PHILOSOPHICAL TRANSACTIONS OF THE ROYAL SOCIETY A MATHEMATICAL, PHYSICAL & ENGINEERING SCIENCES

Measurement of Wave-Induced Loads in Ships at Sea [and Discussion]

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Measurement of wave-induced loads in ships at sea

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The problem of predicting wave-induced loads in ships is considered with particular reference to measurements at sea. The capabilities and limitations of theoretical analysis and strain measurements are discussed together with the role of each in the prediction of loads for design purposes. Some gaps in knowledge are identified and priorities for future research suggested.

1. Introduction

Design of a ship's hull is a complex process in which an intricate balance must be sought between the conflicting requirements of safety, space, weight limitation and minimization of procurement and through-life costs. An essential part of that process, on which design of the hull structure depends vitally, is prediction of wave-induced loading. In larger ships and particularly in long, slender ships of the frigate and destroyer type, the dominant form of load is wave-induced bending and shear of the hull girder. For design purposes it is necessary to predict (*a*) the maximum moments and shear forces which a ship's hull will experience during its life, ideally in terms of whole-life probability of non-exceedance; (*b*) the histogram of cyclic (including vibratory) loads which the hull will experience during its life and which will tend to cause fatigue failure.

As part of a limit state design procedure, estimates of maximum wave loads are now commonly related to an assessment of ultimate hull-girder strength to ensure a very low probability of hull collapse during the ship's life. Design against fatigue failure requires (Clarke 1987) (*a*) stress analysis of the hull structure with the aim of converting primary hull loads (bending moments and shear forces) into histograms of stress at any point: analysis must be sufficiently refined to determine stress levels at structural discontinuities such as hatch corners and welded joints, whose fatigue strength is defined by established endurance curves; (*b*) reference to empirically based fatigue endurance curves, e.g. those of BS5400 (1980) for steel structure or CP118 (1969) for aluminium, in conjunction with a cumulative damage (Miner's Rule) failure criterion to evaluate fatigue life.

To provide guidance to designers, the Admiralty Research Establishment at Dunfermline has conducted for more than 30 years a programme of research into the loads experienced by warship hulls and the strength of structure required to withstand these loads. Research on wave loading has included development and evaluation of theoretical analysis methods, participation in small-scale tank testing of ship models and short-term seakeeping trials on ships at sea, together with a long-term programme of strain measurement on ships during normal service. The purpose of the present paper is to describe some of this work and to discuss the capabilities

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and limitations of theoretical analysis and strain measurements and the role which each plays in the prediction of wave-induced loads for design purposes. Some inadequacies in the present state of knowledge are identified and some suggestions are made on priorities for future research.

2. Theoretical prediction methods

Static balance analysis

Maximum wave bending moments have traditionally been estimated by supposing a ship to be balanced statically on a wave whose length L equals that of the ship and whose height is a specified fraction of L (e.g. $L/20$). The static balance loading assumption, together with assumptions regarding permissible stress and safety factors and use of particular materials are interdependent parts of a design process, still used in some codes, whose adequacy derives from the safe performance of previous similarly designed ships. As discussed later, strain measurements on RN frigates and destroyers have shown that the maximum lifetime load is typically 1.2–1.7 times the $L/20$ value.

Rigid body dynamic response to regular waves

More realistic estimates of wave-induced loads, accounting for dynamic effects, have been sought by solving the equations of motion

$$M\ddot{\delta}+C\dot{\delta}+K\delta = F(t), \tag{1}$$

which in their simplest form contain two degrees of freedom corresponding to rigid body heaving and pitching motions (ISSC, Korvin-Kroukovsky 1955). Mass and damping coefficients M and C, determined at a number of stations along the ship, include both hydrodynamic and structural components, the former usually estimated assuming two-dimensional flow (strip theory) at each station. The exciting forces F, assumed to be harmonic, represent the action of regular, unidirectional sinusoidal waves of specified length and direction. Equation (1) may be extended to include three additional degrees of freedom corresponding to rigid-body sway, roll and yaw motions. Solution of this equation allows net forces to be determined at each station, which may be integrated along the ship to obtain vertical and horizontal bending moments and shear forces and torsional moments.

Response to irregular waves

Solutions may be obtained for a complete range of wave lengths and directions and ship speeds, defining 'response amplitude operators' (RAO) which relate motion and load amplitudes linearly to wave height. Irregular sea conditions may then be described in terms of directionally distributed wave energy spectra (ISSC, St Denis & Pierson 1953), from which response spectra follow directly: wave spectrum × (RAO)^2 = response spectrum. Most of the analysis carried out by ARE have used the ISSC two parameter formula (ISSC) to represent a unidirectional sea.

$$\phi(w) = (A/\omega^5)\exp(-B/\omega^4), \tag{2}$$

where $A = 173H_{\frac{1}{3}}^2/T_1^4$, $B = 691/T_1^4$, ω is the wave frequency (radians per second), $H_{\frac{1}{3}}$ is the significant wave height (metres) and T_1 is the characteristic period (seconds). Equation (2) gives the energy density per unit frequency interval. This is related to wave amplitude (a_0) by

$$\phi(w)\,\mathrm{d}\omega = \tfrac{1}{2}a_0^2. \tag{3}$$

In practice the sea will not be unidirectional because of the change in wind direction it will have experienced. The usual way of allowing for this is to use a spreading function $S(\mu)$ where μ is measured from the predominant wave direction such that

$$\phi(\omega, \mu) = \phi(w)\, S(\mu). \tag{4}$$

For many applications the relationship

$$S(\mu) = (2/\pi)\cos^2(\mu) \tag{5}$$

is adequate.

