RULES FOR THE CLASSIFICATION OF TRIMARANS

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ABSTRACT

The idea of trimarans can be traced back for more than a thousand years to Polynesian boats. More recently, these ships consisting of a main hull stabilised by two smaller side hulls have been re-discovered and are gaining currency in both military and commercial service.

Lloyd's Register (LR) has developed the *Rules for the Classification of Trimarans* in an attempt to reduce the risk associated with the advancement of trimaran technology. The Ministry of Defence (MoD) has collaborated with Qinetiq and the United States Navy culminating in the building of the Trimaran Demonstrator "RV Triton". Triton sea trials have produced extensive data which has been processed and utilised. Additional technical support during Rule development has been provided by Qinetiq whilst University College London have undertaken a review investigating the completeness and suitability of the Rules.

LR already publishes extensive sets of rules covering a wide range of ship types, however, these requirements are more suitable to either monohulls or catamarans.

This paper will review trimaran characteristics, load prediction, and the development of the *Rules for the Classification of Trimarans.*

INTRODUCTION

A Trimaran configuration is certainly not a new idea; canoes with outriggers (FIG.1) have been in use since Polynesian times. As these canoes were constructed from long, thin hollowed-out tree trunks, when the fisherman stood on board the centre of gravity was too high to keep the canoe upright. It was therefore necessary to provide additional transverse stability, in the form of outriggers or side hulls.

FIG.1 – CANOE WITH OUTRIGGERS

A trimaran platform has already become the basis of a large high speed vehicle ferry and a surface combatant project. Other trimaran applications that are already emerging include passenger-only ferries, container ships, patrol boats and supply vessels, thereby increasing their size from small, unpowered canoes to large, powered ships of several thousand tonnes. The enhanced sea-keeping of the trimaran will enable a better service on existing routes and offer the potential of opening up new routes previously requiring an uneconomically large vessel because of sea conditions. A big advantage of the trimaran is the de-coupling of sea-keeping from vessel capacity to a much greater extent than a monohull or catamaran, by the manipulation for the side hull positioning.

The MoD is investigating the Trimaran platform as a viable alternative to their current Future Surface Combatant designs. A trimaran is a new concept for naval ship design and construction and with MoD funding over a three year period LR has developed trimaran rules to reduce the risk associated with the advancement of trimaran technology and design[1]. Our research and development work has centred on a trimaran configuration consisting of a main hull stabilised by two much smaller side hulls.

The MoD's collaboration with Qinetiq and the US Navy culminating in the Trimaran Demonstrator "RV Triton" has produced extensive data which has been processed and utilised. Additional technical support during Rule development has been provided by Qinetiq. University College London (UCL) have undertaken a review investigating the completeness and suitability of the Rules.

This paper reviews examples of trimaran design and some of the unique attributes of multihulls. For such a novel hull-form load development is not straightforward and so the reader is provided with background information on loading and structural response analysis. Classification and Rule structure are then explained, as well as the project organisation and work task responsibilities. Finally, aspects of Rule development for the Rules and Regulations for the Classification of Trimarans are discussed – namely load prediction methods, load and strength formulae and the direct calculation procedure.

TRIMARANS IN OPERATION

Trimaran Demonstrator

At the time of build Triton was the world's largest motor powered trimaran vessel, with a length of 90m and beam of 22m. Triton is owned by QinetiQ who founded the design and manufacture of the vessel used to quantify the structural and seakeeping performance. The vessel was delivered in August 2000.

FIG.2 – TRITON

The trials programme to determine the suitability of the Trimaran hullform began in October 2000. The first phase of the trials was directed to examining and confirming the naval architectural performance. Triton successfully completed replenishment at sea (RAS), structural loading and seakeeping trials and landing and take-off trials by a Royal Navy Lynx Mk 8 helicopter. The trials were completed in September 2002, successfully providing that the design could operate in exactly the same way as an equivalent monohull vessel.

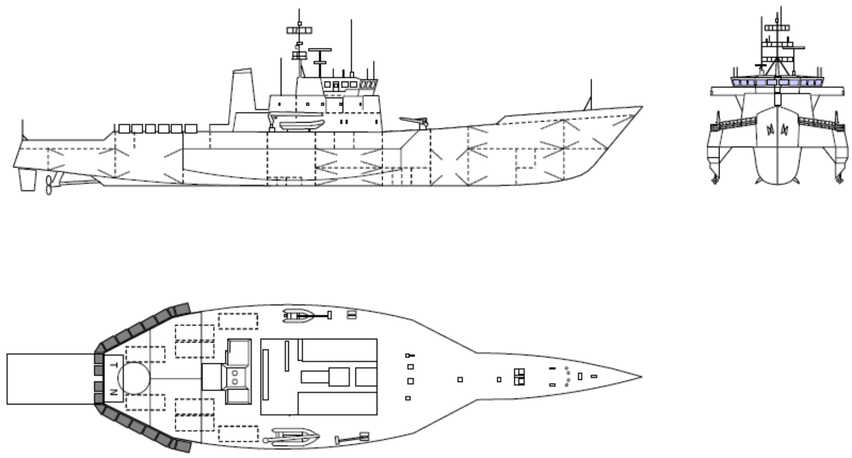


FIG.3 – TRIMARAN DEMONSTRATOR

Triton is now being used as a general trials ship for a range of military and commercial systems. The ship can be used as a test-bed for a range of naval systems including sonobuoys, small towed underwater systems, electronic warfare and the development of signature control technologies.

The demonstrator has two laboratories. One laboratory houses the Trials Instrumentation System (TIS) which collects data including wind speed and direction, temperature, wave height and ship's motion, and the other is for general trials' purposes. The TIS system can record over 400 channels of data.

FIG.4 – RO-PAX TRIMARAN

The world's largest trimaran fast ferry is the "Benchijigua Express". It was delivered on 13 April 2005, commenced commercial operation in early May and has since maintained its Canary Islands service without any major faults. The trimaran is capable of carrying 1,350 passengers and 341 cars or, alternatively, a combination of lorries on 450 truck lane meters plus 123 cars. During sea trials "Benchijigue Express" reached a top speed of 40.4 knots.

"Benchijigue Express" is the most significant vessel to arrive on the fast ferry stage in recent years, opening up a new dimension in fast short-sea transportation for passengers and cargo, even in areas which have so far not been accessible to high speed craft due to rough sea conditions. The ship's design also bears huge potential for military transportation purposes. Moreover, "Benchijigue Express" demonstrates operators' beliefs in technology such as a trimaran platform.

FIG.5 – LITTORAL COMBAT SHIP

The Littoral Combat Ship (LCS) is the first of a new family of surface ships for the US Navy. The LCS is a fast, highly manoeuvrable, networked surface combat ship, which is a specialised variant of the family of US future surface combat ships known as DD(X). LCS is designed to satisfy the urgent requirement for shallow draft vessels to operate in the littoral (coastal waters) to counter growing potential "asymmetric" threats of coastal mines, quiet diesel submarines and the potential to carry explosives and terrorists on small, fast, armed boats.

