.