SSC-283

A LITERATURE SURVEY ON THE COLLISION AND GROUNDING PROTECTION OF SHIPS

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SHIP STRUCTURE COMMITTEE

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An Interagency Advisory Committee
Dedicated to Improving the Structure of Ships

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SR-1246

Worldwide concern for protection of the environment from the consequences of maritime casualties is increasing. As a result, interest and activity in ship casualty research has accelerated. Such activity, now involving cooperation on an international level, includes the documentation and analysis of casualties, analytical and experimental research in structural design and response, vessel maneuverability and human factors studies, and design of collision/stranding structural protection systems.

As a prerequisite to planning and initiating future collision/stranding research projects, the Ship Structure Committee has undertaken a project to review all past, ongoing and planned activity in this area.

The present report contains the results of this review, a bibliography of technical reports on the subject, a summary of the current state of research, and recommendations for future work.

Henry H. Bell
Rear Admiral, U. S. Coast Guard
Chairman, Ship Structure Committee

SSC-283

FINAL REPORT

on

Project SR-1246

"Surveillance of Ship Collision/

Stranding Research Studies"

A LITERATURE SURVEY ON THE COLLISION AND GROUNDING PROTECTION OF SHIPS

by

Norman Jones

MASSACHUSETTS INSTITUTE OF TECHNOLOGY

under

Department of Transportation
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16. Abstract

This report contains a literature survey on various types of ship collisions including stranding and grounding. Comments are made on several aspects of the ship collision problem and the report concludes with a suggested program of research.

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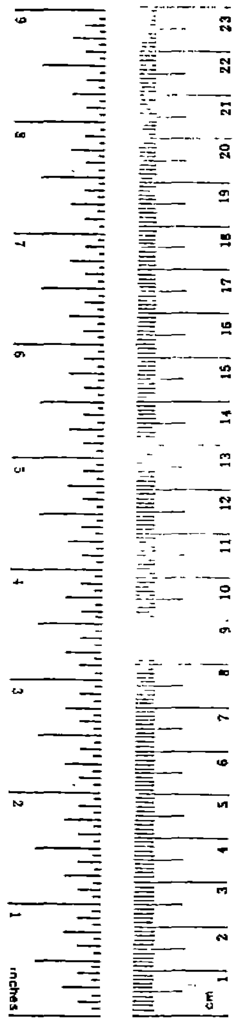
METRIC CONVERSION FACTORS

Approximate Conversions to Metric Measures

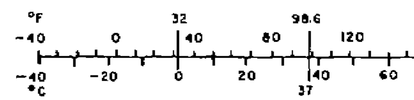
Symbol	When You Know	Multiply by	To Find	Symbol
		LENGTH		
in	inches	*2.5	centimeters	cm
ft	feet	30	centimeters	cm
yd	yards	0.9	meters	m
mi	miles	1.6	kilometers	km
		AREA		
in^2	square inches	6.5	square centimeters	cm^2
ft^2	square feet	0.09	square meters	m^2
yd^2	square yards	0.8	square meters	m^2
mi^2	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). For other exact conversions and more detailed tables, see NBS Misc. Publ. 286, Units of Weights and Measures, Price $2.25, SD Catalog No. C13.10:286.



Approximate Conversions from Metric M

Symbol	When You Know	Multiply by	T
		LENGTH	
mm	millimeters	0.04	
cm	centimeters	0.4	
m	meters	3.3	
m	meters	1.1	
km	kilometers	0.6	
		AREA	
cm^2	square centimeters	0.16	
m^2	square meters	1.2	
km^2	square kilometers	0.4	
ha	hectares (10,000 m^2)	2.5	
		MASS (weight)	
g	grams	0.035	
kg	kilograms	2.2	
t	tonnes (1000 kg)	1.1	
		VOLUME	
ml	milliliters	0.03	
l	liters	2.1	
l	liters	1.06	
l	liters	0.26	
m^3	cubic meters	35	
m^3	cubic meters	1.3	
		TEMPERATURE (exact)	
°C	Celsius temperature	9/5 (then add 32)	



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Nomenclature

g	gravitational constant
ℓ_o	reference length
t_o	reference time
B	width of beam or plate
E	Young's Modulus
E_a	energy absorbed
E_T	total energy absorbed in ton-knot^2 units
H	beam or plate thickness
$2L$	span of beam or plate
P	concentrated load
$\bar{P}_c$	concentrated static plastic collapse load
R_T	resistance factor defined by equation (2.5)
T_{ij}	surface traction tensor
W	transverse displacement at the mid-span of a beam or plate
$\dot{\varepsilon}$	uniaxial strain rate
ν	Poisson's ratio
ρ	density of material
σ_y	uniaxial yield stress
σ_y'	uniaxial yield stress at a strain rate $\dot{\varepsilon}$
σ_u	ultimate tensile stress

NOTES

1. Introduction

This article focuses on the structural response of ships during collisions and groundings, and does not discuss other important topics such as traffic and pollution control and collision and stranding probabilities [1,2, etc].

Despite this restriction, a surprisingly large number of articles have been published on various aspects of the structural strength of ships during collisions since Minorsky's pioneering paper on the protection of nuclear powered ships [3]. However, by way of contrast, very little effort appears to have been expended on the ship grounding problem.

The field of collision protection is not only relevant to the design of nuclear-powered submarines and aircraft carriers and a few other vessels examined in the earlier work, but now includes within its province,oil tankers, LNG carriers, and chemical carriers with hazardous cargoes. Moreover, existing studies must be continued, or even initiated in some cases, to investigate the collision protection of large nuclear-powered tankers (600,000 dwt in [4]), the effects of collisions between supply ships and various offshore structures, collision protection of offshore oil storage tanks [5], protection of ships transporting spent nuclear fuels to nuclear reprocessing plants (e.g., from Japan to Windscale in the U.K.), the protection of bridge piers against ship impact, ice collision damage of ships navigating in Arctic waters [6,7], and many other topics, including the collision protection of oil barges [8] and high-speed marine vehicles.

Minorsky prepared a comprehensive review [9] of the literature extant in 1975 on ship collision protection and other reviews have been published by Woisin [10] and in Reference [11]. Thus, to prevent further redundancy, these earlier efforts are not duplicated in this report and the early work on ship collisions is reviewed only when it is required for completeness of presentation. However, all the known published work on the structural strength of ships during collisions which is not quoted in the References of this report is presented in Appendix 2 for convenience.

2. Some general Remarks on the Collision Protection of Ships and Marine Vehicles

2.1 Minor and Major Collisions

There appears to be no universal agreement as to what constitutes minor and major ship and marine-vehicle collisions. The important characteristics used to describe a major collision of an oil tanker, for example, could lie within the classification of a minor collision for a nuclear-powered ship because of the quite different design requirements. Nevertheless, the following, perhaps restrictive, definitions are used in this report:

Minor Collision: this is used to describe a collision when the hull damage of a ship sustained by whatever means is accommodated by elastic and inelastic material response without fracture. In other words, the shell plating of a ship could be badly dented but, if fracture did not occur in the outer plating of a single-hull ship or in the inner plating of a double-hull ship, then it would be classed as a minor collision (i.e., a low energy collision). This is the kind of behavior examined in Reference [12] for oil tankers.

Major Collision: this is used to describe a collision which causes large inelastic strains and fracture of the shell plating. Minorsky's work in Reference [3], for example, was developed for major collisions of nuclear-powered ships.

It is quite clear that further classifications may be required in practice to provide a more accurate description of a particular ship collision. One example is the following classification scheme which is presented here to promote discussion of this topic:

I - dents in shell plating, with maximum permanent transverse displacements up to the order of five plate thicknesses without fracture.

II - similar to I, but with local fracture.

III - dents in shell plating, with maximum permanent transverse displacements up to the order of one-quarter of the longitudinal distance between adjacent transverse web frames without fracture.

IV - similar to III, but with local fracture.

V - protrusion of an object (e.g., bow, bridge pier) into the hull a distance greater than one-quarter of the longitudinal spacing between adjacent transverse web frames, but with no damage of any transverse web frames.

VI - similar to V, but with one damaged transverse web frame.

VII - similar to V, but with two damaged transverse web frames.

VIII - similar to V, but with three or more damaged transverse web frames.

Clearly, even more classifications could be defined in order to cater for damage to the bottom, decks, and bows, and to distinguish between damage above and below the waterline.

2.2 Energy-Absorption Schemes

The protective structural arrangements which have been examined in most of the studies on nuclear-powered ships, oil tankers, and LNG carriers [2] are similar and utilize either the normal structural designs for these vessels or a slight modification which includes additional decks specifically designed to absorb the kinetic energy lost during a collision. However, it is clear that the design requirements for these various ships are different. Clearly, the bow of a striking ship must not penetrate the containment vessel of a nuclear-powered ship. Presumably a similar design requirement would be used for an LNG carrier, except that a number of tanks would require protection. The entire length of an oil tanker requires protection so that it is only feasible in this case to provide protection for minor collisions. No doubt even other design requirements might be necessary for the various other ship collision scenarios mentioned in Section 1.

It emerges clearly that a designer needs a collection of possible collision protection schemes and devices so that the most suitable for the particular problem at hand may be used. Many energy-absorption methods have been investigated in the engineering literature [13], particularly in the automobile crashworthiness field [14, 15]. A brief review of this activity up to 1975 was presented in Reference [11]wherein it was observed, from an economic viewpoint, that most energy-absorption schemes were not practical for ships because of the large surface areas requiring protection. However, it was suggested that honeycomb structures, or nests of tubes, provide a feasible alternative to deck structures, particularly for marine vehicles.

It is interesting to observe that the collision protection scheme for a nuclear-powered ship has evolved into a cellular structure [16, 17], which is superficially similar to the honeycomb or tube nest schemes proposed in Reference [11]. However, the protective scheme proposed in References [16] and [17] is a resistance-type structure which must be strong enough to withstand the maximum possible impact force, while absorbing only a small fraction of the total impact energy. On the other hand, the honeycomb structure discussed in Reference [11] is an absorption-type structure which is designed to absorb most of the impact energy.

It appears that a resistance type of protective system would be expensive to build and quite heavy because the associated static collapse load must be greater than any impact load which it may encounter. However, it may be quite difficult to absorb a great deal of energy with an absorption type of protective system. In some cases it might be attractive to develop a hybrid system in which a honeycomb structure would absorb the energy of minor collisions (and then be replaced), while the heavier supporting structure of the resistance type would remain available to protect a ship from the damaging effect of the residual impact during a major collision. It is conceivable that the weight of a hybrid protection arrangement required for major collisions would be less than a wholly resistance type of structure because the absorption structural portion (e.g. honeycomb) could act as an attenuator by reducing the accelerations and therefore the forces which act on the resistance portion.

It is important to emphasize that many research groups are currently investigating and developing new energy-absorption devices in a number of engineering fields, so that the foregoing conclusions and those in Reference [11] must be held somewhat tentative in case improved systems become available. Recently, Johnson, Reid, and Singh [18] found that a rolling torus device absorbs considerably more energy than previously reported in the literature. The possible excitation of higher mode dynamic plastic structural responses in order to improve energy-absorption characteristics was recently discussed in References [19] and [20]. Moreover, the use of composite materials in energy-absorption systems requires further exploration. In References [21] and [22], for example, it was observed that fiber-reinforced beams absorbed more kinetic energy than similar isotropic ones. Thus, the research work on various energy-absorption systems throughout the field of engineering must be continuously monitored in order to more quickly generate a number of collision protection design options needed by naval architects for a variety of applications.

2.3 Scaling

Similarity principles have been developed extensively in the field of fluid mechanics but have received less attention in structural mechanics, even though structural model testing is probably more widespread throughout the various branches of engineering [23]. The demands of proper scaling can be very restrictive. In reference [23], for example, it is shown, for a linear elasto-dynamic problem, that $1/(1-2\nu)$, $2(1+\nu)\ \rho g\ \ell_o/E$, and $2(1+\nu)\ \rho\ell_o{}^2/Et_o{}^2$ must be the same for a prototype and a geometrically similar model, where E and ν are the elastic constants, g is the gravitational constant and ℓ_o and t_o are the characteristic or reference values of length and time. In addition, the ratio T_{ij}/E must be the same on the boundary of an elastic prototype and a geometrically similar model.