In May 2004, the United States Department of Defence and the US Navy announced the selection of two separate defence contracting teams led by Lockheed Martin and General Dynamics to each carry out system design and options for the detailed design and construction of two first generation LCS ships. The General Dynamics design is a trimaran with a slender stabilised monohull, scheduled for commissioning in 2008 and 2009.

TRIMARANS VS. MONOHULL AND CATAMARAN

Initial stages of design involve discussions with the owner, proposing various ways in which the owner's wishes can be fulfilled, matching the operations envisaged to the investment that would be necessary to perform them. In the case of commercial operations this would suggest the review of many different possible transportation systems and their probability and chances of success. In the case of military operations it would propose many different ways of achieving offensive or defensive operations and the cost and effectiveness of each solution.

In very simple terms, the decision to go for either a monohull or multihull can be made fairly easily considering whether the ship will be volume or mass limited. A volume limited ship (or capacity carrier) is one which when fully loaded is not down to its minimum freeboard. A mass limited ship is one where there will be unused volume when the mass required is carried. For the latter, a monohull configuration currently offers the lowest building cost per ton of displacement[2] and in the vast majority of cases would be the best choice for a deadweight carrier such as a tanker, bulk carrier or heavily armoured battleship. Passenger ships and light warships are typical of volume-limited ships and it is these types which are most suited to the wider and more convenient deck spaces above the waterline that the multihull offers.

A more complex investigation into the choice of hullform must be based on the required attributes of the ship, such as arrangement, speed and seakeeping. However, for the time being any meaningful comparison between the trimaran and a monohull must assume that the ship is volume limited.

Arrangement

Multihulls provide a good and useful deck layout with a greater deck area per ton of displacement compared to monohulls. For commercial operations, the increased deck space provides a more convenient arrangement for passengers, vehicles and deck cargo, whilst for naval ships weapons can be placed at higher elevations from the base line and flight deck area can be increased.

With the trimaran design it is now possible to build a vessel without the correspondingly large box-style garage deck and superstructure found in catamarans. The trimaran can effectively be a long slender monohull with only side supports. The car carrying and passenger volume is located only above the centre hull and can be adjusted to equate to that of an equivalent catamaran.

Resistance and Propulsion

One of the most significant operating costs of ships is the fuel required for propulsion and so it is important to minimize required installed power. Trimarans have reduced hull resistance at high speeds with a saving of up to 20% at the higher speeds[3] and, as it is this top speed that determines the size of the machinery, a smaller power plant is necessary. The propulsion machinery can be located in the main hull only or divided between the main and side hulls. Location of the prime movers and propulsors are not without careful considerations however, constrained by both the narrow beams of the hulls and the 'matching' between the main and side hull installations.

Seakeeping

The longer the ship, the better the seakeeping in head seas. A trimaran is typically 20 – 30% longer than a monohull of equivalent displacement. Trimarans have better performance in heave and pitch than a similar monohull ship[4]. Trimarans, due to their relatively greater length, also have improved seakeeping characteristics, particularly in the shorter wavelength region of sea states 4, 5 and 6[5]. Trimaran roll amplitudes are also reduced and, hence, they will be able to maintain the same speed in a higher sea-state than a monohull for the same motions.

Monohulls have generally good seakeeping characteristics in head seas, but they can be poor in stern quartering seas. Catamarans are generally good in head seas, although they can be limited by slamming of the cross deck structure and in beam seas the roll acceleration can be high, leading to discomfort. The trimaran appears to provide the best features of both catamaran and monohull, as illustrated in (FIG.6). The poor performance of the monohull in stern quartering seas, and the relatively poor performance of the catamaran in beam seas is apparent, with the trimaran skirting along the top of both.

The stiff roll motion of a catamaran is due to the high transverse inertia of its waterplane area that causes a reduction in roll amplitude but increases roll accelerations. The waterplane area of a trimaran is much smaller and can be tuned to be very similar to that of a monohull ship and the natural roll frequency tuned to avoid commonly encountered wave frequencies. Catamarans may also experience unpleasant 'corkscrew motions' because their natural pitch period and roll period can be close to each other. As the pitch period of a trimaran ship is similar to that of a monohull ship and is far away from the roll motion period, the corkscrew motion will be avoided.

The leading edge of the trimaran cross-deck structure can be located fairly well aft, compared with most catamarans, and the likelihood of wet-deck slamming is reduced. The trimaran also has a very low wake-wash and this is a vital characteristic that can be exploited on ferry services close to communities.

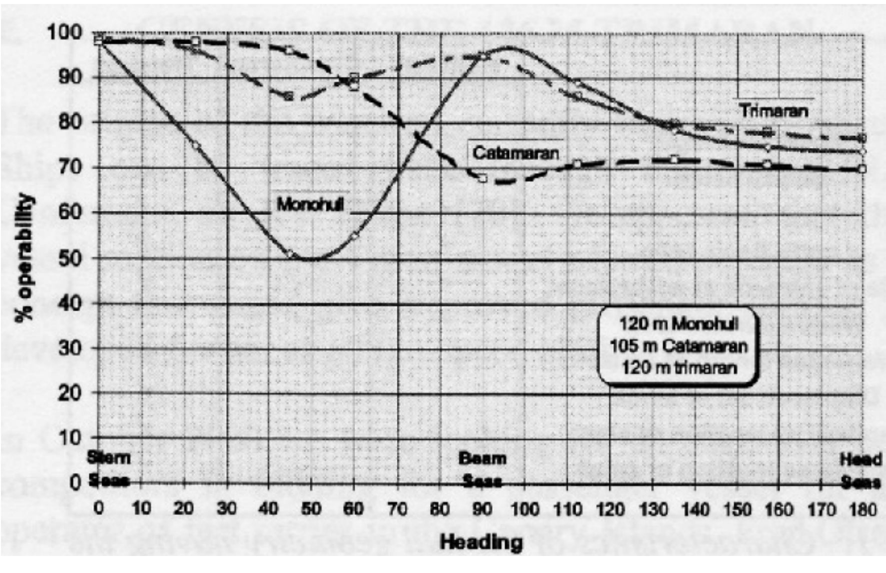


FIG.6 – OPERABILITY ANALYSIS OF THE EQUIVALENT MONOHULL CATAMARAN AND TRIMARAN DESIGNED FOR A 1000 TONNE PAYLOAD[6]

Side Hull Integration

The structural continuity of trimarans is challenging due to the novel configuration incorporating side hulls. A wide range of possible structural configurations allows the designer to tailor the structural arrangement to the required attributes of the ship. (FIG.7) gives some examples of side hull integration layout identified by UCL[17]. The 'unusable' compartments are shown in red, unusable in the sense that the compartment can not be used for cargo deck loading.

Stability

Intact stability is fundamentally determined by the metacentric height GM, the distance between the Vertical Centre of Gravity VCG and the Transverse Metacentre. For multihulls the VCG is higher than for monohulls, a direct consequence of ensuring adequate wet-deck clearance. The position of the transverse metacentre is also far more variable on the multihulls. Multihulls, particularly trimarans, have the advantage of being able to vary the transverse position of the side hulls to obtain the required GM.