The probability distribution of wave amplitude for a given sea state generally closely approximates to a Rayleigh distribution, i.e.

$$P(a) = (a/m_0)\exp(-a^2/2m_0), \tag{6}$$

where m_0 is the mean square amplitude. Since in strip theory the response is assumed to vary linearly with wave height, this also follows a Rayleigh distribution. Measurements of bending moment from sea trials have given a good agreement with a Rayleigh distribution if slamming effects are ignored (Clarke 1982).

The probability of exceeding a given response at any time in the life of the ship can be calculated by combining the short-term statistical distribution described above with the mission profile of the ship (speeds V and headings θ) and wave statistics giving the probability of meeting specific conditions defined by significant wave height ($H_{\frac{1}{3}}$) and wave period (T_1).

The probability of exceeding a given response (e.g. vertical bending moment M_{v}) is thus given by

$$P(M_{\mathrm{v}}) = \iiiint P(V, H)\, P(\theta, H)\, P(H_{\frac{1}{3}}, T_1) \exp\left(\frac{-M_{\mathrm{v}}^2}{2M_{\mathrm{v}0}(V, \theta, H_{\frac{1}{3}}, T_1)}\right) \mathrm{d}V\, \mathrm{d}\theta\, \mathrm{d}H_{\frac{1}{3}}\, \mathrm{d}T_1, \tag{7}$$

where $M_{\mathrm{v}0}$ is the mean square amplitude of M_{v}, obtained by integrating the regular wave response over the wave spectrum.

Hydroelastic response

The analysis method outlined above may be extended to evaluate deformational (hydroelastic) hull responses (Bishop *et al.* 1977). The equations of motion then include up to five degrees of freedom at each station along the ship, flexural, torsional and shear stiffness coefficients for the hull being included with buoyancy terms in the matrix K. Solutions can be obtained efficiently by transferring hydrodynamic terms to the right-hand side of equation (1) and using a modal analysis which operates on the 'dry ship' vibration modes (ISSC, Bishop *et al.* 1977). The resulting responses include wave-excited vibration ('springing') of the hull-girder which in most ships does not significantly affect hull loading but which may, in long slender hulls with exceptionally low natural frequencies, contribute to fatigue loading.

Slamming effects

Theoretical analysis may be extended further to deal with the important problem of slam-induced 'whipping', i.e. vibratory response of a hull caused by impulsive forces on the bow which can cause large bending moments and shear forces in the midships region and along the forward half of a ship (Kawakami *et al.* 1977; Belik *et al.* 1980). Linear analysis of ship motions is used to predict conditions of bow emergence from the water and velocity of impact on re-entry. The resulting slamming forces have two components: (*a*) virtually instantaneous impact pressure

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Figure 1

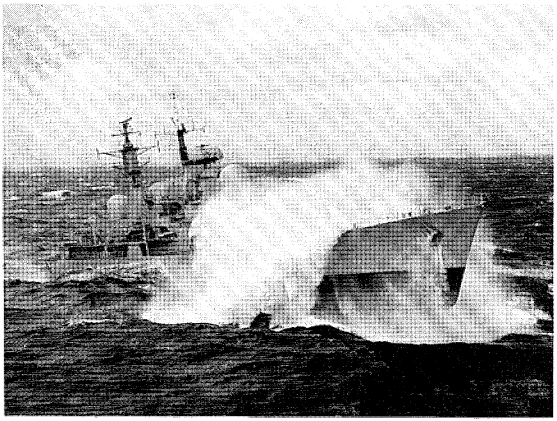

Figure 2

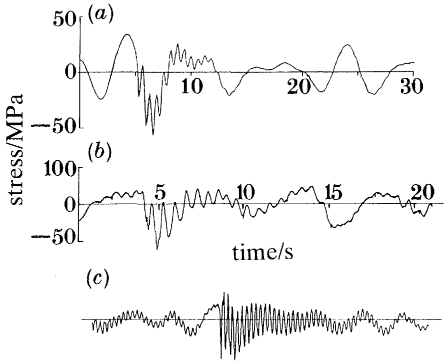

Figure 1. Frigate under slamming conditions.

Figure 2. Computed and measured slam-induced whipping stresses. (*a*) Computed response: frigate at 18 knots in head seas. (*b*) Measured response: same ship (more severe conditions). (*c*) Measured response: landing craft with low damping.

which is most severe in the case of flat or U-section bottoms and less severe in the case of V-sections, attenuated to some extent by air entrapment; (*b*) more extended, 'momentum-transfer' loading associated with rapid immersion of bow sections.

Because of the complexity of the slamming process, influenced by air entrapment, by irregularity of both water surface and structure and by local elastic response, these forces are usually estimated empirically using the results of drop tests on representative ship sections (ISSC). Slamming forces may cause local structural damage: crude but effective design methods of avoiding such damage have been devised assuming quasi-static loads based on analysis of observed deformation in ships bottoms. Of much more importance is the problem of slam-induced whipping, which contributes substantially to extreme hull-girder bending moments in slender, shallow-draught hulls (such as warships) and may also cause significant fatigue damage (Clarke 1986), particularly in hulls with low damping characteristics.