Invariance of ν is obviously restrictive but can be achieved when using the same materials for the prototype and a geometrically similar model. In this circumstance, it is evident that ρ and E are the same so that the dimensionless parameter $2(1+\nu)\ \rho g\ell_o/E$ can only be satisfied if the prototype and a geometrically similar model have the same physical dimensions when tested at the same location. Invariance of the parameter $2(1+\nu)\ \rho\ell_o{}^2/Et_o{}^2$ requires equality of ℓ_o/t_o for the prototype and model which would be difficult to achieve in practice, while the surface tractions T_{ij} must be the same in the prototype and model. Thus, difficulties with proper scaling are encountered even for the linear elasto-dynamic case. Fortunately, the invariance of ν should not present any difficulties for a ship collision problem, while gravitational effects are not an important factor in the structural response. Furthermore, if a ship collision may be regarded as quasi-static then the parameter related to inertia forces is not relevant so that proper scaling could be achieved for any linear elastic effects during a ship collision.

It appears that no published investigations, except Reference [24], have examined whether the structural response of ships during collisions may be considered to be static, or whether it is necessary to retain the influence of inertia forces. It was suggested in

Reference [25] that the structural response of a panel in a marine vehicle during a severe slam could be accurately predicted with a static analysis, provided the duration of the pressure pulse is longer than the fundamental period of elastic vibration. Indeed, encouraging agreement was obtained between the theoretical predictions according to a static analysis and some experimental results which were recorded on a one-quarter scale model of a section of the bottom of a Coast Guard cutter [25]. However, the inertia terms must be retained when the duration of a pressure pulse is short. It was shown in Reference [24] that the structural response of the shell plating of the particular tanker design considered in Reference [12] could be predicted with sufficient accuracy using a static analysis. It would therefore appear worthwhile to develop further these simple ideas in order to provide guidelines which indicate when static analyses could be used with no sacrifice in accuracy, although it is likely that the retention of inertia terms would be unavoidable when analyzing even minor collisions of high-speed marine vehicles.

The collision or grounding of a ship is likely to involve extensive inelastic behavior and other non-linear effects, which would introduce additional parameters into the basic equations governing the structural response. Duffey [26] has shown that, in addition to various restrictions, the influence of material strain-rate sensitivity cannot be properly scaled when a prototype and a geometrically similar model are constructed of identical materials. Further theoretical objections may well be encountered when attempting to correlate the responses of prototypes and models which are described by more accurate constitutive equations (including strain-rate history effects, for example) and when fracture features in the response. It should be remarked, however, that very little is known about dynamic inelastic fracture, although a criterion for the dynamic inelastic failure of beams is presented in Reference [27].

It is of some interest to assess the importance of material strain-rate sensitivity on the plating response during a ship collision, since mild steel is notoriously strain-rate sensitive [28] and exercises an important influence on the dynamic plastic response of various structural members [29, 30]. The dynamic flow stress (σ_Y') in a uniaxially loaded mild steel specimen, which is stretched at a constant strain rate ($\dot{\varepsilon}$), approximately obeys the empirical Cooper-Symonds constitutive relation

$$\sigma_Y'/\sigma_Y = 1 + (\dot{\varepsilon}/40)^{1/5} \quad , \tag{1}$$

where σ_Y is the uniaxial static flow stress. It was estimated in Reference [24] for a particular case that the duration of impact of a striking vessel travelling with an initial velocity of 2 knots is 2 seconds, approximately. This calculation assumed that the speed of the striking vessel decreased linearly until motion ceased when the 1-inch-thick steel shell plating of the struck ship had deformed 40/150 of the span, which is the threshold of rupture according to the experimental results. If the rupture strain of the plating in a uniaxial test is 0.3 then the average strain rate during the response is 0.15 sec^{-1} and equation (1) predicts $\sigma_y' \simeq 1.33\sigma_y$. In other words, the dynamic flow stress at the maximum strain location in the plating for this particular case is about 33 percent larger than the uniaxial yield stress recorded in a conventional tensile test. It should be noted that equation (1) was developed from experiments on mild steel specimens which suffered relatively small strains. The experimental results of Campbell and Cooper [31] indicated that the corresponding relationship for the ultimate tensile stress was quite different. Indeed, Wierzbicki et. al. [32] and others have observed that a linear counterpart of equation (1) may be adequate for structural members which undergo large plastic strains.

The calculation for the average strain rate at a particular location is obviously very approximate and it is undoubtedly quite different for other collision situations. Nevertheless, this calculation does appear to suggest that the influence of material strain-rate sensitivity is sufficiently important to merit further consideration. Akita et al [33] conducted some idealised static and dynamic ship collision model tests and observed that the energy absorbed in a dynamic test was larger than that which was absorbed in the corresponding static test, a circumstance which was attributed to the influence of material strain-rate sensitivity.

Woisin [34] has examined the influence of various factors in the similarity laws for the structural damage associated with ship collisions and also observed that not all non-dimensional parameters may be simultaneously satisfied for a prototype and a geometrically similar model. Woisin then makes some suggestions as to which effects are unimportant and uses approximate relations for those that must be considered.

2.4 Influence of a Striking Bow

The ship collision problem is obviously a very difficult one to analyse theoretically or even experimentally because the rather complicated ship structure responds in the inelastic range, with large deflections and other non-linear effects such as fracture and buckling. This complexity is further compounded by the many possible collision scenarios between different types of ships having different weights and travelling at different speeds with different angles of approach and with various impact locations, etc. Moreover, the potential for energy absorption in the bows of different striking ships is quite variable and therefore influences the partition of energy between a striking ship and a struck ship. Minorsky [3] considers this effect in his semi-empirical procedure by including the volume of material in the damaged portions of decks, longitudinal bulkheads, and shell plating in a striking vessel.

An interesting experimental study was reported by Akita and Kitamura [35] who observed that the bow structure of a striking ship plays a very important role during a collision between two ships. The included stem angle, rake and framing of a bow clearly are important, but the ratio between the strength of the bow of a striking ship and the strength of the side of a struck ship has a major influence on the partition of energy absorption between the two ships as shown in Figure 1. Generally speaking a stiff bow

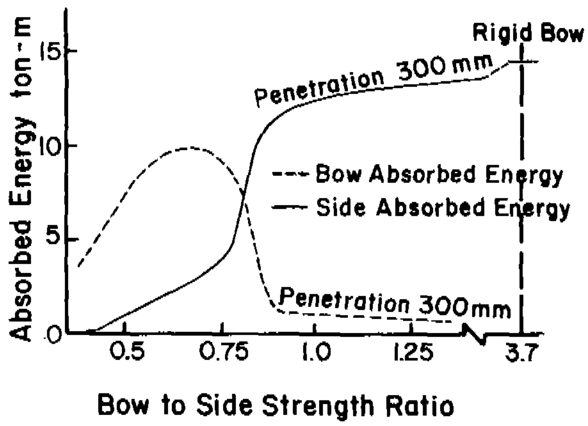


Figure 1

Variation of Absorbed Energy with Ratio of Bow to Side Strength [35].

(e.g. icebreaker) would absorb very little energy so that most of the kinetic energy lost during impact must be absorbed by the side of the struck ship. On the other hand, a weak bow may absorb most of the kinetic energy lost during a collision, leaving the side of the struck ship essentially undamaged. Incidentally, Cheung [36] has suggested a design for a soft bow.

The authors of Reference [12] evaluated the plastic energy absorbed by a ship when struck by a vertical rigid bow and compared it with the plastic energy absorbed in an identical ship struck by a rigid bow with a 15° rake. The bow imprints on the sides of the struck ships were quite different as shown in Figure 1 of Reference [12]. It turned out that the plastic energy absorbed in a ship struck by a raked rigid bow ranged from 49 to 60 percent of the plastic energy which would have been absorbed if struck by a vertical rigid bow.

3. Recent Published Work on Ship Collision Research

3.1 Japan

The outstanding theoretical and experimental work on the structural mechanics of ship collisions which was conducted in Japan prior to 1975 has been reviewed in References [9] to [11]. Moreover, Reference [33] contains an excellent summary of the major results from this comprehensive Japanese ship collision research program.

More recently, Ando and Arita [37] examined experimentally and theoretically the collision protection characteristics of double-hull structures which consist of an outer hull plating and an inner shell connected by flat horizontal and vertical girders welded to form cubical cellular spaces as shown in Figure 2 of [37], with the dimensions listed in Table 1 of [37]. This structural arrangement is similar to that proposed in Reference [11] except that the cellular spaces in [11] are hexagonal.

The authors conducted experiments on 11 structural models penetrated statically by rigid idealized bows which were perpendicular to the double-hull structure as indicated in Figure 2 of [37]. It is evident from the idealised curves in Figure 4 of [37] and Figure 2 here that a typical bow load/bow penetration curve has two distinct humps (see also Figure 3(a) here). The first peak is associated with the membrane forces which are developed in the hull plating and is presumably reached when the plate ruptures, after which the bow load drops sharply, while the bow penetrates with little resistance from the horizontal girders until it contacts the vertical girders. The second peak is related to the ultimate strength of the vertical girders. Typical absorbed energy/bow penetration curves with bow penetrations up to approximately the total thickness of a double-hull structure are presented in Figures 11 to 13 of [37] and Figure 3(b) here, and the relation between the absorbed energy and the corresponding volume of a double-hull model is shown in Figures 14 and 15 of [37]. The authors found that considerable energy would be absorbed due to the development of membrane tension in the shell plating during a minor collision which supports the conclusions in Reference [12].

The influence of bow radius on the structural response of a double-hull model is shown in Figures 10 and 13 of Reference [37]. All the bows had an included angle of 60^{o} and no rake.

An approximate theoretical analysis based on the idealised behavior described in Figure 2 here was developed in Reference [37] using a strength of materials approach and found to predict the overall features of the response.

It appears from Figure 16 of Reference [37] that a tenfold increase in the average ramming speed of the bow from 20 mm/min to 200 mm/min does not produce a markedly different response.