The biggest challenge with regard to stability is apparent when we consider damage to one of the side hulls. It may be difficult to meet the legislative requirements for allowable angle of heel if one side hull is damaged. Cross-flooding is not always a satisfactory solution, particularly for passenger ships. In such cases if one side hull is damaged, resulting in a heel to the damaged side, the opposing side hull is then progressively cross-flooded, and an undesirable situation can arise if all passengers subsequently move to and crowd on the undamaged (but cross-flooded) side.

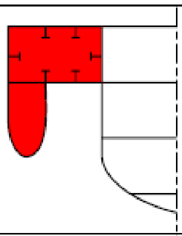
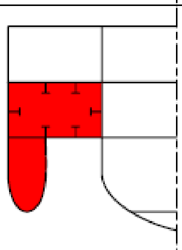
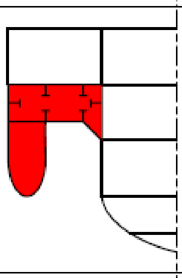
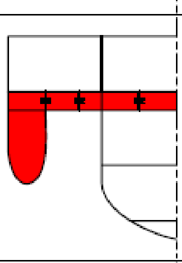
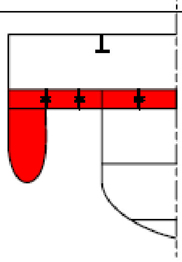
	Unusable cross-deck given over to either void spaces or tanks.
	Two deck high cross-deck structure with usable space on top deck. Lower deck is unusable, with the space given over to either void space or tankage.
	One deck high cross-deck with double bottom. The double bottom is terminated outside the main hull and aligned with the lower deck of the main hull.
	One deck high cross-deck with double bottom. The double bottom extends the entire breadth of the ship.
	One deck high cross-deck with double bottom. The double bottom extends the entire breadth of the ship. The vertical continuity of the main hull side has been broken, and replaced with a girder. Removal of the longitudinal bulkhead presents no major structural concerns[7].

FIG.7 – SIDE HULL INTEGRATION LAYOUT

Very narrow side hulls may have insufficient reserve of buoyancy to prevent the damage heel angle exceeding the legislative maximum, and even increasing the number of transverse bulkheads in the side hulls may provide little benefit. Designing flare into the outer face of the side hull enables waterplane area to rapidly increase with heel; the addition of haunch to the inside of the side hulls has the same effect. However, any increase in hydrostatic stability needs to be balanced against the necessary seakeeping performance, chiefly roll response.

A novel idea proposed by the MoD[5] is to fill the side hulls with ballast water, to a level above the waterline, so that when damaged the water will discharge and heel the ship away from the damage. The ballast water would also help to increase the roll inertia of the ship although the mass of water required to achieve this would be significant, and possibly detrimental to high speed operation.

Current statutory requirements have not been developed with trimarans in mind and this could result in difficulties with interpretations.

Fatigue

In fatigue the most important parameter is working stress range, which is the total variation in the cyclic stress. In a ship structure there are three main sources of this cyclic stress: wave induced loads, alternation between loaded and ballasted conditions and mechanical sources such as the engine and propellers.

Fatigue failure is prevented by controlling the working stress range, and usually the most efficient way to control stress is to either increase local scantlings or modify geometry to reduce stress concentrations and discontinuities. In the overall process of structural design the prevention of fatigue falls mainly within the scope of detail design. But for cyclic stresses that are not locally controllable, such as wave-induced hull girder stresses, care must be taken to ensure that these stresses remain sufficiently small so as not to cause fatigue in the hull girder.

At this time, very little work has been carried out on fatigue analysis of trimaran structures. However, it is expected that the cross-deck locations as shown in (FIG.8), particularly in way of transverse bulkheads should be examined in the first instance.

Another area for concern is the fatigue performance of the main hull at the point of intersection with the cross deck. See (FIG.9) this may act in a similar manner to a break of forecastle, known as an area in need of particular care during the design process due to the change in global load paths.

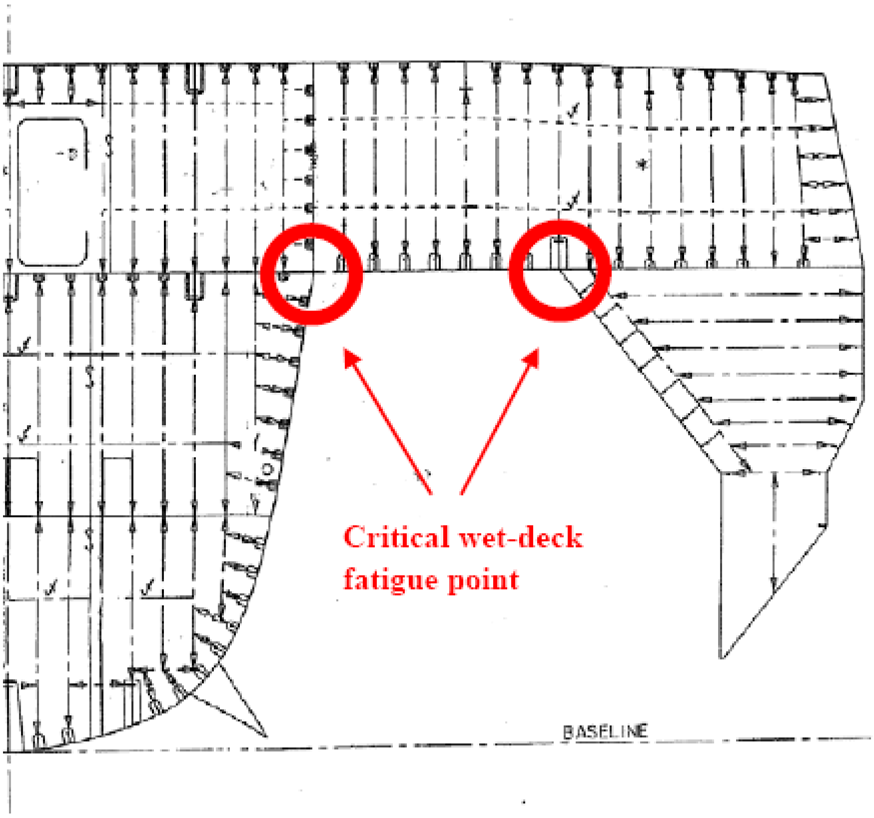


FIG.8 – WET-DECK FATIGUE LOCATIONS

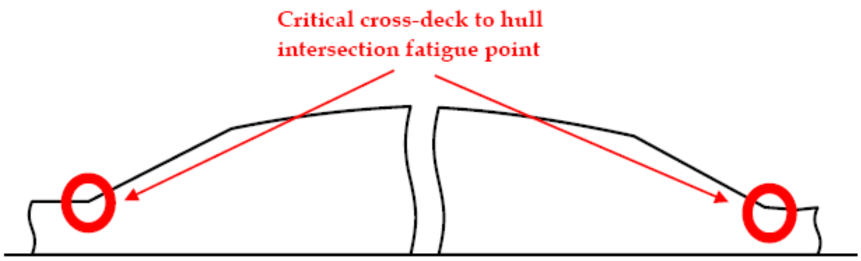


FIG.9 – INTERSECTION FATIGUE LOCATIONS (PLAN VIEW)

CONSIDERATIONS FOR LOAD AND STRUCTURAL RESPONSE ANALYSIS

Load Analysis

The assignation of loads as primary, which affect the hull girder, secondly, which affect large panels of the hull such a bulkheads, and tertiary, which affect local details, are made for convenience in relation to structural consideration. Most loads originate from forces or pressure applied over small areas and whether these loads are subsequently treated in a local or an integrated form is largely a matter of analytical convenience.