Figure 1 shows a frigate under slamming conditions and figure 2 shows some illustrative computed and measured slam-induced whipping stresses in a frigate and a landing craft.

3. Measurement on ships at sea

Extreme-value strain measurement

In 1963 Yuille (1963) proposed a method of extreme load prediction based on measurement of maximum strains on ships during normal service. Measurements were made using an extreme-value mechanical extensometer developed by ARE (then NCRE), in which maximum strain excursions during successive 24-hour periods were recorded as scratch marks on a static roll of paper, wound forward daily by a crew member. Records obtained over many months were analysed statistically and extrapolated to provide estimates of lifetime maxima using a method described by Gumbel (1959).

An improved version of the ARE maximum-reading strain recorder was then developed (Smith 1966, 1976), incorporating an automatic winding mechanism and clock control designed to move the recording paper forward at 4-hour intervals (taken as the maximum during which operating conditions could reasonably be assumed constant). 115 of these instruments have been made and installed on a total of 68 RN warships, representing most classes, during the past 25 years. Strain records are divided into hogging and sagging components by plotting an estimated mean line through each record, using small excursions and the marks made when the recording paper is wound on to fix the datum position. Typically, two to six gauges are fitted at a midship section to provide corroboration of strain measurements and to provide data on horizontal and vertical distributions of peak strains over the cross section. On some ships several sections have been strain-gauged to investigate longitudinal BM distributions.

Strain recorders are normally attached to the webs of longitudinal stiffeners, clear of structural discontinuities and virtually free from local bending effects, to measure strains representative of hull bending moments.

Strain measurements using electronic recorders

While maximum-reading strain recorders have proved to be a successful and inexpensive means of monitoring extreme loads on a large number of ships over long periods of time, they have a number of limitations which include a requirement for tedious and somewhat subjective manual processing of records together with an inability to provide accurate histograms of strain occurrences or to distinguish vibratory (slam-induced whipping) from lower-frequency strains corresponding to wave encounter. To overcome these limitations a digital electronic strain recorder, designed by Brown (1986), has been developed at ARE with the following features: (*a*) up to eight channels of data can be processed simultaneously; (*b*) filters incorporated in the instrument allow high-frequency (vibratory) components of strain to be recorded separately from low-frequency, (wave-encounter) strains; (*c*) a refined zero-drift compensation circuit provides correction for changes in the strain datum (caused for example by variations in temperature or in a ship's static condition) and allows accurate separation of hogging and sagging strain components; (*d*) protection against electromagnetic interference and disturbances to the power supply, which have plagued previous electronic recorders.

Following successful shipboard trials of a prototype, six recorders of this type have been manufactured and are now in service. Measurements have generally correlated well with records from extreme-value gauges.

Analysis of extreme-value strain data

Some typical extreme-value probability distributions, representing the combined strain measurements from groups of ships in each of eight different classes, are shown in figure 3. Separate analyses have been made of the hogging and sagging components of strain. The strain variations x are plotted on a base of cumulative frequency $F(x)$ and on a linear base y, called the reduced variate, which is related to $f(x)$ by the expression,

$$F(x) = \exp(-e^{-y}). \tag{8}$$

$F(x)$ indicates the probability that a typical 4-hourly extreme is less than the specified value x. The related scale of return period, given by $T(x) = 1[1-F(x)]$, indicates the number of 4-hour intervals which on average elapse between extremes equal to or larger than x. The number of readings included in an analysis is usually too large to allow each reading to be plotted individually. Instead the strain range

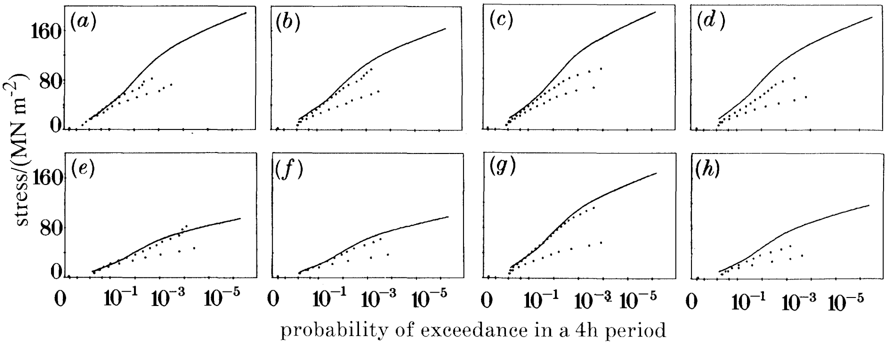

Figure 3. Comparison of measured stresses and theoretical estimates based on linear response analysis. (*a*) Type 12 (Rothesay); (*b*) Tye 81 (Tribal); (*c*) County Class; (*d*) Type 42; (*e*) narrow beam Leander; (*f*) broad beam Leander; (*g*) Type 21; (*h*) Type 22.

is divided into equal intervals and the number of readings in each interval is counted. The mean level x of each strain interval is plotted against the cumulative frequency $F(x)$ of the interval, the latter being taken as

$$F(x) = m/(N+1), \tag{9}$$

where m is the rank of the interval and N is the total number of readings analysed. The rank m is approximated by

$$m = \sqrt{[(p+1)(p+q)]}, \tag{10}$$

where q is the number of readings in the specified interval and p is the number of readings outside and below the interval. The straight line through the observed data is defined by

$$x = u + y/\alpha, \tag{11}$$

where u and $1/\alpha$ are respectively measures of the average value and the dispersion of the observed maxima. These parameters are given by

$$1/\alpha = s/\sigma, \quad u = \bar{x} - \bar{y}/\alpha, \tag{12}$$

where s is the standard deviation of the observed maxima; $\bar{x}$ the mean of the observed maxima; σ the expected standard deviation; and $\bar{y}$ the expected mean. The quantities σ and $\bar{y}$, which depend only on N, may be obtained from tables given by Gumbel (1959). Extrapolation from plots of the type shown in figure 3 for design purposes is discussed later.