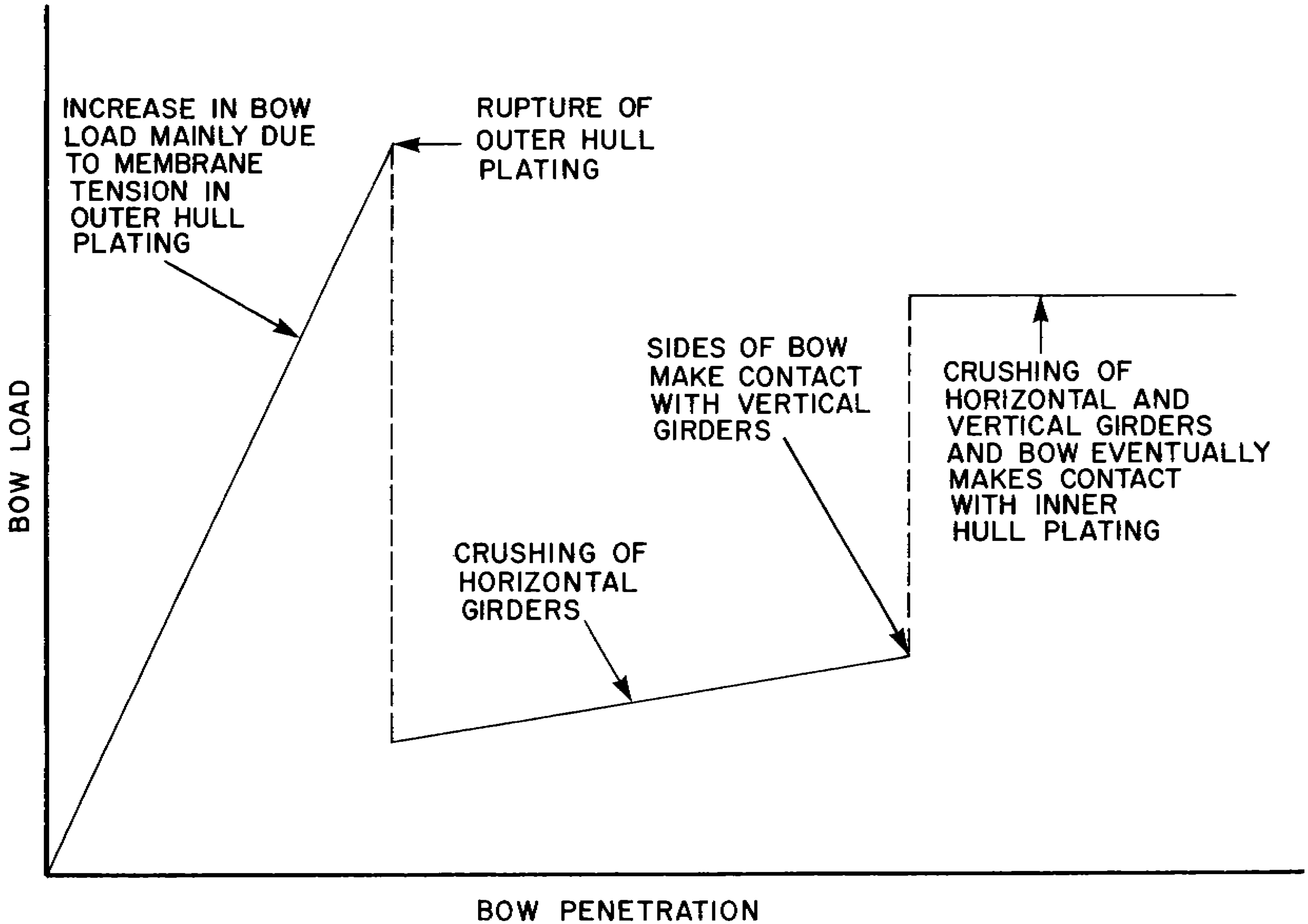
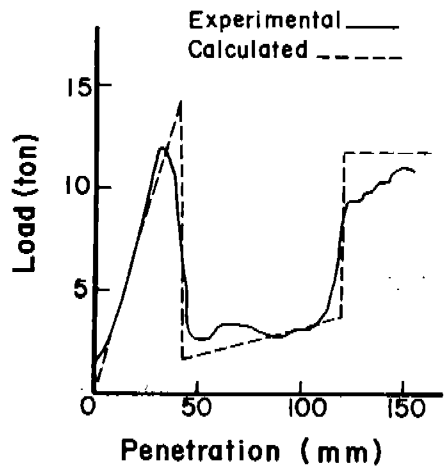
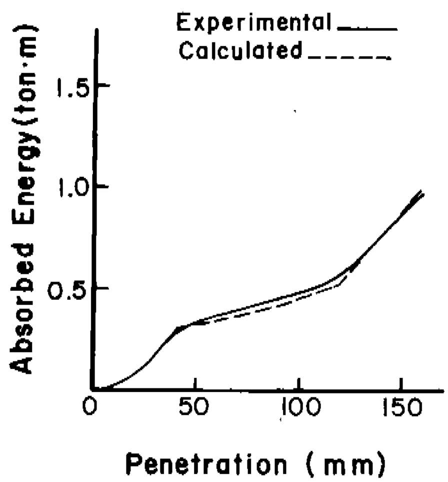

INCREASE IN BOW LOAD MAINLY DUE TO MEMBRANE TENSION IN OUTER HULL PLATING
RUPTURE OF OUTER HULL PLATING
SIDES OF BOW MAKE CONTACT WITH VERTICAL GIRDERS
CRUSHING OF HORIZONTAL AND VERTICAL GIRDERS AND BOW EVENTUALLY MAKES CONTACT WITH INNER HULL PLATING
CRUSHING OF HORIZONTAL GIRDERS
BOW LOAD
BOW PENETRATION

FIGURE 2



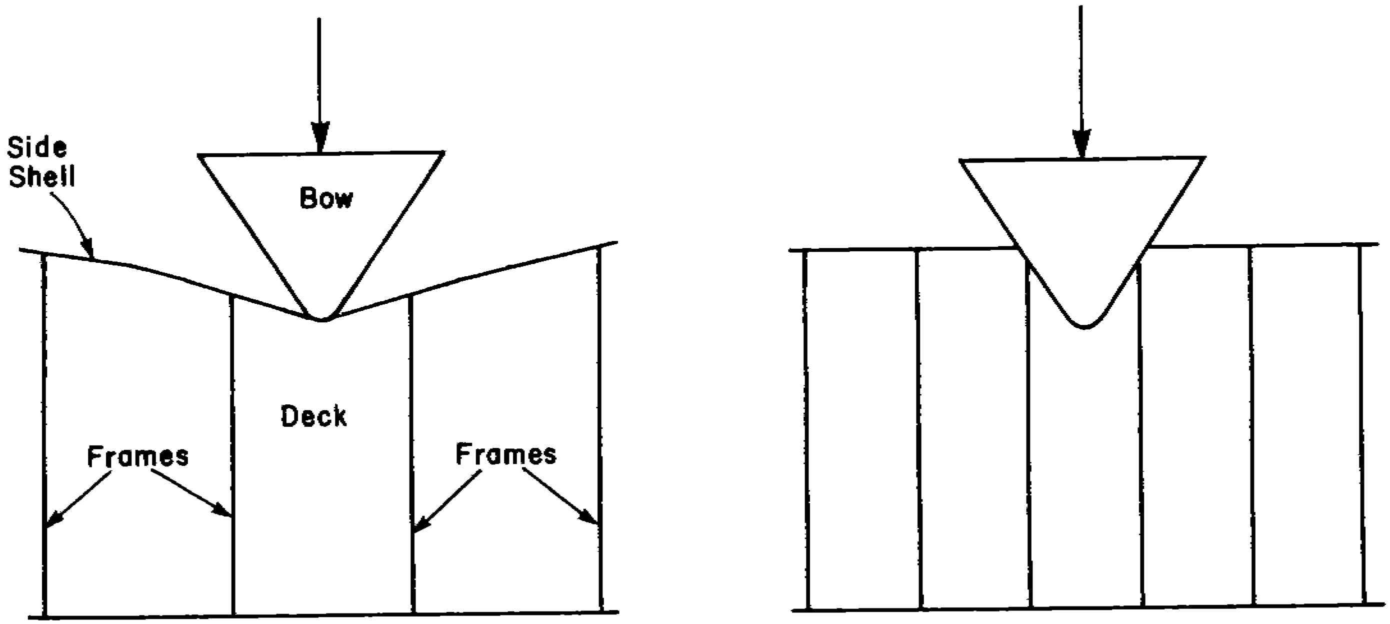
(a)



(b)

Experimental Results For Test Number N1 in Reference [37]

FIGURE 3



(a) Deformation or Buckling Failure Mode in References [33] and [38]

(b) Crack or Encroaching Failure Mode in References [33] and [38]

FIGURE 4

Akita et al. reported in Figures 8, 9, and 12 to 14 of Reference [33] the experimental results obtained from 8 idealised ship side models penetrated statically by rigid bows. The ship side models consisted of a side shell, two decks and transverse framing. The behavior of 11 other side structural designs was also examined in Figures 41 to 53 of Reference [33]. Arita, Ando and Arita [38] have recently presented some additional test results which explore more fully the influence of stiffener spacing, deck spacing, side shell thickness, and deck plate thickness on the potential collision protection of ship structures.

The authors of Reference [33] observed that deformation-and crack-failure modes sketched in Figure 4 here were responsible for the failure of the ship side models. A deformation-failure mode is characterised by buckling of decks and stiffeners over a relatively large area of the side shell and a large portion of the external load is supported by membrane tension prior to the rupture of the side shell. This behavior contrasts with a crack type of failure which is characterised by a local penetration of a rigid bow which ruptures the side shell and decks as indicated in Figure 4(b) here. Arita, Ando and Arita [38] also found these kinds of behavior in their more comprehensive test series. However, a deformation-failure mode is now called a buckling-type failure, and the crack-type failure is renamed an encroaching-failure mode. The experimental results in Figure 5 of [38] indicate that the failure mode changes from a buckling type to an encroaching type as the stiffener or deck spacing decreases. A semi-analytical criterion was developed in equations (7) to (9) and Figure 8 of Reference [38] in order to predict the type of failure mode for a particular side shell.

In the case of an encroaching-failure mode, the energy absorption continues to increase approximately proportional to the square of the bow penetration even after the rupture of the side shell. This contrasts with the behavior associated with a buckling-failure mode. However, at a given small penetration of a bow, an encroaching-failure mode absorbs less energy than a buckling-type as shown in Figure 5 here. This observation led the authors of Reference [38] to suggest that the dimensions of a side shell may be selected in order to achieve optimum collision protection. Thus, a buckling-failure mode would be sought to provide protection against a minor collision and an encroaching-failure mode for a major collision.

Section 2 [38] on the overall mechanics of ship collisions and Section 4 [38] on dynamic effects are similar to the studies reported in Reference [33].

In order to generate information on the design of buffers for the collision protection of both ship hulls and bridge piers, Nagasawa et al. [39] have reported recently the results of an experimental investigation into the static structural behavior of small idealised ship side and bow models which collide with a rigid bridge pier.

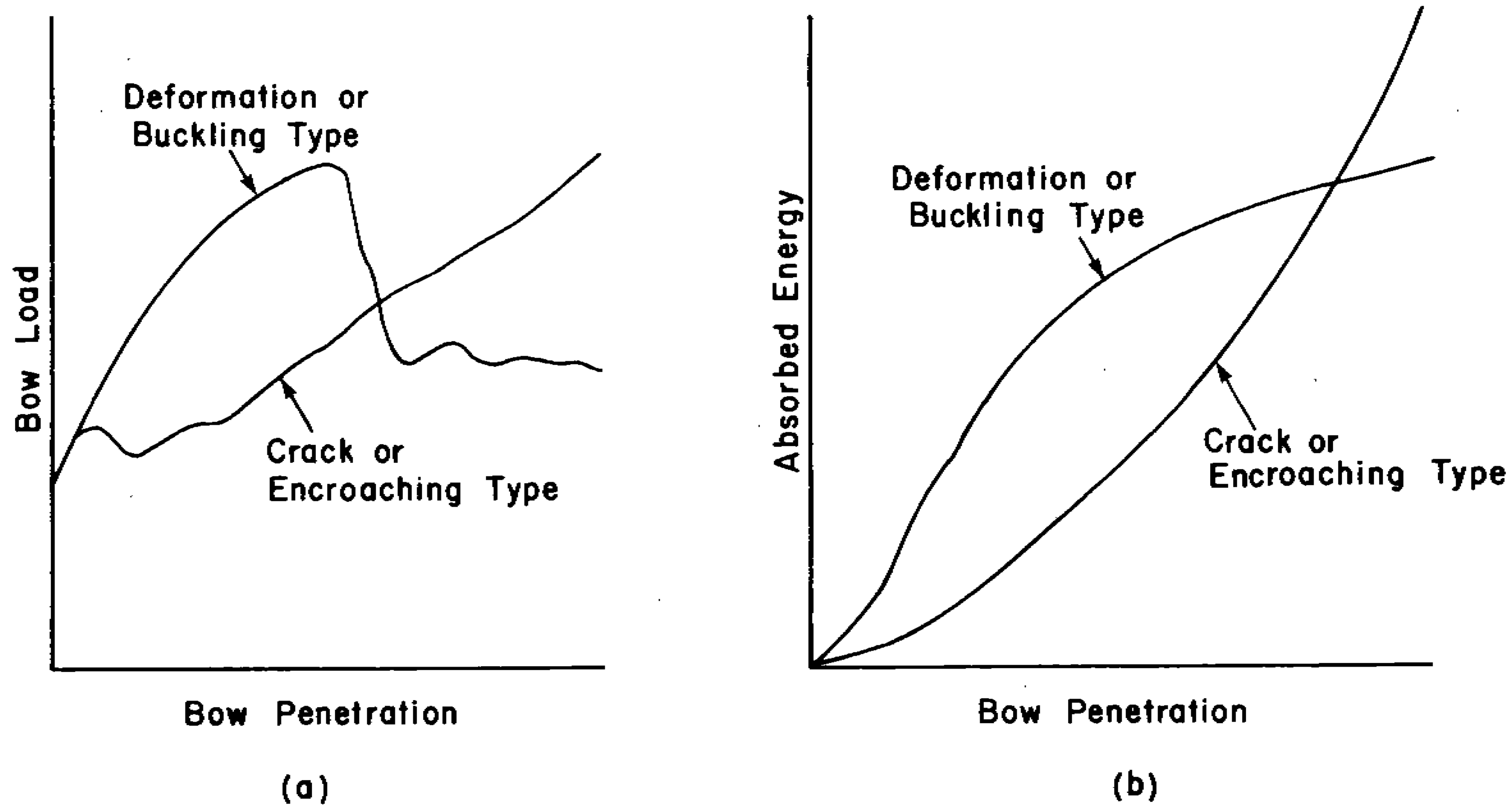

Deformation or
Buckling Type
Bow Load
Crack or
Encroaching Type
Bow Penetration
(a)
Deformation or
Buckling Type
Absorbed Energy
Crack or
Encroaching Type
Bow Penetration
(b)

FIGURE 5

The first part of this study explored the behavior of the idealised ship bows illustrated in Figure 1 of Reference [39] when loaded statically through the flat plane of a rigid bridge pier. It turned out that the buckling of the side bow plating (see photograph 1(b) in [39]) controlled the strength of these particular bows. In fact, this conclusion also could be predicted from the experimental results reported in Reference [35] for weak bows colliding quasi-statically with strong ship side structures. It is evident from the experimental results presented in Figure 3 of [39] that the maximum load experienced by a vertical bow for small amounts of bow deformation is considerably larger than the load associated with a raked bow. However, the experimental results in Figure 5 of [39] indicate only a small difference in the absorbed energies for large deformations of a bow.