Another way of classifying loads is according to how they vary with time[8]: static, slowly varying, or rapidly varying. For these load effects there are three types of corresponding structural analysis: static, quasi-static and dynamic.

In a dynamic load analysis the effects of the time variation of the loading are fully accounted for. Almost any irregular dynamic loading can be represented as a combination of regularly varying loads and if the force-displacement relation is linear or only slightly nonlinear then the problem of calculating the load effects can be solved "in the frequency domain", with frequency as the principal independent variable instead of time, which greatly simplifies the calculations. The frequency-based distribution of a load or a load response is called a spectrum, and so we speak of a wave spectrum and a response spectrum. If the force-displacement relation is non-linear then the problem must be solved "in the time domain", with time as the independent variable. Frequency domain and time domain are discussed under 'Frequency domain and time domain solutions'.

A quasi-static load analysis is simply a static analysis in which the motions are estimated and their effect on the structure is accounted for approximately by including some inertial forces. However, it is usually sufficient to refer to this as a static analysis.

Slowly varying loads are those for which even the shortest component period is appreciably longer than the fundamental (longest) natural period of vibration of the structure. In most cases slowly varying loads can be dealt with by means of static analysis with only a small loss of accuracy, whereas rapidly varying loads such as slamming, usually require a dynamic analysis for sufficient accuracy.

Probabilistic or Deterministic

In addition to the choice between static and dynamic, there are also two different types of load analysis depending on whether an explicit statistical approach is used to define the loads and to calculate the load effects:

Probabilistic

The characteristic values of load effect are calculated explicitly for the particular structure and load. This type of response analysis is preferable for ships such as trimarans where pre-derived characteristics of wave bending moment are not well established.

Deterministic

The characteristic values are obtained from approximate expressions which have been derived previously by means of a systematic series of probabilistic analyses.

Modelling of Waves: Frequency Domain and Time Domain Solutions

The irregular wave systems met at sea can be represented as a combination of regular (sinusoidal) waves and the basic element of ship motion and load responses is the response to a regular train of waves. In practice there are two established methods of applying this irregular surface to derive the ship response – frequency domain and time domain solutions.

Frequency domain solutions consider the irregular sea surface in the form of a wave spectrum, typically following a Rayleigh or Gaussian distribution. A wave spectrum represents the wave amplitude and corresponding wave frequency for a very large number of regular waves with random phase angles. A limitation of this method is that the derived responses are assumed proportional to wave height. The frequency domain simulation can produce reasonable estimates of responses for moderate sea states but may be inadequate for extreme conditions. However, it is worth nothing that the most onerous response does not always occur in the most extreme sea states, particularly for smaller ships.

Time domain solutions reproduce the ship response to a sea state with wave amplitude/period varying over time. It can be generated from the superposition of regular waves with random phase angles. This enables a more accurate representation of the constantly changing shape of the immersed hull as it moves through the waves. Faltinsen[9] demonstrates the connection between a frequency domain and time domain representation of waves in a short term sea states as shown in (FIG.10), where wave energy (proportional to the wave amplitude squared) is plotted against frequency and time.

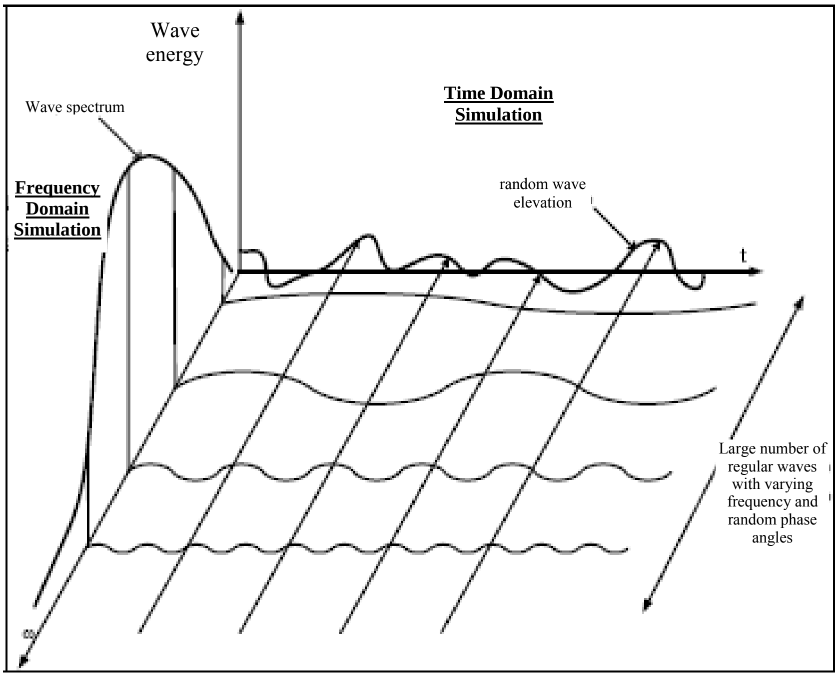


FIG.10 – FREQUENCY AND TIME DOMAIN ILLUSTRATION

Linear or Non-Linear

When we talk about the ship-wave interaction i.e. the forces generated due to the ship motion in waves we can consider this as both linear and non-linear, a linear analysis being more convenient and less problematic. The principal assumption for linear theory are that wave amplitudes are small and the response varies linearly with wave height, and that the ship is wall-sided meaning hydrostatic restoring forces are linearly proportional to ship submergence. Also, in linear theory the free surface condition is satisfied at the mean waterline rather than at the exact surface of the incident wave, and the hydrodynamic pressure are integrated around the body up to the mean water line.

Non-linear theories instead try to lift at least some of these assumptions. The principal non-linearities are associated with the variable wetted surface of the ship hull and with the non-linear hydrodynamic conditions on the free surface. One widely accepted solution is to consider the exact variation of the underwater form as the ship moves through the waves and to integrate the pressure of the undisturbed incoming waves over the actual wetted surface (to obtain the so-called non-linear Frounde-Krylov force). This is especially affected by flare but may be more severe when the wet or main deck becomes immersed or any part of the hulls leaves the water. Some non-linear theory also accounts for such contributions from diffraction waves which is more complicated and considered not important in most cases.

Linear theory can, to a large extent and with corrections for non-linearities, describe the wave-induced motions and loads on trimarans, however non-linear effects are important in severe sea states. In Rules, hogging and sagging factors are introduced for global moments and shear forces to reflect the consideration of non-linearities for moderate sea states.