Strain–M_b relationship

An assumption inherent in all theoretical treatments of ship response to wave action and also in most measurements of strain data is that the ship's hull can be treated as a non-uniform beam. Provided that the material remains elastic at a point of interest (e.g. a strain gauge location) and that transverse stresses are negligible, strain ϵ at that point is then related to stress σ simply by $\sigma = E\epsilon$, where E is Young's modulus, and to the associated hull bending moment by

$$M_b = \epsilon EI/z, \tag{13}$$

where I is the moment of inertia of the hull section and z is the distance from the neutral axis. It is well known, however, that a ship's plating, which comprises

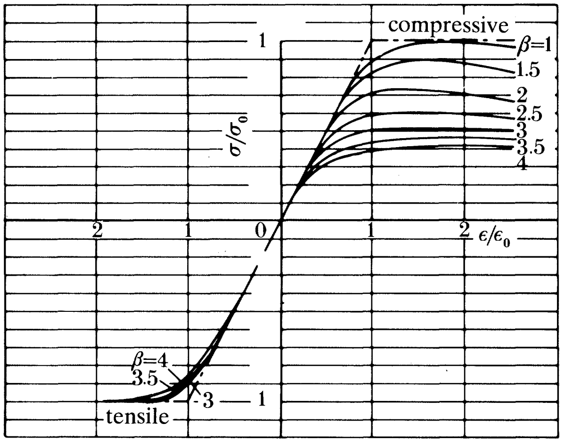

Figure 4. Effective stress–strain curves for ships' plating (average imperfections).

typically between 60 and 80% of the hull section, suffers loss of stiffness and load-carrying capacity as a result of initial distortions and residual stresses caused by welding and flame-cutting and as a result of incipient buckling under compressive load. Such loss of plating effectiveness can have a strong effect on I and z.

Some design curves developed recently (Smith *et al.* 1988) for ships' plating with average imperfections under longitudinal tensile and compressive load are shown in figure 4, indicating the nonlinear variation of average plate stress σ_{ave} as a function of average strain ϵ_{ave} over a range of plate slenderness β ($\beta = (b/t)\surd(\sigma_0/E)$, where b is plate width, t is thickness and σ_0 is uniaxial yield stress). Methods of analysis are now well established (Smith 1977) for evaluating the ultimate strength of a ship's hull accounting for progressive loss of stiffness in parts of the hull section: from such analysis, which is normally carried out as a standard part of the design process for a new ship class, it is possible to extract a nonlinear relationship between the bending moment acting on a hull section and the longitudinal strain at any point in the section. Figure 5 shows, for example, a frigate midship section and figure 6 shows the computed M_{b}-strain relationship, up to and beyond hull collapse, at a longitudinal deck stiffener to which a strain recorder might be attached: the nonlinearity which occurs at $M_{\text{b}} > 30\%$ of the ultimate value is immediately evident. Reduction of plating effectiveness should clearly be accounted for when processing strain data; corresponding reduction of hull rigidity may also significantly affect larger-amplitude dynamic responses of hull girders.

To account for the influence of major structural discontinuities such as deckhouses and large hatch openings, accurate determination of I and z in equation (13) may require reference to detailed stress analysis of the hull structure. Such analysis, using finite-element methods, is now always carried out as part of a new ship design. Appropriate representation of reduced plating effectiveness should of course be included in the finite-element model.

Full-scale seakeeping trials

ARE (Dunfermline, Haslar and Portsdown) have been responsible for coordinating trials on several ship classes for a variety of reasons. A major trial of this type was carried out to compare the relative seakeeping performances of two different ship classes (Clarke 1982; Bishop *et al.* 1984). The two ships ran in parallel over a set

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Figure 5

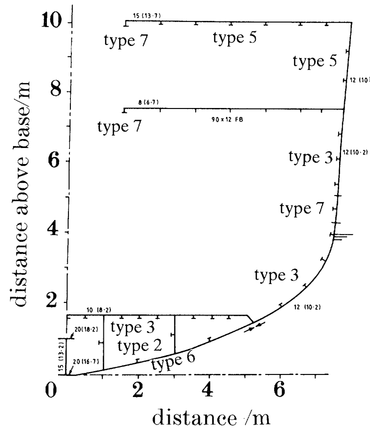

Figure 6

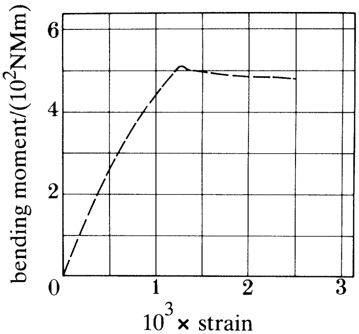

Figure 5. Frigate midship section. Frame spacing is 2.5 m. Material properties: yield stress is 280 MPa; Young's modulus is 207 GPa; Poisson's ratio is 0.3. Plate thickness (with corrosion reduction) is in millimetres. Stiffener types: (1) 25 × 76 mm TEE; (2) 44 × 114 mm TEE; (3) 63 × 127 mm TEE; (4) 76 × 152 mm TEE; (5) 89 × 178 mm TEE; (6) 102 × 203 mm TEE; (7) 127 × 254 mm TEE.