In the second part of the study reported in Reference [39], the authors examined the structural behavior of longitudinally framed and transversely framed ship side models loaded statically by a rigid cylindrical body which is an idealisation of the corner of a bridge pier (see Figures 2 and 18 of [39]). It is apparent from photograph 2 and Figures 14 and 15 in Reference [39] that the structural behavior of the transversely framed ship side models were quite different to the longitudinally framed ones. The particular experimental results in Figures 10 to 13 of [39] indicate, for a given deformation of a side shell, that the corresponding load and absorbed energy of a transversely framed side shell are considerably larger than for a longitudinally framed side shell.

The collision forces developed during bow and side collisions with bridge piers are estimated in Figures 17 and 19 of [39]. Again, it transpired that the collision forces in ship sides with transverse framing are larger than those associated with longitudinal framing.

This bridge pier collision work is continuing and the behavior of curved plate models which simulate tanker bows are being examined currently. Further investigations are also in progress in Japan on the collision protection of offshore oil-storage facilities.

3.2 The Netherlands

A numerical study was undertaken in the Netherlands to evaluate the collision resistance of double-hull L.N.G. tankers with double bottoms struck by ships having rigid bows [40].

The basic equations for this numerical work were developed by isolating the various structural members which participate in a ship collision and estimating the associated load-deformation characteristics. In this way, approximate relations for the behavior of the hull plate, decks, stringers and web frames were derived using various simple formulae from several sources. This general approach to the problem was similar to that developed in Reference [12] except the collisions were not only minor ones. It was then assumed that the total force acting on the side of a ship equalled the sum of the contributions from all the structural members of an L.N.G. tanker which deformed during a collision. This implies that the behavior of the various members are completely uncoupled and act in parallel with none in series. The importance and accuracy of this assumption was not examined in Reference [40] and therefore remains contentious.

The authors then formulated the equations of motion for an L.N.G. tanker struck by a ship when both were idealised as rigid bodies, but with the local forces between the vessels described by the approximate force-deflection characteristics discussed above. Virtual added mass coefficients of 0.5 and 0.4 were used for an L.N.G. tanker in deep water and a striking ship, respectively. The yield stress of the ship structural material was taken as $\sigma_y'=280N/mm^2$ in order to cater for the phenomenon of material strain-rate sensitivity. However, no information was offered on the choice of this value and obviously no distinction could have been made between the temporal and spatial variations of strain rate for different structural members during various collision scenarios.

The equations of motion were integrated numerically with respect to time using the Runge-Kutta method and 0.01-second time steps. Numerical results are presented for L.N.G. tankers with integrated cargo tank systems, free standing cargo tanks made of aluminum alloy, and spherical pressure vessel cargo tanks. It was found that a 10,000 ton striking ship travelling at 4 knots could damage the cargo tanks of the three types of L.N.G. tankers. However, the amount of damage is obviously sensitive to the details of both struck and striking ships but it is surprisingly insensitive to the impact locations, obliquity, and eccentricity of a collision according to the numerical results.

A great deal of interesting information is presented in Reference [40] but since it is a numerical-empirical study the actual values of the various parameters must remain tentative until some supporting experimental evidence becomes available. However, the philosophy of approach to this difficult problem is a sensible one and is capable of further refinement as additional experimental results are generated.

3.3 United Kingdom

It appears that only one paper has been published in the United Kingdom since an appealingly simple approximate theoretical procedure was developed by the Naval Construction Research Establishment (NCRE) and published in 1967 [41]. Haywood [42] used Minorsky's [3] approximate calculation procedure to assess the collision resistance of a 57,000 ton displacement LNG carrier (75,000 m^3). A number of assumptions allowed simple calculations to be made which indicated that the inner shell LNG carrier tanks and insulation system are likely to be ruptured by striking ships of 20,000, 50,000, or 100,000 tons displacement travelling at approximately 9, 6, and 5 knots, respectively.

Johnson and Mamalis [43] have recently prepared a monograph on the crashworthiness of vehicles with a section on ship collisions which largely discusses the work contained in References [3] and [11].

3.4 United States

It has already been remarked that Minorsky prepared a valuable survey of most ship collision papers published up to 1975 [9]. This report contains a bibliography of 74 papers with extended summaries of 34.

Zahn presented data on the details of 134 ship collisions in Reference [44]. Minorsky [45] has added to this data base and observed that about one-third of struck tankers (oil, L.N.G., L.P.G., ore-oil, chemical) caught fire or exploded, while about one-fifth of struck non-tankers sank as the result of a collision.

Minorsky, Parker, and Gotimer [46] have further studied the available ship accident data in an attempt to predict the casualty statistics for nuclear tankers. It is concluded that a nuclear tanker equipped with a proper collision barrier (e.g.[16, 17]) would be collision-proof, except for possible rupture of piping and equipment in the reactor compartment due to excessive dynamic loads.

The procedure in References [12], [47] and [48] was developed in order to evaluate the ability of longitudinally framed ships, such as tankers, to withstand minor collisions. The plastic energy absorption of the side of a struck ship was estimated using plasticity theory and various empirical relations from several sources for the load-deflection and energy-absorbing characteristics of the structural members which were deformed during a ship collision. It transpired that the elastic energy associated with a ship collision was negligible compared to the plastic work. To simplify the calculations, the bow of the striking ship was assumed rigid. The proposed calculation procedure was used to examine six particular cases of single-shell and double-shell 120,000 dwt tankers struck by a 20,000 ton displacement ship.

It is evident from Table 4-2 in Reference [48] that the membrane strain energy in the longitudinal side members makes a major contribution to the total plastic strain energy absorbed before shell plate rupture. The membrane strain energy absorbed in the deck is the next most important contribution to the total strain energy. It turns out that the membrane strain energies developed in the sideshell and deck account for about 88 to 98 percent of the total strain energy in the six cases examined in Reference [48].

The configuration of the striking bow, even though rigid, has a significant effect on the energy absorption characteristics of the struck ships in References [12] and [48] as already remarked in Section 2.4. It was also observed in Reference [48] that the total plastic energy absorption in a hull is approximately proportional to the shell plate thickness. Thus, similar energy-absorption capacities were found for single-hull and double-hull ships with the same overall side plating thickness. However, a double hull is superior to a single shell when punching or tearing action with little energy absorption occurs, since the inner shell may remain intact and prevent leakage of the cargo after rupture of the outer shell.

Oblique collisions were also examined in Reference [48] and it was observed that less energy was absorbed by a struck ship than would be during a right-angle collision. Moreover, the actual impact location of a rigid bow relative to the transverse bulkheads of a struck ship was found to exercise a very important influence on the amount of energy absorbed in a struck ship.

An estimate of the energy absorption capacity of an L.N.G. ship with spherical aluminum cargo tanks when struck at right angles by a 20,000 ton displacement vessel with a vertical rigid bow was also examined in Reference [48]. It was found that a striking ship with a velocity of 7.4 knots could be accommodated by plastic energy absorption in a 125,000 m^3 L.N.G. ship prior to plating rupture. However, the amount of energy absorption depends strongly on the location of the strike. Moreover, except for strikes within two web spaces of the bulkheads, the proximity of the L.N.G. cargo tank to the shell prevents the full potential of plastic energy absorption being realised in the side of the struck ship before a cargo tank is ruptured.

The calculations in Reference [48] were predicated on a large number of simplifications and assumptions, many of which require further experimental justification. For example, no fracture-mechanics criteria were incorporated in the method of analysis so that there was no possibility of plating fracture prior to the attainment of the ductility limit. Dynamic effects were neglected. Moreover, neither the destructive capability of nor the energy absorbed in a striking bow were examined, although some consideration was given to this latter point in Section 7 of Reference [48]. Furthermore, the ship bottom, bilge strake, and transverse bulkheads did not buckle, yield, or rupture. Thus, the damaged area in a struck ship was confined between two consecutive transverse bulkheads and above the bilge strake. Nevertheless, the theoretical method developed in Reference [48] provides a useful framework for the future study of minor ship collisions and is capable of further refinement as experimental results become available.

Chang [49] examined the collision protection of nuclear-powered ships with the grillage arrangement proposed by Woisin [16]. Chang used the well-known limit theorems of plasticity in order to study the forces in the collision barrier of a stuck ship and in the bow of a striking ship.

The plastic-limit analysis theorems were developed using equilibrium equations and geometrical relations for structures which undergo infinitesimal displacements. However, the post-yield characteristics of many structures are highly non-linear (see References [50] and [51] for a list of those references relevant to Naval Architecture). In fact, the experimental and theoretical transverse load-carrying capacity of the long plate in Figure 5 of Reference [52] is four times the classical static plastic collapse load when the maximum transverse displacement is double the associated plate thickness. Furthermore, the elastic or plastic buckling and post-buckling characteristics of structures are not recognised by the static plastic-collapse theorems. These severe limitations are acceptable for the design of a protective grillage because it is a resistance type of barrier and must therefore remain intact and not deform(appreciably) during a collision. On the other hand, the bow of a striking ship may deform significantly upon impact with a resistance type of protective barrier. Thus, the limit theorems should not be used to study the strength of a bow since the predictions could have significant errors and may underestimate or overestimate the actual failure load of a bow. Nevertheless, Chang's statement that the failure load of the protective grillage must be larger than the failure load of a striking bow must remain true.

Chang [49] employed a numerical finite-element method to study the behavior of a protective barrier subjected to the collision load estimated from the experimental tests of Woisin. The numerical stress distribution associated with the peak collision load indicated that most portions of the structure remained elastic, with plastic yielding at only a few locations, which was apparently confirmed by the experimental results. The preliminary results in Reference [49] will be incorporated into a final report to be published in the near future.

The authors of Reference [53] conducted some collision tests on a stationary floating steel ship model which was impacted at right angles on the centre transverse bulkhead (located at the mid-ship section) by a floating wooden ship model. It was the object of these tests to record the shock response during a ship collision in order to assess the importance of this phenomenon for the design of shipboard nuclear reactor safety systems. The tests simulated low-energy collisions and the deformations of the idealised ship models were essentially elastic. Instrumentation was attached to both ship models in order to record several velocity and acceleration time histories during a collision.

An added-mass coefficient of 0.384 was found for the horizontal vibration of the struck ship model, which is similar to the expected theoretical value.

The striking ship behaved essentially as a rigid body throughout the response, while the struck ship experienced a rigid-body acceleration when the striking vessel was in contact, together with a significant horizontal (hull whipping) vibration mode. The maximum accelerations in the struck vessel did not occur at the location of the impact but were 70 percent larger at the end bulkheads (bow and stern for the model), which is consistent with the classical vibration modes of a free-free beam. Over 100 vibration cycles of the struck model were required for the water medium to reduce the horizontal vibration amplitudes by 50 percent.