STRUCTURAL RESPONSE ANALYSIS

Static Only or Static and Dynamic

For our levels of structural response analysis, from hull girder to tertiary, the need to carry out a dynamic structural response analysis depends entirely on whether that level of structural is subjected to any significant rapidly varying loads – loads for which the shortest component period is the same order of magnitude or shorter than the longest natural period of that level of structure. Since natural period differs markedly for different levels, with longer periods for larger levels of structure, a load that is slowly varying at a lower level may constitute a rapidly varying load at a higher level (providing that it qualifies as a significant load at that higher level). Springing is an example of this; wave loads have too long a period to cause any excitation at a primary structural level, but the hull girder may vibrate if the ship is flexible enough. An impulsive or high-frequency load, if it is large enough, will cause a vibratory response at several levels. Slamming in particular, can cause a response at all three levels and can trigger whipping when the ship is relatively flexible globally[32]. However, most rapidly varying loads are not large enough to induce vibration of the hull girder.

Generally, therefore, a dynamic structural response analysis is not required for most ship types. However, the trimaran is a relatively flexible ship, due to the simultaneously occurring dynamic loads acting on its three hulls, both steady-state including symmetrical (longitudinal bending), anti-symmetrical (transverse bending) and unsteady loadings such as slamming and green water on deck. The trimaran design therefore, may benefit from a dynamic analysis.

SHORT TERM AND LONG TERM DESIGN LOADS

Short Term Design Loads

In fully developed seas it is generally held that the statistics of the seaway remain essentially constant (stationary) for a period of time lasting from one hour up to ten hours. The loads arising from such short duration seaways are known as short term loads. Short term design loads are commonly used in the estimation of bow impact or slamming pressures, as these phenomena are likely to occur in severe sea states over a short duration.

Long Term Design Loads

In contrast to short-term loads, the expression 'long-term design loads' relates to those loads encountered throughout the intended life of the ship and so must account for the many combinations of short term loading conditions over that time. The lifetime distribution of loads on a ship is going to be influenced by, amongst other things, speed, heading, loading and environmental conditions. The probability of encountering various sea states may be obtained from wave scatter diagrams, a design load with a probability of exceedance of 10^{-8} is acceptable[10], which is the inverse of the approximate number of low frequency stress reversals that might occur on a period of 20 years. Robinson's LRTA paper[11] gives further detail on methods to obtain the long term response from wave data and regular wave analysis.

CLASSIFICATION OF TRIMARANS AND RULE STRUCTURE

Classification

LR already publishes extensive sets of rules covering a wide range of ship and craft types, many of which have potential for a trimaran platform. In order to develop Trimaran Rules for applicability to ship and craft types contained in the Rules for Ships[12], Special Service Craft[13] and Naval Ships[14] without duplicating requirements contained in these Rules, rule development concentrated on trimaran hull structure only, with full Classification requiring compliance with these Rules plus a Complementary Rule set. The resulting document is therefore an overlay set of requirements to the aforementioned existing LR Rules (Complementary Rules), as demonstrated by (FIG.11).

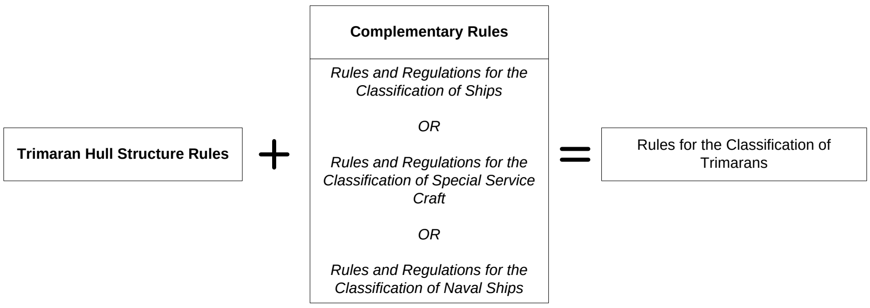


FIG.11 – RULE COMPLIANCE

The interface between the Trimaran Hull Structure Rules and Complementary Rule parts and volumes is shown in (FIG.12). The Rule parts developed specifically for Trimarans are shown in bold, with the remainder cross-referenced to the Complementary Rules.

The content of these four parts can be summarised as follows:

- Volume 1 Part 1 sets out the Rule philosophy, notations, relationship to the Complementary Rules and Rule applicability;
- Volume 1 Part 5 concentrates on the global and local loadings;
- Volume 1 Part 6 provides global strength and local scantling requirements;

- Volume 4 Part 1 is a direct calculation procedure containing structural strength analysis and verification, and load development as an alternative to the rule belongings.

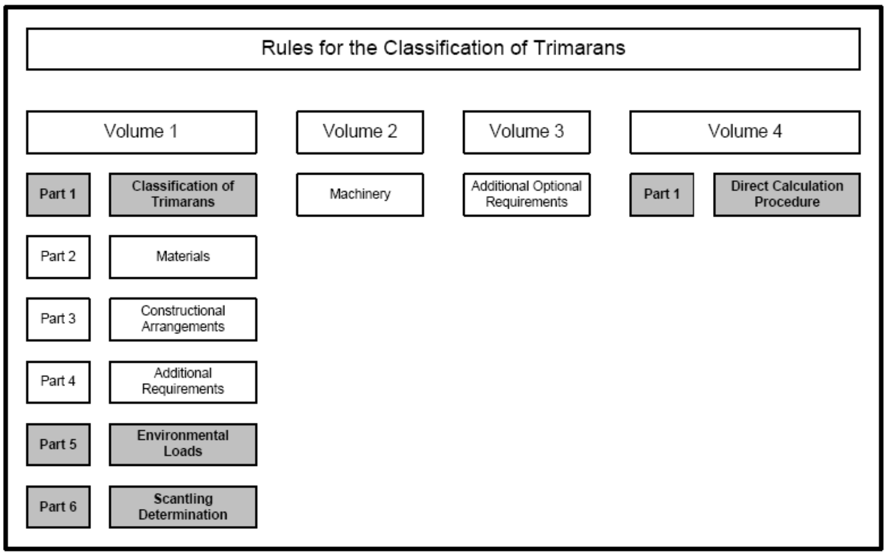


FIG.12 – TRIMARAN SPECIFIC RULE PARTS

As already mentioned a Trimaran Classification will be available to a wide range of ship types classed in accordance with the Trimaran Rules and examples of tentative notations are as follows:

- 100A1 Passenger Ferry TRI
- 100A1 SSC Yacht G6 TRI
- 100A1 NS2 Frigate TRI

The Rules will be applicable to a trimaran hullform with three hulls of displacement form, a main centre hull stabilised by two much smaller side hulls. The Rules are applicable for a length range of 70 to 250 metres, and with a maximum displacement of each side hull not exceeding 6% of total displacement. These were the physical limits of the Trimarans used during rule development.

Hull Structure Rule Philosophy

In short the Hull Structure Rules provide methods of structural response analysis against an acceptance criteria. Once the initial design is developed and validated by the empirical portions of the Rules, a direct calculation procedure is performed to enable a detailed stress analysis. The main emphasis of rule development has been the determination of design loads.