Figure 6. M_b–strain relationship for frigate section.

pattern of headings relative to the waves at increasing speeds from 12 to 22 knots. Ship motions and wave induced strains were recorded in analogue form using FM tape recorders on both ships while the sea conditions were monitored by wavebuoys. Theoretical predictions of slamming responses obtained using the hydroelastic theory of Belik *et al.* (1980) were compared with recorded strain time histories, showing similar results. The use of electronic filtering allowed the modal frequencies and hull-girder damping characteristics to be determined from the strain time histories. In addition to providing information on the frequency of occurrence of slamming and the nature of the slamming components, this trial also provided an insight into the contribution of slamming to fatigue life (Clarke 1987).

Another trial involved measurement of hull flexure in rough seas, during which hull strains were also recorded for comparison with the deformation measurements (J. C. Brown, unpublished work). The opportunity was taken to use a laser system developed at ARE to measure vertical and horizontal bending angles over a distance of about 20 m in the midships region. These measurements were used to provide estimates of relative angular movement between points on the hull as may affect alignment of weapon and sensor systems.

4. Correlation of theory with strain measurements

Using wave statistics derived by R. F. Lofft & W. G. Price (unpublished work) from data published by Hogben & Lumb (1967), long-term statistical loads were calculated for five classes of warship by C. S. Smith & J. C. Brown (unpublished work). These were compared with in-service recording from these classes of ship for a total of 5500 days at sea. The theoretical predictions of extreme stresses were generally considerably greater than the values obtained by extrapolation of measured data.

A total of 9210 four-hourly records from 32 warships of six different classes were subsequently analysed together with data on latitude, longitude, ship speed and heading, wind speed and direction, Beaufort number and wave height from ship logs. The results of the analysis were compared with theoretical predictions by Brown (unpublished work). Despite the fact that measured stresses included a slamming component while theoretical prediction based on linear ship response analysis did not, theoretical predictions were considerably higher than the measured values.

The correlation between extreme stresses and sea state was further investigated by J. C. Brown (unpublished work). In this study records from eight ship classes totalling 24127 four-hourly records over a period of eight years were analysed. As in previous comparisons the agreement between theoretical predictions and measured stresses was poor for large values of stress. It was concluded that poor correlation was attributable mainly to (*a*) inaccuracy of assumptions regarding extreme sea conditions; (*b*) use of linear response theory to deal with essentially nonlinear ship behaviour (compensated to some extent by the occurrence of slam-induced whipping); (*c*) inaccuracy in the evaluation of hull section moduli I/z as used in equation (13).

Some illustrative results referring to several frigate and destroyer classes are shown in figure 3. Solid lines correspond to theoretical estimates of hogging or sagging stress (equal according to linear theory): upper and lower sets of points are respectively sagging and hogging measured values. The difference between hogging and sagging responses and the generally conservative nature of linear response theory are immediately evident. As discussed by Clarke (1986), response nonlinearity and the inequality of hogging and sagging M_b can be accounted for largely by (*a*) dependence on wave height of the buoyancy terms entering into matrix K of equation (1), which can be examined quite effectively using static balance analysis; (*b*) slam-induced whipping, in which sagging M_b is predominant.

The accuracy of linear ship response analysis has been further investigated by comparison with motions and loads measured in a series of tests on a segmented model in regular head and oblique waves in the manoeuvring tank at ARE, Haslar (Lloyd *et al.* 1980). Midship M_bs in waves of moderate height were found to correspond fairly closely with theoretical predictions, emphasizing that the deficiencies of linear response analysis arise mainly in severe wave conditions.

5. Derivation of design values for extreme and fatigue loading

As discussed previously, the Gumbel plots of hull strain measurements can be extrapolated to give a design extreme value with a sufficiently low probability of exceedance. Bearing in mind the conservatism inherent in a linear extrapolation it is considered that a probability of exceedance of 1% is a reasonable design figure. Comparison of results for a number of ships shows that the linearly extrapolated 1% exceedance value is typically 1.42 times the expected maximum value in the life of the ship (Clarke 1986). When possible errors in gauge readings and ship section modulus are taken into account an upper limit of 1.54 is obtained and this is the factor which is currently used in deriving design values for extreme midship bending moment. The expected maximum value for a new ship design can be estimated either by static balance or linear dynamic response analysis with application of long-term statistics. In both cases the estimates have to be empirically adjusted to make them consistent with strain measurements on existing ships. Since linear dynamic

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Table 1. *Comparison between expected maximum bending moments based on measured strains and static balance calculations for $H = 8$ m*

ship class	H/L	M_b from static balance MNm hog	sag	ratio meas./calc. hog	sag
Type 12 (Rothesay)	0.073	122	180	1.02	0.97
Type 81 (Tribal)	0.075	111	160	0.98	1.14
NB Leander	0.073	123	177	1.12	1.17
BB Leander	0.073	127	194	0.97	1.11
County	0.052	325	425	1.00	1.05
Type 21	0.073	129	221	1.08	1.11
Type 42	0.067	165	264	0.81	0.90
Type 22	0.064	188	282	0.90	0.97
CAH	0.042	2200[a]		0.96[a]	
	average (not including CAH)			0.98 ±0.10	1.05 ±0.10

[a] Total range.

response analysis gives higher bending moments than measured at a given probability level it is necessary to use a fictitiously high probability of exceedance value in deriving design values using this method. It has been found that the theoretical expected maximum value is in fact a reasonable approximation for the 1 % exceedance value. However, for ships of length 100–200 m it has been found that the expected maximum value can be predicted with similar accuracy using static balance analysis (without the Smith correction) and a design wave of length equal to that of the ship and of height 8 m. Table 1 shows comparisons based on this strongly empirical approach. Comparison of measurements with predictions based on linear dynamic analysis, with and without nonlinear corrections based on static balance analysis, are given by Clarke (1986).