It appears that the water supports the floating bodies essentially as a frictionless medium during the initial collision phase, when the maximum accelerations occur. Thus, any collision tests of constrained struck models conducted in air would not give valid estimates of the shock response for similar floating bodies.

The rectangular shape of the steel struck vessel roughly corresponds to the mid-ship section of a modern tanker. However, no attempt was made to properly scale the ship models so that the experimental results in Reference [53] cannot be scaled up to predict the shock response during a full-scale ship collision. Nevertheless, the experimental results are useful in identifying the major response features and in providing some data which might be used for checking numerical schemes.

3.5 West Germany

Woisin has continued his long association with the collision protection of nuclear-powered ships [10, 34, etc] and recently examined the characteristics of a resistance type of collision barrier [16, 54, 55]. Woisin [16] pointed out that a resistance type of collision protection scheme (grillage) occupies about 1/12 of a ship's breadth, whereas a conventional energy absorption deck structure requires at least 1/5 of a ship's breadth. Furthermore, a resistance type of protection barrier enables a vessel to remain seaworthy after a collision and can protect the vessel during a secondary collision. However, the weight and cost of a resistance type of protective scheme would probably be greater than a conventional energy-absorption type. Moreover the shock forces developed during a collision might be larger and possibly lead to failures in regions remote from the impact area, as suggested by the experimental results reported in Reference [53]. Nevertheless, Woisin [16] concluded that a properly designed resistance type of protection barrier is an attractive alternative for a nuclear-powered containership because of the severe width limitations imposed by the Panama Canal.

In Reference [16], Woisin reported the results of 8 model experiments with scale ratios of 1:7.5 and 1:12 using the same experimental arrangement as described in Reference [56], but with side structures having resistance-type barriers. Tests were conducted using four different kinds of bows: a typical bulbous bow for tankers and other bulk carriers, a typical cylindrical bow for bulk carriers and crude oil tankers, bow of the containership TOKYO BAY, and the extremely sharp bow of the liner FRANCE. In some cases, the bows were filled with ballast water. The side models were protected by three different types of resistance barriers so as to examine the influence of various structural dimensions.

Instrumentation was used to measure the impact velocity, deceleration of the carriage mass (with attached bow), impact forces, and strains. However, it transpired that the instrumentation frequently failed, or the recordings were not valid, so that none of these detailed results were presented in References [16, 54,55]. This is unfortunate because such data would be required to validate wholly numerical schemes. Nevertheless, the experimental results in References [16, 54, 55] are valuable and do demonstrate the feasibility of a resistance type of collision protection system, but further work is required to establish the accompanying shock forces.

Parallel to the experimental investigations of Woisin, Reckling [17, 57] developed an approximate theoretical procedure to examine the structural behavior of striking and struck ships during collisions. Reckling established that elastic effects were small and therefore the collision problem was almost entirely plastic. From an examination of Woisin's experimental results, Reckling distinguished three major types of damage: accordion-shaped folding of longitudinally stressed plating (e.g., outer hull of striking ship and deck of struck ship, etc), tearing open of longitudinally stressed plating where the collision opponent intrudes, and tearing open of laterally stressed plating due to large membrane strains (e.g., outer hull of struck ship). Reckling developed semi-analytical methods for calculating the energy absorbed in each of the structural members which responded to any of the three major types of behavior. The total energy which could be absorbed by the struck and striking ships in each of the failure modes was estimated and compared with the corresponding experimental results for two cases in Tables 1 of Reference [17] and [57]. It is evident from these results that Reckling's method fails to account for only 4.3 percent of the total absorbed energy in the first case and 3.7 percent in the second. This is a remarkable achievement, though it must be tempered with the fact that a number of assumptions have been introduced into the theoretical analysis which might not remain valid for other cases.

Reckling then examined a resistance type of protective grillage which was discussed by Woisin in Reference [16] for a nuclear-powered containership. In this case, the limit load of the protective system on the side of a struck ship must be larger than any impact load applied by the bow of a striking ship. Reckling calculated the instability loads of all the longitudinally stressed plating in a striking bow and added them to estimate the quasi-static impact force. These values agreed to within 10 to 20 percent of the impact forces measured during the experimental tests of Woisin, in which the bows were completely destroyed. The maximum forces expected from various full-scale bows according to this theoretical procedure are presented in Table 2 of Reference [17].

Reckling [17, 57] idealised the protective barrier on the side of the struck ship as a grillage and used the upper-bound theorem of plasticity to estimate the corresponsing quasi-static collapse load, which he found to be 1.5 to 2.5 times larger than the load required to cause initial plastic yielding. In addition, the failure loads of the decks and the plate elements were estimated and the smallest of the three calculated failure loads was taken as the ultimate collapse load of the side of a struck ship. Finally, by comparing the ultimate collapse load of the protective system for the struck ship with that required to collapse a striking bow allows a designer to judge the degree of protection afforded a particular ship.

4. Grounding of Ships

The various shipping accidents discussed on page 297 of Reference [58] clearly illustrate that grounding is not an uncommon event. Card [59] examined the casualty data from 30 tanker bottom grounding damage incidents in United States waters and observed that if all had been fitted with B/15 double bottoms, then 27 of them would not have caused pollution and 87% of the pollution would have been prevented.

Minorsky [46, 60] collected the statistics for 333† vessels over 6000 gross tons which grounded during the six-year period 1970-1975. It turned out that an average of 4 to 5 ships per thousand ships grounded each year in each of the weight categories 6000-10000 tons, 10000-30000 tons, 30000-60000 ton, and 60000 + tons. One-third of all these groundings were written off as a total loss although only 8 percent actually sank. Minorsky found the probability of grounding for a container ship on the New York-Rotterdam run was about 0.04 in its lifetime, that of a ship over 6000 gross tons on the U.S.-Persian Gulf trade about 0.004 and 0.01 for a VLCC on the same route.

In contrast to the extensive literature on ship collisions, very few articles appear to have been published on the local structural damage of ships sustained in grounding accidents.

† 333 Vessels in Reference [60] and 336 in Reference [46].

Coker observed that the structural failure of the stranded cargo vessel LOCHMONAR was initiated at a discontinuity in the hull plating [61]. Coker idealised the situation and conducted an experimental photoelastic investigation into the pure bending behavior of a beam of depth D which changes to a depth d with a circular fillet at the discontinuity. These experimental results demonstrated the presence of important stress concentration effects at such sections which should be avoided in ships by redesign or the area strengthened if unavoidable.

In an interesting paper, Thomson [62] examined the circumstances of seven stranded vessels and calculated the stresses in the deck plating when instability occurred.

The authors of Reference [63] investigated the behavior of a bow of a ship which struck a rock that penetrated horizontally the bottom structure. In some tests, a rigid wedge was forced in an in-plane sense into the outer plating of a 1/4 scale steel model of a ship double bottom. In other experimental tests, a rigid wedge was forced into the entire double-bottom structure in order to simulate a grounding accident in which a rock or other obstruction tore through the entire double bottom.

In addition to the above in-plane tests, the authors of Reference [63] also conducted some experiments on 1/8 scale double-bottom steel structures subjected to lateral (i.e., transverse) loads in order to simulate the grounding or stranding of a ship on a rock. The loads were applied through solid conical protruders which were forced laterally (i.e., transversely) into the outer plating of a double bottom travelling horizontally.

The authors observed from their rock-striking experimental tests that crushed (buckled or torn) members such as outer and inner bottom plating, longitudinals, and girders, collapsed at 80 percent of the associated yield load, approximately. On the other hand, deflected members, such as floors, develop membrane forces and rupture at about 20 percent elongation (for mild steel). The same situation prevails in the case of grounding on a rock except the crushed members are now floors and girders, while the membrane members are the bottom plate panels supported by the floors and girders.

The authors of Reference [63] estimated on the basis of these tests that a rock would penetrate 0.2 to 0.5 of the ship length of a fully laden 100,000 ton oil carrier with a double bottom when travelling at normal speed. If the oil carrier stranded on a rock, then the bottom plate would rupture at a force of 900 ton and the rock would reach the inner bottom plating at 3600 ton.

The authors of Reference [64] investigated the theoretical and experimental behavior of a double bottom when stranded on a rock with a cylindrical shape. An experimental test was conducted on a 1/7 scale model of a double bottom which was loaded through a solid cylindrical protruder with an axis perpendicular to the bottom plating. A theoretical analysis was also developed in Reference [64] and found to give good agreement with the corresponding experimental results.

Vaughan [65, 66] has developed recently a simple semi-empirical procedure to estimate the damages associated with the collision and grounding of ships. Minorsky's [3] approximate procedure was developed for major ship-ship collisions with large volume distortions of the structure. On the other hand, grounding incidents may involve relatively little volume distortion but significant plate tearing which cannot be examined using Minorsky's method.

Vaughan [66] presented a dimensional analysis of a plate which was penetrated along the mid-plane by a rigid wedge and observed that the work done (equation (2.6)) consisted of two parts. One part was related to the volume of distortion as in Minorsky's work [3], while another part was proportional to the total area of fracture of tearing of a plate. Equation (2.7) of Reference [66] gives a simplistic formula for the energy absorbed by a ship bow which is crushed as it impacts against a side structure. The two equations (2.6) and (2.7) involve three unknown constants which were determined using the empirical formula of Minorsky [3] and the experimental results on idealised models reported by Akita and Kitamura [35].

It is evident from Figure 4 of Reference [66] that Vaughan's equation (4.1) agrees quite well with Minorsky's formula [3] for the penetration of a side structure by a rigid bow. However, Minorsky's formula is not appropriate when damage consists primarily of torn plating with negligible volume distortion which may occur when a ship runs aground on a sharp reef or on an ice projection. Vaughan [65] examined the grounding damage of a new design for a 107,000 ton displacement LNG carrier which collides with a fixed sharp object and found that the safe operating speeds associated with a tear length of 8.4 m. are only 1.5 to 2.8 knots depending on the geometry of the obstacle.

5. Current Ship Collision and Grounding Work

An attempt is made in this section to review the theoretical and experimental investigations on ship collisions and grounding which are currently being conducted or contemplated in various research centers around the world. The author would be grateful for communications from anyone reading this Section who is aware of any studies omitted.

5.1 Canada

Vaughan [67] is continuing his recent investigations into the grounding damage of ships which he initiated in References [65] and [66]. The tearing strength of plates is being studied more closely with the aid of experimental tests on mild steel plates of various thicknesses which are penetrated in an in-plane sense by sharp rigid wedges. The experimental results appear to support the main simplification introduced in References [65] and [66] which involves splitting the total work done into one part due to tearing and another part associated with bending. In order to reveal the dependence on the plate thickness, Vaughan [67] in this case assumed that the work done due to tearing was proportional to the penetration distance, while the bending energy was taken to be proportional to the deformed area (equation (2.1)).

It appears that Vaughan [67] intends to conduct further experimental tests in which the plating has to be pierced before it can be cut.

5.2 Denmark

Pedersen [68] is currently developing a finite-element numerical scheme to examine the behavior of a linear elastic rectangular beam (idealised ship) which impacts at right angles with another beam. The impact forces and duration of impact are being sought to determine when a collision is quasi-static or dynamic. A somewhat related problem has been examined by Garnet and Armen [69] who also used a finite-element procedure but to examine the mechanics of impact and rebound of an elastic linear work hardening rod which hits a rigid wall at right angles. However, Pedersen [68] intends to develop further his numerical work to examine the dynamic response when both beams are floating in water in a manner similar to the experimental arrangement in Reference [53] which was discussed in Section 3.4. It is also intended to include non-linear springs at the impact location in order to cater for realistic impact forces.

5.3 France

Loisance [70] has recently published a brief review on ship collisions. The author made some calculations using Minorsky's method [3] for a 120,000 m^3 methane carrier and compared the predictions for the critical speeds of various striking ships with the Japanese rules which provide a lower bound as shown in Figure 2 of Reference [70]. It appears that future articles will examine the statistical aspects of ship collisions and resistance type of collision protection schemes.

5.4 Greece

Manolakos [71] is currently preparing a literature survey on ship collisions and is examining the potential role of plasticity theory in ship collision studies.