Stages of Rule development are shown in (FIG.13). Firstly, the operational requirements of the trimaran will determine the geographical limits of the service area and its environment. Associated with the operating area are loads, both static and dynamic caused by the environment and the motion of the trimaran. Loadings can be estimated from wave data, long term measurements (strain gauges etc.), global and local environmental theoretical models, or similar techniques. Structural response is the calculation of the group of load effects (bending moment, shear force etc) caused by the environmental loads.

The load effects on the structure are then assessed using a theoretical strength calculation model for chosen strength criteria (direct stress, buckling stress, fatigue etc.), with boundary conditions and limiting stress fractions. Comparative studies verify acceptable safety margins and combined with service experience of other ships determine the final strength calculation model.

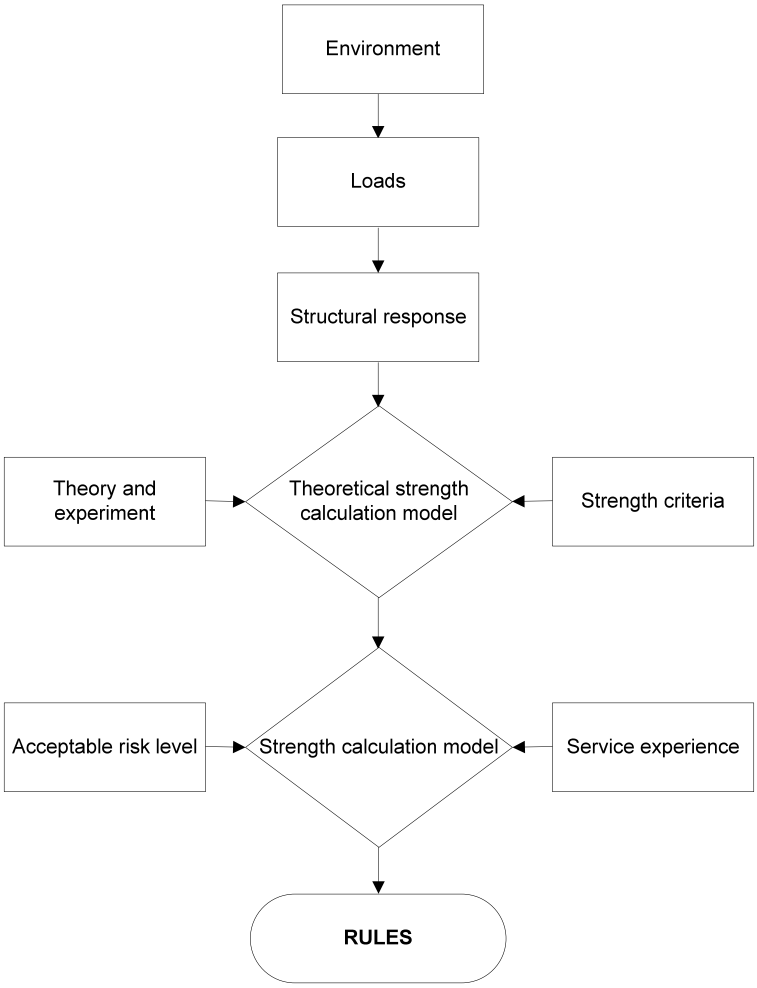


FIG.13 – HULL STRUCTURE RULE DEVELOPMENT PROCESS

TECHNICAL SUPPORT AND REVIEW

LR has experience of a wide range of ship and craft types, including multihulls. However, because of the nature of the work required, particularly for the loadings and structural analysis, a formal collaboration with Qinetiq and University College London (UCL) was set up to provide technical support and review during the rule development process. Qinetiq[7] have been working on the design and analysis of trimaran warships for over ten years, and were involved in the design of RV Triton as well as the subsequent structural sea trials. UCL have been pioneering investigations and naval and commercial trimaran design studies since 1989[15,16]. LR identified key areas of work necessary and these were allocated to Qinetiq and UCL as shown in (FIG.14). The results of this work carried out by Qinetiq and UCL[17] have contributed greatly to the Trimaran Hull Structure Rule development, in that they have been used to develop load prediction formulae and also strength acceptance criteria. The development of these Rules was also reported in LR's paper for the RINA Trimaran conference[18].

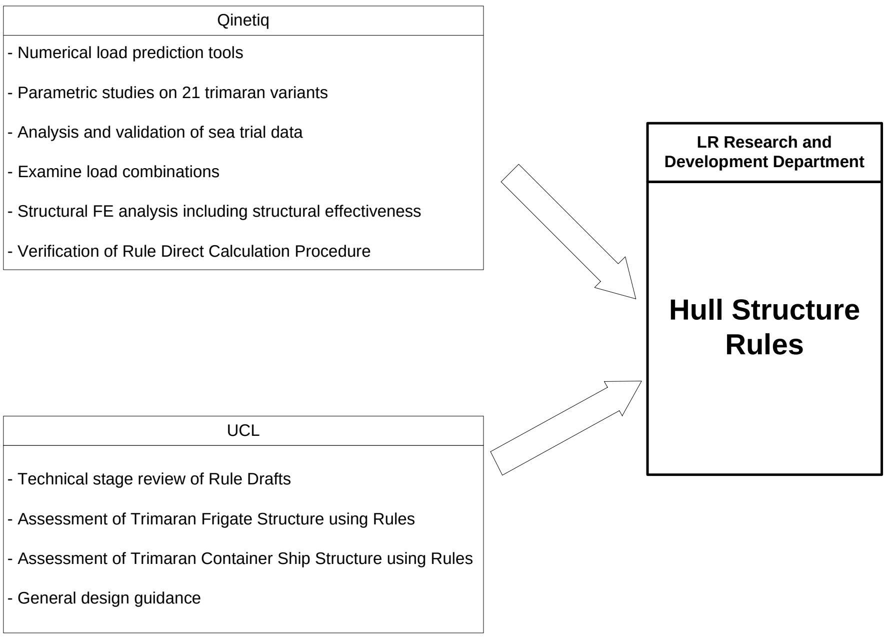


FIG.14 – RULE DEVELOPMENT COLLABORATION

LOAD DEVELOPMENT AND CALCULATION METHODS

Background

Adequacy of a structure can only be realistically determined if it is assessed with sound knowledge of the loads it is likely to experience. Predicting the motion and structural response of a vessel in its operational environment forms a vital part in the design for any new ship or novel hull form.

For the trimaran there is not the same historical experience of performance that there is for more traditional designs. This has meant that increased importance has been placed on computational methods for motion and wave load prediction. The journey to arrival at rule loadings is as shown in (FIG.15), where it can be seen that a variety of methods have been considered to calibrate and develop the rule formulae. As an alternative to the Rule loads a direct calculation procedure is given which will enable a higher level of analysis to be carried out.