The distribution of design bending moment along the length of the ship is also discussed by Clarke (1986). Comparisons with the limited data available suggest that extension of the midships design value forward to a point $0.35L$ from the forward perpendicular and then linear reduction to zero at the bow is an adequate assumption to allow for slamming effects.

Since the long-term measurements at sea currently give only the maximum hogging and sagging values in a four-hour period it is necessary to extend these theoretically to derive fatigue loading. Within a four-hour period a ship will typically encounter 2000 waves. Hence there will be approximately this number of stress cycles plus any additional cycles caused by slam-induced whipping. The method adopted by ARE (Clarke 1987) has been to use the long-term statistics distribution derived from linear dynamic response theory to extrapolate from probabilities per four-hour period to probabilities per wave encounter. Using this method it has been shown that the distribution of bending moment per wave encounter can be represented reasonably accurately by an exponential distribution through the design bending moment at a probability of exceedance of $\frac{1}{100}N_L$ where N_L is the number of wave encounters experienced in the life of the ship. The fatigue loading history is then obtained by multiplying these probabilities by N_L and subdividing into bending moment groups. Allowable stresses based on this assumption and available fatigue data are suggested by Clarke (1987).

6. Conclusions

The problem of predicting wave-induced loads on ships has been reviewed with reference to theoretical analysis methods and measurements of strain at sea and to the role of each in the design process. Linear ship response analysis has been correlated with recorded strain data and the limitations of theory in describing extreme loads have been clearly demonstrated. Difficulties involved in the interpretation of strain records have been identified, including correction for zero drift and allowance for effects on hull section moduli of structural discontinuities (e.g. superstructure and deck openings) and reduced plating effectiveness: methods of overcoming these difficulties have been indicated.

Extreme wave-induced loads remain one of the main uncertainties affecting reliability of ship structures. Reasons for this uncertainty and suggested priorities for future research are as follows:

(*a*) Inadequate definition of extreme wave conditions; the priority requirement here is for improved understanding of the mechanics and statistics of extreme waves including breakers and abnormal conditions caused by wave-current and wave-seabed interaction which are known to have caused both structural (Faulkner *et al.* 1924) and capsizing (Pierson 1972) disasters. Such understanding is unlikely to emerge from conventional analysis of masses of oceanographic data for moderate waves: the need is rather for selective analysis of extreme conditions both at sea and in experimental tanks, together with a theoretical attack on the mechanics of extreme wave processes.

(*b*) Nonlinearity of ship response to severe waves: improved nonlinear analysis methods are now becoming available (ISSC) but much remains to be done, particularly in the statistical treatment of nonlinear responses.

(*c*) Inaccuracy in present methods of slamming analysis: a particular need exists for improved definition of gross impulsive forces associated with bottom impact and rapid bow-flare immersion.

In the absence of reliable analysis methods, continued strain measurements on ships at sea must also remain a high priority, both for evaluation of future theoretical developments and as an empirical basis for design. Use of extreme-value strain measurements using low-cost maximum-reading recorders, supplemented by selective use of digital electronic strain recorders capable of distinguishing slam-induced strains and determining load histograms, has proved to be an effective strategy and will continue to be the ARE approach to this problem.

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Discussion

R. Cazzulo (*RINA*, *Genova*, *Italy*). I think that one of the main obstacles to the implementation of reliability-based codes for ship design is still due to the lack of reliable system failure models.

It is true that a perfect mathematical model of the reality never exists but there is no hope in the reliability methods for taking into account model uncertainties, which indicate the deviation of the model from reality, when the model uncertainties are considerably higher than the physical ones.

Ships, for their long tradition, involve procedures based on experience, i.e. material selection, attention to details, stiffeners and scantlings, fabrication procedures which serve their role but cannot be easily treated on rational basis by collapse equations.

Even in the simplest model for the strength of the midship section due to the vertical bending moment, the 'true' collapse is rather difficult: the compressed panels (alternatively below and above the neutral axis) can go to yielding and instability, whereas the tensioned ones can have yielding, fatigue growth and cracking thus the neutral axis and the effective cross section are continuously changing.

Its behaviour as a whole seems more a complex combination of different collapse modes, i.e. plasticity, large deformations, buckling and post-buckling, fatigue and fracture; the interactions among them being very difficult to understand.

What are the present modelling uncertainties in the ultimate ship collapse?