5.5 Italy

It appears that no ship collision protection studies have been conducted in Italy since the publication of Reference [72] in which it was shown that the NCRE calculation method [41] mentioned in Section 3.3 gave good agreement with their experimental results when due allowance was made for the energy absorbed in a striking bow.

5.6 Japan

Further work is being conducted in the study on the collision protection of both ship hulls and bridge piers which was reported in Reference [39] and discussed in Section 3.1. Experimental tests are currently being performed using curved plates which represent scale models of tanker bows [73].

The collision protection of floating oil-storage tanks is also being investigated at the Ship Research Institute [73]. Scale models of the side structures of offshore oil-storage tanks are being loaded statically by a rigid wedge which simulates the bow of a striking ship.

5.7 Norway

Larson is currently exploring various energy-absorption schemes made from rubber which could be used to protect offshore platforms during ship collisions [74].

Det Norske Veritas has initiated a research project "Impacts and Collisions Offshore" sponsored by the Royal Norwegian Council for Scientific and Industrial Research. As part of this project, Hysing [75] recently investigated the force-deformation and energy-absorption characteristics of a ship hull which impacts a leg of an offshore platform. The struck object was assumed to be rigid and the indentation of a hull constant over the entire height of the ship's side. Additional assumptions and simplifications were introduced in order to obtain the individual characteristics of the transverse web frames, side, deck, and bottom plating which were then used to estimate the striking load and generate the load-penetration and energy absorption curves. The calculation procedure of Hysing [75] follows the spirit of the approximate methods developed in references [40] and [57].

Hysing [75] compared his approximate theoretical predictions with the corresponding Japanese experimental test results conducted on idealised models having vertical rigid bows which were reported in Reference [33, etc]. Generally speaking, the comparisons between the theoretical predictions of the load-indentation behavior and the corresponding experimental results in Figures 16 to 21 are quite reasonable for this type of problem. The energy absorbed by the idealised models in Figures 22 and 23 during indentation also agrees reasonably well with the corresponding experimental results presented in Reference [33, etc].

5.8 Poland

It appears that no studies on ship collisions or grounding are being conducted currently in Poland [76].

5.9 The Netherlands

The study on the collision protection of LNG tankers in Reference [40], which is discussed in Section 3.2, is part of a larger project which includes investigations into the spreading and evaporation of burning LNG spills on land and water and the heat radiation from LNG fires on land and water [77, 78]. In addition, risk assessments of various aspects of LNG transportation and storage have been explored in the Netherlands [79,80].

The general method which was developed in Reference [40] is currently being used to examine the collision resistance of a new design of LNG carrier [81]. In addition, a project on the safety of small boats which transport hazardous materials on inland waterways has just commenced [81].

5.10 United Kingdom

As far as the author is aware no studies in the ship collision field have been conducted in the United Kingdom since the publication of Reference [42] as noted in section 3.3.

5.11 United States

Minorsky is continuing his work on ship collisions and is using the approximate method of Reckling [17, 57] for the collapse strength of ship bows to predict the maximum impact forces which act on struck ships [82].

In Reference [83], which has recently become available, Minorsky demonstrated that reasonable values of the impact forces and energy absorption can be predicted using the empirical formulae of Gerard for an experimental test on a model bulbous bow of the ESSO MALAYSIA which was conducted by Woisin [54]. However, Minorsky found that the measurement of the impact forces during the test was not satisfactory and observed a vibration of the girder which supported the side model.

Minorsky [84] has used Reckling's method [57] to examine the collapse resistance of a model cylindrical bow of the OBO TARIM which was also tested by Woisin [54]. However, it was found that the area of the impact force-bow penetration curve was 39 per cent less than the energy expended during the actual test. Minorsky suggests that many structural members continue to fold or tear after collapse thereby contributing to an additional resistance.

As remarked in Section 3.4, Chang has presented in Reference [49] some preliminary numerical finite-element results on the collision protection of nuclear-powered ships. This numerical work is continuing and the numerical results are being compared with the experimental tests of Woisin [16] on resistance type protective grillages.

Three reports prepared by Chang have become available recently [85-87]. The first report [85] contains a simplified version of the input instructions for the finite-element computer program ANSYS when used for the elastic-plastic behavior of a grillage. Chang uses the numerical scheme ANSYS in the second report [86] in order to study a resistance protection barrier for a nuclear-powered vessel. Chang simplified this difficult problem by calculating the static response associated with a few peak values of the load-time histories of the scale models tested by Woisin. It was found that the stress field in the barrier tested by Woisin was always below the corresponding yield stress. This led Chang to claim that collapse would not occur according to the limit-analysis theorems, which, as remarked in Section 3.4, were developed for the infinitesimal-deflection behavior of elastic perfectly plastic structures loaded statically. The third report [87] essentially contains a more complete rendering of the work already presented in Reference [49].

The experimental work of Pakstys, Konigsberg, and Sheets is complete and the author understands that Reference [53] is the final report for this project.

Van Mater [88,89] is currently studying the available theoretical and numerical procedures which have been developed to predict the structural damage during minor ship collisions. It was also intended to seek a well-documented case of an actual minor ship collision in order to evaluate the accuracy of the various theoretical methods. However, although 728 ship collision cases were examined, it transpired that none of the minor collisions without rupture had sufficiently well-defined data with large enough indentations to be of interest. This study is continuing and Van Mater [89] is currently examining in depth the approximate theoretical scheme of References [12, 47, 48] for minor collisions and the numerical finite-element work of Chang [49], both of which were discussed in Section 3.4.

5.12 West Germany

It appears that the experimental ship collision research program of Woisin [16, 55, 90] has been terminated [91]. Reckling [91] is currently examining the impact of ships including the influence of torsional effects and vibrations.

6. Discussion

6.1 Experimental

It is evident from the foregoing literature review that many more experimental investigations are required before the ship collision problem is fully understood. Basic experimental studies must be undertaken in order to reveal various features of fracture mechanics, buckling and post-buckling behavior, constitutive equations, strain rate effects, etc, but these are not considered henceforth. In addition, further experimental tests are required on idealised models of ships or ship sections in order to clarify the mechanics of ship collisions, while full-scale tests are necessary for final verification.

The full-scale results could be obtained either by conducting full-scale collisions on old ships immediately prior to scrapping, or by using data collected from actual ship collisions. Unfortunately, vital data is often missing from the records of actual ship collisions [88, 89] which is similar to the situation for slamming and ice damage [25, 51]. Full-scale collisions on old ships would be an expensive but worthwhile exercise provided a team of active structural mechanists, structural naval architects, and hydrodynamicists oversaw the entire project to ensure that meaningful and complete data for future as well as current use was obtained. This team of specialists should come from outside the establishment responsible for the experimental tests and should assist in the preparation of interpretive reports which properly condense the raw data (which should also be published as a report by the original contractor). One struck ship could be used for several minor collisions and at least one major collision since both sides of a hull are available for testing. In addition, if the bottom of the struck ship was not damaged during the collision tests, then a couple of grounding or stranding experiments could be performed similar to the type examined on idealised scale models in References [63] and [64]. Furthermore, it is imperative to test idealised models of the same ship which is used for the full-scale ship collision and grounding tests in order to provide reliable information on scaling.

It is unrealistic to attempt to resolve all aspects of the complicated ship collision problem with full-scale tests. Thus, it is necessary to conduct experimental tests on idealised models to explore various characteristics of ship collisions e.g., influence and destructive capabilities of different types of bows, obliquity of ship collisions, impact locations, different ship constructions and types, different energy-absorption systems (absorption, resistance, honeycomb, hybrid, etc), minor and major collisions, etc. The available literature in the ship collision field has examined several of these aspects but much work still remains to be done. Furthermore, it is important to record the shock response during any experimental tests since only one experimental study has examined this important topic. Incidentally, it should be noted that experimental tests on constrained models in air rather than freely floating models in water might give rise to misleading shock test results [53] as remarked in Section 3.4.

It becomes quickly apparent from the foregoing comments that a wholly experimental approach to the ship collision problem either using full-scale vessels or idealised models is not realistic and therefore various methods of analysis must be developed with occasional fully instrumented experimental tests to provide checks on the theoretical predictions.

6.2 Numerical

It was remarked in the previous Section that it is not feasible to obtain all the required information on the many facets of ship collisions from experimental tests on full-scale vessels or using idealised models. Thus, recourse must be made to numerical methods which are considered in this Section and theoretical methods which are discussed in Section 6.3.

There are two major types of numerical studies: either computational procedures which automate the approximate theoretical methods developed in References [12, 17, 40, 47, 48, and 57], or wholly numerical studies using finite-elements, finite-differences, or some other scheme to integrate numerically the differential equations which govern the response. This section focuses on wholly numerical schemes, while approximate theoretical methods are considered in the next section.

A wholly numerical scheme for a ship collision is certainly not a trivial exercise. Apart from the wide variety of scenarios referred to in the Introduction and in Section 6.1, a numerical method must cater for elastic-plastic-strain-rate sensitive structures which suffer large deflections, large strains, buckling, and fracture. In addition, both the striking and struck ships must be discretised spatially as well as temporally and the impact force-time history is, of course, unknown a priori since it is part of the solution. Moreover, the contact area between the bow of a striking ship and the side of a struck ship is time-dependent and the fluid surrounding the ships must be properly idealised.

It is worthwhile at this juncture to make a few cautionary remarks about the development of numerical methods in plasticity. The form of the multidimensional constitutive equations for elastic-plastic materials is still not clear, even for static problems. This shortcoming was discussed at a workshop [92] organized as the result of unexpected findings which were recently revealed when comparing some numerical results generated with different constitutive equations. The role of transverse shear forces on the plastic yielding of structures is another contentious topic [93, 94]. Moreover, it is also important to carefully monitor the convergence of elastic-plastic problems. Belytschko and Hodge [95] examined a plane-stress problem which some previous authors had solved using a finite-element method, and observed that the predicted collapse load was significantly larger than it should have been, due to accumulation of errors in the numerical scheme. The dynamic fracture of structures is yet another topic which requires considerable further study in order to develop reliable failure criteria [27].

Thus, it is not possible at this point in time to obtain an exact numerical solution of the actual ship collision problem because some of the more basic mechanics are not yet fully understood. Therefore, the results of wholly numerical schemes have to be viewed with caution. However, it may be worthwhile to introduce various simplifications (e.g., strain-rate insensitive constitutive equations, rigid bows, etc) and generate wholly numerical solutions for these idealised situations.

6.3. Approximate Theoretical Methods

Approximate methods for the structural analysis of ship collisions have been developed in References [12, 17, 33, 37, 38, 40, 47, 48, 57, 75]. These theoretical procedures use somewhat similar approaches in the sense that they all idealise the structures participating in a ship collision as a collection of simpler components with a known structural response. In some cases theoretical formulae are available for the components, while in others, empirical formulae must be used. This approach is attractive because all the ship structures involved in the various ship collision scenarios referred to in the Introduction and in Section 6.1 may always be resolved into a number of standard structural elements.

The approximate theoretical methods in References [12, 17, 33, 37, 38, 40, 47, 48, 57, 75] are by no means definitive because some consider rigid-striking bows and make various other simplifications. Nevertheless, it appears that the general approach is a fruitful one since it is capable of further development and refinement as new information becomes available.

Numerical results obtained on a computer with such methods would cost only a fraction of those from a wholly numerical scheme, while the degree of accuracy possible is probably adequate from an engineering viewpoint. Indeed, Reckling [17, 57] has already predicted the magnitude of the bow forces which act on a struck ship during a collision within 10 to 20 percent of the corresponding experimental value as remarked in Section 3.5.

If energy is absorbed in both a striking bow and the side structure of a struck ship during a collision, then the methods in Reference [12], for example, require a simple iterative scheme because the partition of the total energy between a struck ship and a striking bow is unknown. The energy absorbed in the side of a struck ship and in a striking bow and the associated quasi-static impact force may be calculated for an assumed amount of damage. The amount of damage in a bow and a struck side must be iterated until the total energy absorbed equals the kinetic energy lost during a collision and the quasi-static force on a striking bow and a struck side are equal.