Empirical Rule Formulae

Over time classification societies have gathered much information concerning wave loads from computer-based methods, model tests and full-scale measurements i.e. in-service experience. From this they have developed explicit formulae for the characteristic design values for standard ship types, expressed to a great extent in terms of the principal dimensions of the ship. Empirical formulae form the basis of prescriptive rule requirements and this approach has resulted in 'envelope' values which may by quite conservative for some designs. Nevertheless, simple rule specification of wave loads is of great convenience, and provides at the very least, a useful calibration reference point.

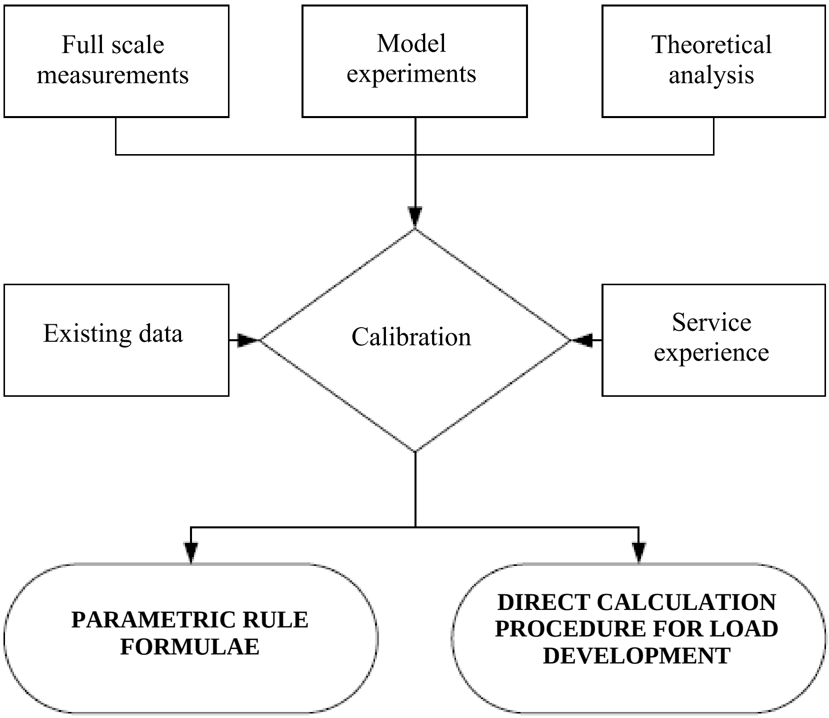


FIG.15 – LOAD DEVELOPMENT AND CALIBRATION

Static Balance Methods

One way of simulating the effects of dynamic wave loading is to imagine the ship momentarily balanced upon a design wave, such that the net force on the ship is zero, and to calculate the corresponding shear force and bending moment distributions. This static balance method is an idealised 'quasi-static' representation and does not directly allow for dynamic effects. There has been much discussion as to suitable methods for specifying design wave heights *H* to be used for the balancing, probably the most common formula are:

- *H = L/20* metres, although this is less commonly used nowadays, particularly for higher ship lengths;
- $H = 0.6\sqrt{L}$ metres;
- The '8m' wave which had been used for naval frigates.

Theoretical Analysis

Qinetiq wave loading capability is focused on the THAFTS suite of software. This is fully three dimensional hydroelastic analysis tool capable of calculating ship motions, induced loads, slam pressures, the effects of whipping and the stress distribution in the structure.

TPRESS is a rigid body implementation which provides a quick method of producing motion and load results. It employs the same hydrodynamic description as the full version of THAFTS, but no longer uses the results of an Eigenvalue analysis and consequently ignores the effects of hydroelasticity. More detailed discussion on various theoretical analysis tools is given later on.

Software tool PPRAO was used to take both the RAO results from TPRESS and wave data information, to carry out the short term and long term response analysis.

The stress analysis tools have been validated for trimaran structures against a number of model scale and full-scale measurements. For the parametric study, numerical predictions for the loadings have been produced using TPRESS.

Model Experiments

To provide data in known conditions a series of model trials were carried out on a free running model at Qinetiq Haslar, see (FIG.16). The experiments investigated five new side hull designs with a programme consisting of:

- Resistance tests over a wide range of displacement and trim;
- Propulsion tests covering a wide range of propeller rpm combinations;
- Seakeeping and structural loadings in regular and irregular waves over a range of speeds, headings and wave conditions;
- Manoeuvring tests to investigate turning ability, directional stability and controllability.

FIG.16 – MODEL TESTING (COURTESY OF QINETIQ)

Results from the model tests[19,20,21] were not used directly for Rule formulae development but to calibrate Qinetiq's numerical load and motion prediction tools.

Full Scale Measurements

Following the launch of Triton, structural sea trials[22,23,24] in the North Atlantic were undertaken (FIG.17). The objectives were to measure wave induced longitudinal and transverse bending in extreme sea conditions to provide validation of the rule design loads, model and numerical predictions. Calibration of the instrumentation was carried out in dry dock. Measurements were taken from a spectrum of headings, sea states and speeds and a comprehensive database of full scale measurements was established.

FIG.17 – TRITON SEA TRIALS (COURTESY OF QINETIQ)

Expected lifetime maxima for the measured responses were calculated, based on an profile of 20 years operation and with an equal probability of occurrence at all headings. These results[25] were used to validate both the Rule formulae and numerical load prediction tools. General agreement was reasonable but it is important to remember that the sample size acquired through sea trials is relatively small.

DEVELOPMENT OF RULE DESIGN LOADS

From a structural design point of view, the rule loads acting on a trimaran are classified as global and local loads.

Global Load Components

Design Accelerations

Long term values of heave acceleration and vertical acceleration due to roll are calculated using well established existing Rule formulae for monohulls. This is certainly reasonable for the heave component as a trimaran has very similar heave characteristics to a monohull equivalent. Vertical acceleration due to roll has been adapted and calibrated using predicted splitting bending moments[25,26]. These accelerations are used in the derivation of both local and global dynamic loads.

Should the trimaran wish to operate in restricted service areas a wave height factor is included in the heave acceleration, formula. The wave height factor is based on the ratio of relative wave heights between unrestricted and restricted service. This factor is not applicable to the roll component, because roll motion is highly non-linear and affected by many parameters apart from wave height.

Still Water Bending Moment and Shear Force

The longitudinal distribution of the weights and buoyancy along the length of the hull in still water are used to calculate the still water bending moment and shear force. This calculation is performed in the same way as for other ship types.

Longitudinal Bending Moment and Shear Force

For both monohulls and trimarans the most important hull girder loads driving the structural design are induced longitudinally by waves in head seas.

The final rule formula bears much similarity to that currently in use for monohulls, which is not so surprising given that we are dealing with a stabilised monohull. Our validation and calibration exercise comparing Triton's sea trial results with the Rule formula[25] also showed this assumption to be reasonable.

$$M_w = F_f \; D_f M_o$$

The Rule formula is essentially made up of three components, a correction for hull form shape F_f, (F_{fh} for hogging and F_{fs} for sagging) a distribution factor D_f and vertical wave bending moment M_o.