C. S. Smith. I agree that accurate evaluation of ship structural reliability is undermined by inadequacies in the modelling of hull failure. Methods of evaluating ductile collapse, as described by Smith *et al.* (1988) and Smith (1977), do not, for example, account adequately for 'three-dimensional' effects associated with discontinuities such as deckhouses and major deck openings; and the statistics of imperfections, which may strongly influence probability of ductile failure, are not yet satisfactorily defined. Further research into these problem areas is currently being conducted at ARE. In the case of fatigue failure the main need is, in my opinion, for calibration and empirical adjustment of fatigue analysis procedures, as described by Clarke (1987), against the many observed instances of fatigue failure on ships in service.

Deficiencies in the modelling of hull failure appear, however, to be outweighed by inaccuracies in methods of wave load prediction, caused, as discussed in our Conclusions, by inadequacies in the definition of extreme wave conditions and in the analysis of nonlinear ship response, including slamming effects. These inaccuracies are complicated by the uncertain influence of intervention by ships' captains, for example in modifying ship speed and course to reduce slam occurrence.

D. W. Robinson (*Lloyd's Register, London, U.K.*). In presenting results of numerous measurements using scratch gauges, comparisons were drawn with calculations by the linear strip theory, sea spectrum, wave climate approach and in the vast majority of cases, predictions based on linear theory exceeded the measured response. Presumably, because the gauges were measuring maximum strains the results included any high-frequency stress components. Is such an instrument capable of measuring such responses accurately? Are long-base gauges used to overcome local stress

C. S. Smith. I can confirm that the maximum reading strain gauges were designed to record strains associated with the lower modes of hull vibration (up to about 10 Hz) and that tests have confirmed that such strains are correctly recorded. In the case of electronic recording, strains are normally measured using short-base foil gauges but such gauges are located, back to back, on the webs of longitudinal stiffeners, close to the neutral axis of plate-stiffener sections, so that local bending strains are not recorded.

P. T. Pedersen (*The Technical University of Denmark, Denmark*). On one of Dr Smith's slides you showed a calculated functional dependence between the hull girder bending moment and average deck strains. Apparently this curve shows that the ultimate strain is only around 0.12%. In combination with the relatively large wave-induced sagging bending moments he showed us this seems to be a surprisingly low value. Could he explain why the designer didn't stiffen the deck structure such that a higher value was obtained.

C. S. Smith. Figure 9 of the paper shows that collapse of the hull shown in figure 8 will occur at a deck strain of 0.00125, which corresponds to about 92% of yield strain. The design shown in figure 8 is a hypothetical one, chosen to illustrate various features of hull collapse behaviour, including the effect of early plate buckling. A more efficient design, associated with a somewhat higher deck failure strain, would certainly be achieved by reducing the slenderness of the deck plating: I would strongly advocate that in primary hull structure the plate slenderness parameter β should be kept below 0.55 and the column slenderness parameter $\lambda(=(a/r\pi)\surd(\sigma_0/E))$ referring to stiffened panels should be kept below 0.45. It is an unfortunate fact of life, however, that many existing ships are designed with values of β and λ far above these levels.