Alternatively, the individual quasi-static impact force-penetration characteristics of a striking bow and a struck side of a ship could be obtained by calculating the quasi-static loads required to produce penetrations up to some arbitrary maximum value. The associated energy absorbed at each increment of the penetration could be calculated in order to construct the individual energy-absorption-penetration characteristics for a bow and a ship side. A straightforward graphical procedure could then be used to equate the impact force in the bow and a side and to ensure that the total energy absorbed equalled the kinetic energy lost during a collision. In fact, it is possible to generate the individual quasi-static impact force-penetration and energy-absorption-penetration characteristics for a large number of standard bows and ship sides. Thus, a designer could then simply use a graphical procedure in order to estimate the damage sustained by a particular ship for a given loss of kinetic energy when it is struck by a variety of standard bows.

The theoretical predictions of these approximate methods should be continuously monitored and compared with the results of full-scale ship collisions (experimental and actual) and with tests on idealised models. In addition, the approximate theoretical predictions could also be compared with the results of wholly numerical schemes for idealised ship collision scenarios to seek any important errors and to establish the range of validity.

Simple approximate theoretical methods have also been developed in References [39], [63], and [64] for the collisions of ships with bridge piers and for the grounding and stranding damage of ships. These approximate methods have predicted reasonable agreement with the corresponding experimental results. However, various simplifications have been made which require further verification before the approximate methods could be programmed and used with confidence in design.

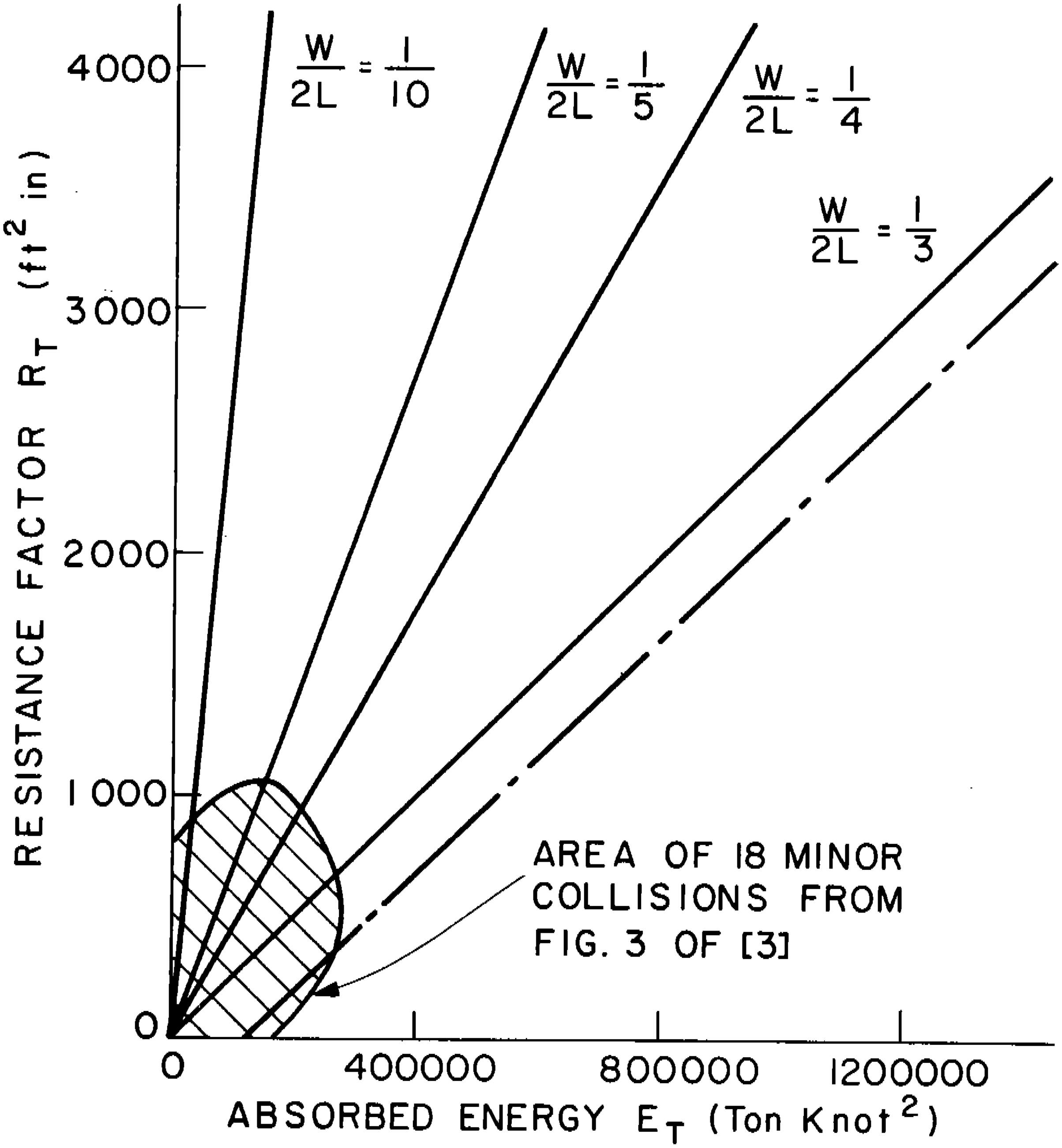
6.4 Simple Design Methods

It appears that the wholly numerical schemes discussed in Section 6.2 and the approximate theoretical procedures in Section 6.3 would be too time-consuming and too expensive for preliminary design and therefore more suitable for the final design stage. On the other hand, Minorsky's [3] simple semi-empirical method is very attractive for preliminary design. This approximate procedure was developed from the data associated with a number of actual major ship collisions so that it does not suffer from any difficulties with scaling. It agrees surprisingly well with the experimental results of Woisin in Figure 21 of Reference [10] and is not too dissimilar from the approximate theoretical predictions for crack or encroaching behavior (defined in Figure 4) in Figure 10 of Reference [33]. However, different factors may be required in the semi-empirical relation to account for striking ships with very strong bows (e.g. icebreaker) or for other conditions not prevailing two decades ago when Minorsky's method was developed.

The Naval Construction Research Establishment [41] derived another simple design procedure for major collisions and Belli [72] observed that it gave surprisingly good predictions of the experimental test results when the energy absorbed in a striking bow was considered.

It would appear worthwhile to further refine these two simple methods [3, 41] for predicting the overall characteristics of major collisions.

Unfortunately, no simple approximate procedures similar in spirit to those in References [3] and [41] appear to be available for minor collisions. The theoretical work in References [12] and [37] has shown that a significant portion of the kinetic energy lost during a minor collision is absorbed due to the development of membrane forces in the hull plating, while References [12], [17] and [57] have demonstrated that the energy absorption due to material elastic behavior is small. These observations led to the suggestion in Reference [96] that the energy dissipated in the hull and deck plating during a minor ship collision (E_T) could be estimated using the load-lateral deflection response for the membrane behavior of a rigid perfectly plastic beam. It is shown in Appendix 1 that this procedure gives



——— EQUATION (2) WITH σy = 30000 lb/in^2

—·— MINORSKY [3]

FIGURE 6

$$E_T = 0.030288\ \sigma_y\ (W/L)^2\ R_T\ ,\ (\text{ton-knot}^2) \qquad (2)$$

where σ_y is the uniaxial yield stress (lb/in^2), 2L(in.) is the span of a member (deck or hull plating), W (in) is the maximum transverse displacement or damage of a member (without rupture) while R_T (ft^2 in) is defined by equation (1.5) and is the volume of material in the participating members which is similar to the resistance factor R_T used by Minorsky [3].

The theoretical predictions of equation (2) for minor ship collisions with various amounts of damage (W/2L) are compared with Minorsky's formula in Figure 6. The plating in the tests which were reported in Reference [12] would rupture when $W/2L \simeq 1/3$ (see also Reference [24]) for a right-angled collision at mid-span. It is evident from Figure 6 that the theoretical predictions of equation (2) with W/2L=1/3 are similar to Minorsky's formula which was developed for major collisions.

The theoretical predictions of equation (2) with W/2L<1/3 give a family of lines which radiate from the origin of Figure 6 and traverse the entire area which contain minor or low energy collisions according to Minorsky [3]. Thus, it is necessary to specify the damage W in a minor collision in addition to the parameters R_T and E_T. Once the amount of acceptable damage W is specified for a minor collision, then an acceptable design associated with a value of R_T could be found for a given required energy absorption E_T. This is, of course, a very simple scheme which clearly requires further justification and development. Nevertheless, it is hoped that it may eventually form the basis of a simple design procedure for minor ship collisions.

It is shown in Appendix 1 (equation 1.8) how the proposed approximate procedure can accommodate deck and hull members with various spans and thicknesses which sustain different amounts of damage during a minor ship collision. Different designs according to equation (1.8) could now be distinguished in a plot such as Figure 6 by associating each design with an average value of damage $\bar{W} = \sum_{i=1}^{N} W_i/N$, or a maximum damage $W_m = \max.\ W_i$.

Van Mater [97] has extended recently the theoretical method in Reference [96] to cater for an applied load acting at an arbitrary location on the span of a beam and has explored the possibility of using other definitions for R_T.

Arita et al [38] have developed a simple empirical formula in order to predict when a ship collision may be classed as a minor or a major one in the sense that minor collisions would be governed largely by buckling or deformation modes and major collisions would be controlled largely by encroaching or crack modes which are illustrated in Figure 4. A simple method to evaluate the dynamic inelastic fracture of beams is presented in Reference [27]. The satisfaction of the criterion developed in this article would ensure that a collision remains a minor one with large ductile deformations of the type examined in References [7] and [25] without fracture.

The value of simple design methods such as those discussed in this section cannot be overemphasized. For example, the availability of a simple design scheme could have obviated the situation found in Reference [48] in which the full energy-absorption potential in certain collision situations could not be realised because the LNG tanks are too close to the hull.

7. Recommendations for Future Work

7.1 Introduction

It is evident from the literature survey in this report that there are many possible directions for future research work in the ship collision field. A resistance type of protective system may be required to protect a nuclear-powered ship, whereas a honeycomb absorption system may be necessary for a high-speed marine vehicle in order to satisfy the stringent weight restrictions. Moreover, it may be necessary because of overriding safety considerations to conduct full-scale tests on the actual protective grillage of a nuclear-powered ship, whereas approximate theoretical methods with some supporting experimental tests might be adequate for many conventional and non-conventional vessels. Classification societies, insurance companies, and the public have special demands, while the collision protection of ships striking bridge piers, ice, and offshore structures present unique problems. In addition, there is the question of shock loads which has received scant attention as remarked in Section 3.4.

Thus, there are many research problems and diverse viewpoints associated with the various ship collision scenarios mentioned in the Introduction. It therefore appears more rewarding to seek the underlying principles of behavior rather than attempt to solve the many special cases. In this manner, a long-range research program could contribute to a better understanding of the mechanics of ship collisions which in the long term would be less expensive because unnecessary experimental testing or other work could be avoided. In other words, a complete understanding of the basic structural mechanics of ship collisions would enable all the special cases to be solved. The ultimate goal of this research program is a body of knowledge on different types of energy absorption, resistance, and hybrid protective systems together with accompanying theoretical methods suitable for design. These theoretical methods should have been verified by occasional well-instrumented full-scale and model testing and should be made available as computer programs which are sufficiently versatile to handle many collision scenarios. Many years will pass before these halcyon days are reached, but the time will be hastened by ensuring that all the ship collision projects around the world are carefully monitored and are mutually exclusive unless there is a good reason for duplication.

7.2 Ship Collisions

The approximate theoretical methods which were developed in References [12, 17, 33, 37, 38, 40, 47, 48, 57, 75] and discussed in Section 6.3 are very attractive candidates for further developement because they idealise a ship structure (bow and/or side structure) as a collection of simpler elements with a known structural response. The ship structures involved in the various ship collision scenarios referred to in the Introduction and in Section 6.1 including collisions with ice, bridge piers, and offshore platforms may always be resolved into a number of standard structural elements. Moreover, this general approach is a fruitful one since it is capable of further development and refinement as new experimental and theoretical information is generated. In fact, the philosophy of approach is not too dissimilar to that developed by Hughes and Mistree [98] for the automated optimisation of ship structures.

The theoretical work that has already been accomplished in References [12, 17, 33, 37, 38, 40, 47, 48, 57, 75] on simple approximate methods should be incorporated into one procedure such as that in References [17] or [48] (see also Reference [99]) or in Reference [40] and a simple computer program written to eliminate the bookkeeping required in the analysis of a particular problem. Some of the assumptions and simplifications in these methods could be eliminated since they are unnecessary for a numerical version. This procedure would be capable of catering for a broad range of striking bow designs and therefore properly partition the lost kinetic energy between the striking and struck ships mentioned in Section 2.4 as well as recognising the important influence of bow rake which was discussed in Sections 2.4, 3.1, and 3.4. Moreover, the suggested procedure could accommodate a variety of absorption, resistance, and hybrid side structures. Incidentally, the author noted in Section 2.2 that it is worthwhile exploring the possibility that a hybrid protection system may weigh less than a resistance structure and have smaller associated shock forces if the absorbing part is designed as an attenuator. Indeed, this is just one of many ship collision protection systems which should be examined with the ultimate objective of providing naval architects with a broad range of protection schemes to match the variety of collision scenarios.

The general theoretical method described above would not provide any information on the shock response of struck and striking ships during a collision. However, if further studies on this topic beyond those discussed in Sections 3.4 and 5.1 and Reference [100] demonstrated that the shock response must be considered in ship collision protection schemes, then theoretical methods would also have to be developed to examine this aspect, perhaps using beam theories. It should be noted that shock tests conducted on constrained ship models in air might give misleading results because the authors of Reference [53] (see Section 3.4) found that the water supporting a floating model essentially acted as a frictionless medium during the initial stage when the maximum accelerations occurred. Furthermore, the impact force generated in a side structure by a striking raked bow is significantly less than that associated with a striking vertical bow according to the experimental results on Reference [39] which are discussed in Section 3.1. If the local structural damage of a ship in the region of a striking bow is essentially quasi-static (i.e., long-term response [30]) then the associated force-penetration characteristics may be calculated using the approximate theoretical procedure discussed previously (e.g. extension of References [17, 48]). The maximum value of this force could then be used as input into a beam analysis in order to generate the approximate shock response associated with horizontal flexural motions of the struck ship. On the other hand, if dynamic effects exercise an important influence on the local structural damage in the vicinity of a striking bow, then the approximate theortical method (e.g. extension of Reference [40]) must be coupled with a beam analysis. The ideal manner to tackle this problem is unknown because more experimental and theoretical investigations are required in order to establish the importance and characteristics of shock in both struck (primarily horizontal or transverse motions) and striking ships (primarily longitudinal motions for a right-angle collision).

The writer remains to be convinced of the value of undertaking a wholly numerical approach to the actual ship collision problem at this point in time for the reasons stated in Section 6.2. Moreover, the cost of a wholly numerical scheme would probably be comparable to the cost of a full-scale collision testing program using old ships. Full-scale experimental results would possibly have greater value and more impact on the ship collision field than wholly numerical results. However, it may be worthwhile to obtain some exact numerical solutions of a few idealised situations in order to guide the development of the approximate theoretical procedures.

If full-scale collision tests are to be undertaken then one fully instrumented ship could be used for several minor collisions at different bow impact locations relative to the transverse bulkheads and various angles of obliquity and one or two major collisions in addition to a couple of grounding or stranding tests as remarked in Section 6.1. Some experimental tests should also be conducted on fully instrumented scale models of the actual ship to examine the influence of scaling and to assess the accuracy of results obtained using scale models. The scale-model tests should, preferably, be completed prior to the full-scale experiments and the results made available in order to assist in the wisest selection of collision cases. It was remarked in Section 6.1 that the data currently available in the records of actual ship collisions are usually incomplete from a structural mechanics viewpoint and are therefore of limited value to a structural analyst. However, this could be a very fruitful and relatively inexpensive source of information if more complete records of actual ship collisions could be gathered and put on file.

Further development of the simple design methods discussed in Section 6.4 and Appendix 1 would appear worthwhile because of their value in preliminary design.

7.3 Grounding and Stranding of Ships

It was remarked in Section 4 that there is a paucity of published papers on the structural aspects of the grounding and stranding of ships. However, the recent Japanese experimental and theoretical investigations [63, 64] which are discussed in Section 4 offer a good start for future work in this field. This theoretical work requires further refinement and additional experimental tests are needed before the method should be programmed for design use. The recent investigations of Vaughan [65-67] provide some valuable information on the tearing of mild steel plating which may occur during grounding incidents.

It was noted in Section 6.1 that any full-scale collision tests on old ships should include a couple of grounding or stranding tests in order to verify the existing theoretical methods and to guide the development of future theoretical work on this topic. In addition, some grounding and stranding tests should be conducted in a laboratory on scale models of the full-scale ships in order to provide reliable information on scaling since most future experimental work in this area would be conducted with models rather than full-scale tests.

7.4 Summary Remarks

It is evident from the foregoing discussion that much work remains to be accomplished before reaching the ultimate goal of a design tool in the form of a computer program with various options which are capable of examining any kind of ship collision including grounding and stranding with a known accuracy. Thus, there is a need for several simultaneous research investigations into various aspects of the multi-faceted ship collision problem:

(i) Develop an approximate theoretical scheme and write a computer program using References [12, 17, 33, 37, 38, 40, 47, 48, 57, 75, 99] as a starting point (see Sections 6.3 and 7.2).

(ii) Organise a full-scale ship collision and grounding experimental testing program (see Section 6.1 and 7.2).

(iii) Execute an experimental ship collision program using fully instrumented scale models of the full-scale ships to be used in research program (ii). These tests should be completed prior to the commencement of the actual tests in research program (ii) (see Sections 6.1 and 7.2).

(iv) Conduct some experimental grounding tests on fully instrumented scale models of the full-scale ships to be used in research program (ii). These tests should be completed prior to the commencement of the actual grounding tests in research program (ii) (see Section 6.1 and 7.3).

(v) Determine the potential importance of the shock response in various types of ship collisions. This study could be conducted in close cooperation with the research group mentioned in Section 5.2.

(vi) Examine the feasibility of various types of ship protection systems (e.g. hybrid system mentioned in Sections 2.2 and 7.2 and honeycomb absorption system for marine vehicles discussed in Reference [11] and Section 2.2).

(vii) Develop an approximate theoretical procedure for grounding damage similar in spirit to the scheme in research program (i) (see Sections 6.3 and 7.3).

(viii) In addition to the above ship collision research programs, some fundamental experimental tests should also be conducted into the behavior of the basic structural elements which will be incorporated into the approximate theoretical method in research programs (i) and (vii). This work is required in order to provide more accurate information on the rupture, fracture, and post-buckling behavior of the structural elements than is available currently.

Once the full-scale tests in research program (ii) have been completed and the results analysed, then the status of all the above research projects could be properly assessed. The next step could then be charted towards the ultimate goal of a proven design tool in the form of a computer program (or possibly graphical results generated with the aid of the program). In the case of a ship collision, a naval architect would use the computer program to estimate the local structural damage in the side of a struck ship in the area of a striking bow and the associated shock damage (which may be near the bow or stern of a struck ship according to Reference [53]) sustained by a vessel which is hit by another ship or which strikes another object (e.g., ice, bridge pier, offshore platform, ship, etc). In the case of grounding, the computer program could be used to estimate the damage sustained by the bottom of a ship when striking a "standard rock" at a certain speed. The computer program could also be used to calculate the force required to puncture the bottom of a ship which is stranded on a rock.

In order to accomplish the above research program it is important to carefully monitor all the ship collision research projects around the world and to avoid duplication.

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APPENDIX 1

Extension of Minorsky's Method for Minor Ship Collisions

It is evident from Table 1 in Reference [12] and Table 4-2 in Reference [48] that the membrane energy absorbed in stiffened hull plating and in stiffened decks is important and is between 88 and 99 percent of the total energy absorbed in a struck ship during a minor collision. Furthermore, the energy absorbed due to elastic deformations was found to be small compared to that absorbed plastically as shown in References [12, 17 and 57]. These observations suggest that Minorsky's general approach [3] could be used for minor collisions but with the resistance factor developed using the equations for rigid-plastic beams loaded transversely into the membrane range.

Now, consider a rigid perfectly plastic beam with fully clamped supports across a span $2L$ which is subjected to a concentrated load P at the mid-span. Thus, equation (23b) in Reference [101] gives

$$P/\overline{P}_c = 2W/H \quad , \tag{1.1}$$

when $W/H \gtrsim 1$, and where

$$\overline{P}_c = \sigma_y BH^2/L \quad , \tag{1.2}$$

σ_y is the uniaxial yield stress, B is the beam breadth, and H is the beam thickness. If it is assumed that the membrane behavior occurs for all lateral displacements $W \lesssim H$ and $W \gtrsim H$, which underestimates slightly the total internal energy according to Figure 1 in Reference [51], then the energy absorbed by a beam with a lateral displacement W is

$$E_a = \int_0^W P \, dW \quad ,$$

which when using equations (1.1) and (1.2) becomes

$$E_a = \sigma_y BHW^2/L \quad . \tag{1.3}$$

Equation (1.3) may be recast in the form

$$E_a = 72 \, \sigma_y R_T (W/L)^2 \quad , \tag{1.4}$$

where

$$R_T = 2LBH/144 \tag{1.5}$$

is the volume of material in ft^2 in. units when B, H and $2L$ are measured in inches. If the same units as Minorsky [3] are used then

$$E_T = 0.030288\ \sigma_y\ R_T\ (W/L)^2 \quad , \tag{1.6}$$

where E_T is the total energy absorbed (ton-knot^2) , σ_y is the yield stress (lb/in^2), and R_T is defined by equation (1.5) (ft^2 in.), while W and L must have the same units (either in. or ft.).

Equation (1.6) with $\sigma_y = 30{,}000$ lb/in^2 is plotted in Figure 6 for various values of $W/2L$ and is compared with Minorsky's semi-empirical result [3].

The foregoing theoretical analysis was developed for a perfectly plastic material. If σ_u is the ultimate stress of the material, then a rough estimate of the influence of material strain hardening could be obtained by replacing σ_y in equation (1.6) by $(\sigma_y + \sigma_u)/2$ (see also Reference [24]), or

$$E_T = 0.015144\ (\sigma_y + \sigma_u)\ R_T\ (W/L)^2 \ . \tag{1.7}$$

Equations (1.6) and (1.7) were derived for a single beam, although the equations would remain valid for any number of similar beams which have the same amount of damage provided R_T corresponds to the total volume of the material. In general, however, different amounts of damage would be sustained by the various deck and hull members which could also have different spans and thicknesses. In this circumstance, equation (1.6) would be replaced by

$$E_T = \sum_{i=1}^{N} 0.030288\ \sigma_{yi}\ R_{Ti}\ (W_i/L_i)^2 \quad , \quad 1 \leq i \leq N \quad , \tag{1.8}$$

where σ_{yi} , R_{Ti} , W_i , and L_i are the values associated with each of the N individual members which participate in a minor collision. Different designs according to equation (1.8) could now be distinguished in a plot such as Figure 6 by associating each design with an average value of damage $\bar{W} = \sum_{i=1}^{N} W_i/N$, or a maximum damage $W_m = \max.\ W_i$.

It should be remarked that equation (1.1) and therefore equation (1.8) is also valid for stiffened plating provided R_{Ti} is interpreted as the total volume of material in the stiffener and the associated plating as remarked in Reference [24].

The corresponding version of equation (1.8) to account for material strain hardening in an approximate manner could be obtained from equation (1.8) by replacing σ_{yi} by $(\sigma_{yi} + \sigma_{ui})/2$.

It would be possible to account for the influence of in-plane displacements at the supports as suggested in Reference [24]. In this circumstance equation (1.1) here would be replaced either by equations (23) of Reference [101] for beams with rotationally fixed supports or by equations (22) of Reference [101] for those beams which could be considered rotationally free at the supports. However, if only the behavior in the membrane range were considered, then the expressions in this Appendix remain valid [24].

See pages 40-46 for References quoted in text.

APPENDIX 2

This appendix contains all known articles on ship collision protection from a structural viewpoint which are not referred to in the References section of the Report. An interested reader may find additional references on collision statistics and probabilities, collision avoidance, and other aspects of ship collisions in References [9] and [10], and some references on the static and dynamic plastic behavior of structures in References [50] and [51].

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