igure 1

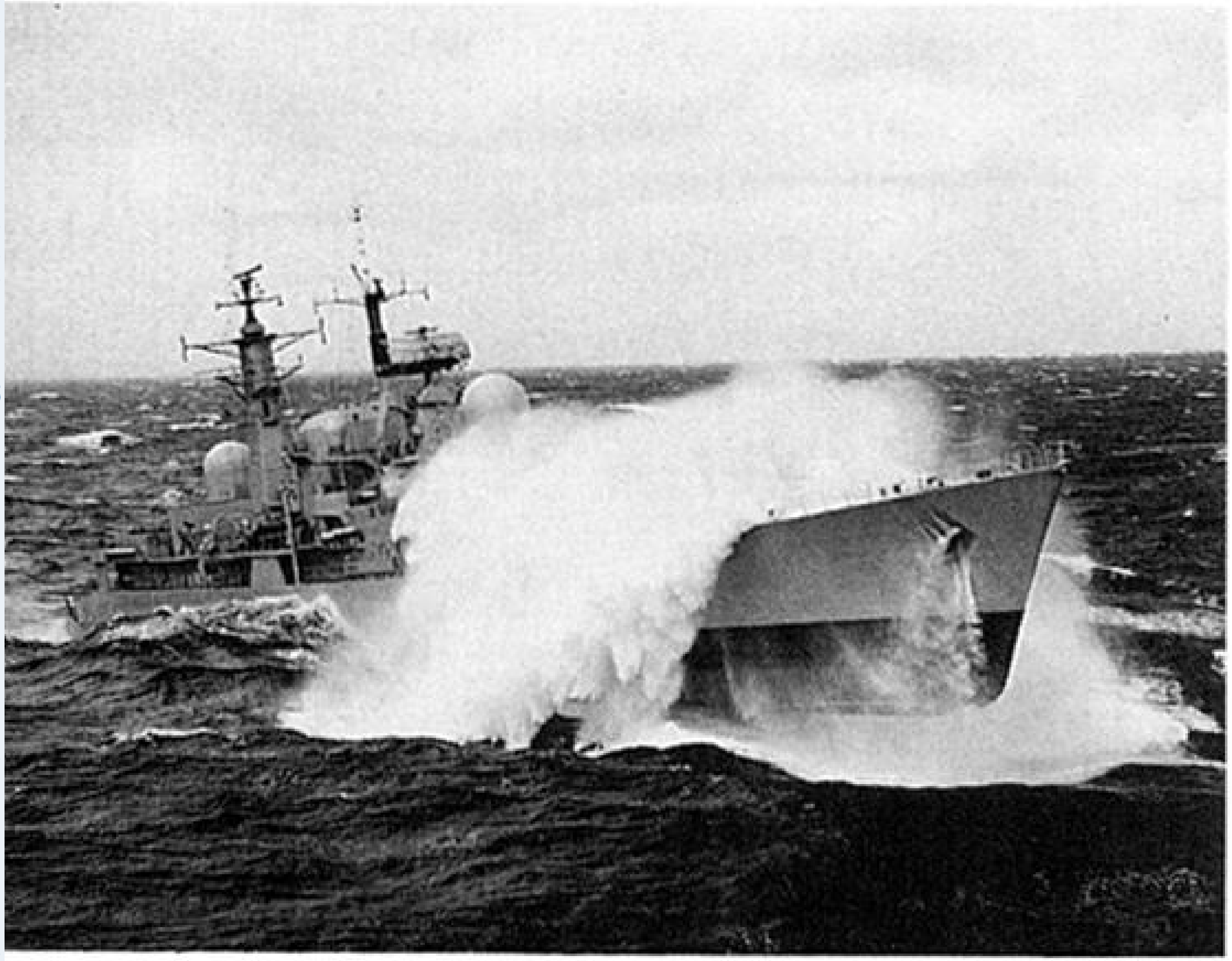
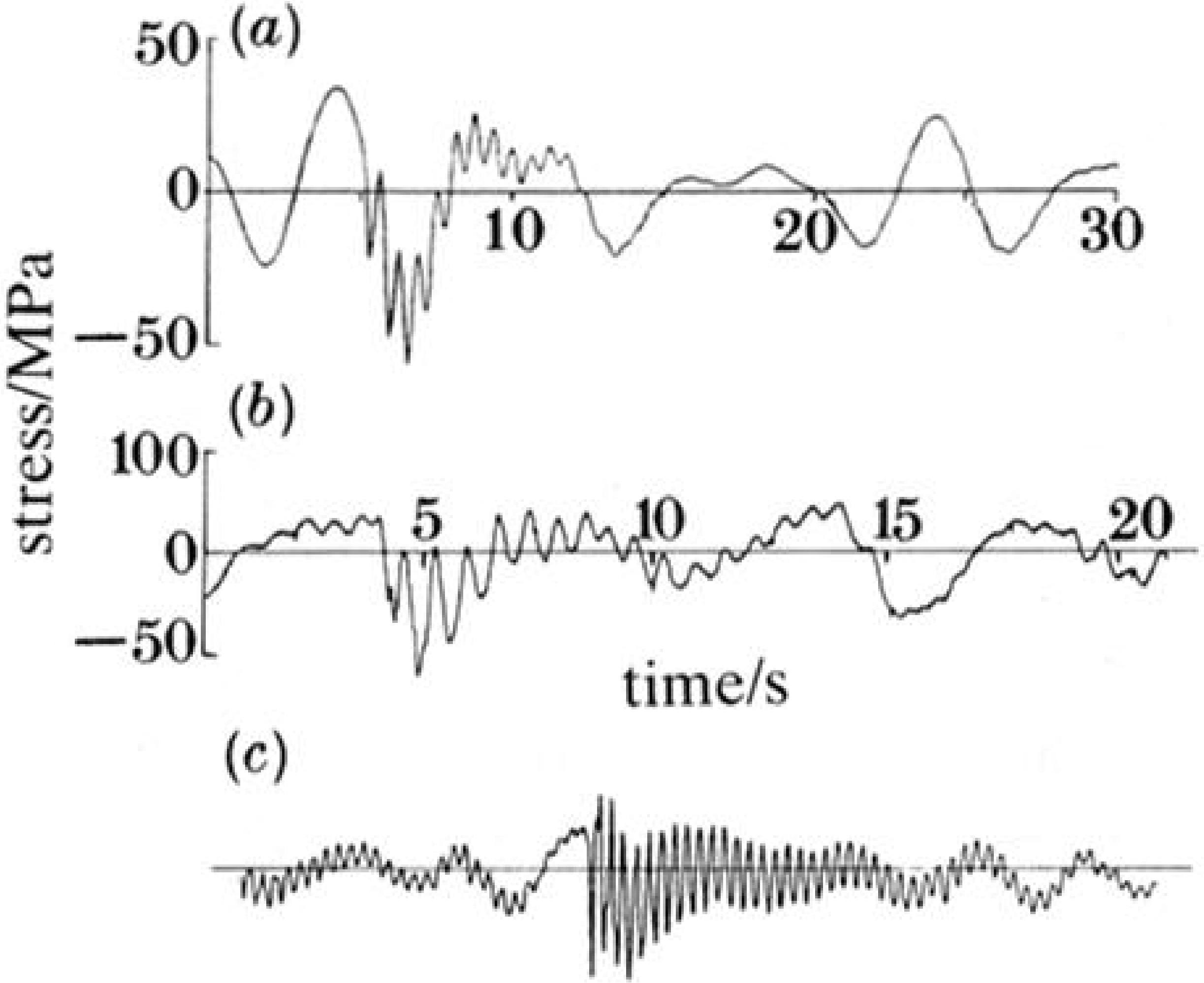


Figure 2



igure 1. Frigate under slamming conditions.

igure 2. Computed and measured slam-induced whipping stresses. (*a*) Computed response: frigate t 18 knots in head seas. (*b*) Measured response: same ship (more severe conditions). (*c*) Measured sponse: landing craft with low damping.