The correction for hogging F_{fh} is the IACS correction factor and is therefore not discussed further. The sagging correction factor F_{fs} estimates the increase in sagging moment for low block coefficient ships with flare at the bow or stern. This is a non-linear correction which takes into account the influence of dynamic wave exciting forces, and the rapid changes in buoyancy due to the shape of the bow and stern sections as they emerge and immerse. There may be particular emphasis on the contribution of the stern when operating at forward speed in stern following seas or at zero speed in severe conditions. In the former case, waves may be overtaking the ship and will pick the ship up by the aft end. In the latter case, the ships bow is picked up and the stern can not submerge due to the very large volume in the after end.

As the buoyancy at the ends of the trimaran increases as the wave profile produces two crests, the buoyancy amidships reduces at the trough. In this respect the waterline breadth of hull considered for the calculation should include the side hulls if they are located amidships.

The magnitude and position of the maximum bending moment varies with the longitudinal position of the side hull, with the position having a greater influence on the distribution rather than the side hull displacement. From the result, a longitudinal distribution factor D_f was developed depending on the location of the side hulls and a sample of the results for a 230m trimaran with 3 different locations of sidehulls but with the same overall displacement is shown in (FIG.18). At zero speed the maximum bending moment tends to occur for side hulls amidships, but at higher speeds there was a tendency for some of the aft side hull variants to produce the maximum bending moment.

$$D_f = 0 \text{ at aft end of } L_R$$

$$= 1 \text{ from } \left(0{,}35 - \frac{X_{sh}}{4L_R}\right)L_R \text{ to } \left(0{,}6 - \frac{X_{sh}}{4L_R}\right)L_R$$

$$= 0 \text{ at forward end of } L_R$$

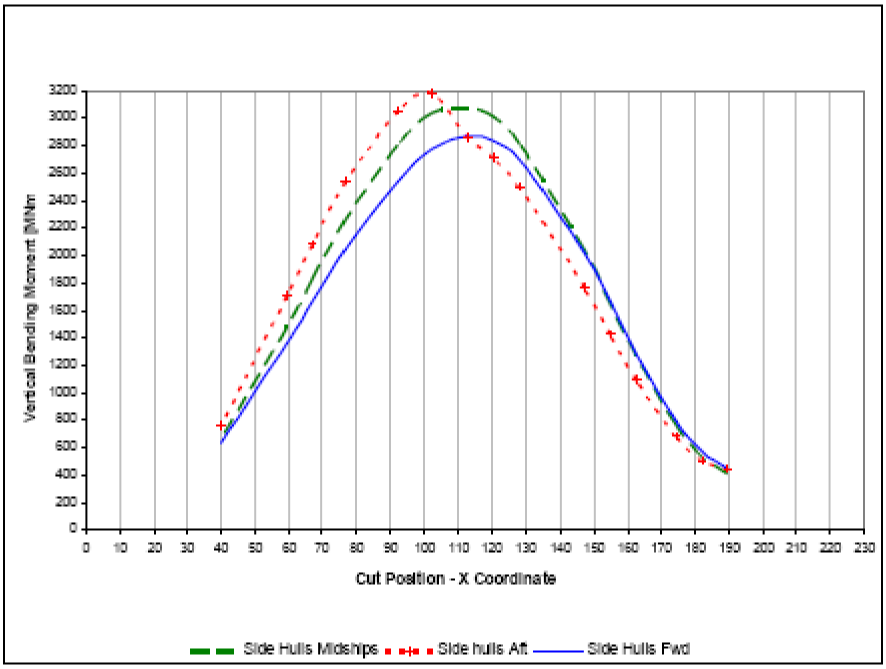


FIG.18 – SIDE HULL INFLUENCE ON BENDING MOMENT

The inclusion of C_b in the formula for M_o is a correction for waterplane shape. As the vertical bending moment increases almost linearly with increasing displacement, and vice versa, the side hulls can have a noticeable effect and should be included in the calculation of Cb to give a representative value for the entire hull shape. The vertical wave bending moment before hog and sag correction is given by:

$$M_o = 0{,}1 L_f f_{serv} L_R^2 B_{wl} (C_b + 0{,}7)$$

Shear force calculation methods also take into account side hull location.

Horizontal Bending Moment

The horizontal bending moment was derived on the same basis as the vertical bending moment except that the wave pressure distribution is integrated across the breadth rather than the depth of the ship. The magnitudes were calibrated from the parametric study and the factor H_f introduced to improve the curve fit.

As with the longitudinal torsional moment, the horizontal bending moment is not likely to be of great significance. However, in combination with other global loads, as prescribed in the Direct Calculation Procedure, the horizontal bending moment may influence some local structure, such as the connection of the cross-deck structure to the main and side hulls. Horizontal bending moment is maximised in quartering seas.

$$M_h = D_f H_f f_{serv} L_R^2 D(C_b+0.7)$$

$$H_f = \left(-70\left(\frac{L_R}{100}\right)^2 + 5L_R\right)10^{-3}$$

Splitting Bending Moment and Shear Force

Splitting, prying, or transverse bending moment is a term meaning the forcing of the side hulls away from the main hull and is widely used in catamaran design. The two load cases considered in the Rules are splitting hogging and splitting sagging. The splitting hogging bending moment occurs when the side hulls emerge, and splitting sagging moment when the side hulls are deeply immersed, as shown in (FIG.19).

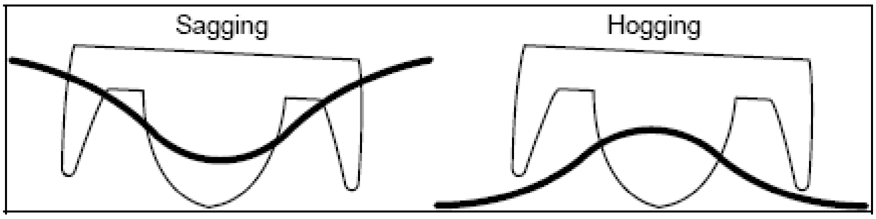


FIG.19 – SAGGING AND HOGGING SPLITTING MOMENTS

The parametric study[26] calculated the transverse distribution of splitting moment along the cross-deck, between the connections to the main and side hulls. Factors which influenced the splitting moment most were displacement and speed, with the greatest loads being experienced in beam seas at zero speed and deep side hull displacement. Peak loads were also calculated when the side hulls were positioned aft, although this was not true for all load cases and therefore not strictly a trend. As the maximum value will occur in beam seas the loads due to pitching are not included. For smaller trimarans the greatest splitting moments will probably be generated in beam seas with wavelengths similar to the breadth of the trimaran.

The Rule formula considers the cross-deck and side hull as a cantilever from the main hull. The splitting hogging and sagging moments are as follows:

Hogging:

$$M_{sph} = 9{,}81\ f_{serv}\ W_{sh}\ (1 + a_z)\left(y_{sh} - \frac{B_{mh}}{2}\right) \text{ at point I, kN.m}$$

$$M_{sph} = 9{,}81\ f_{serv}\ W_{sh}\ (1 + a_z)\ (y_{sh} - y_O) \text{ at point O, kN.m}$$

Sagging:

$$M_{sps} = 9.81\, f_{serv}\, \frac{(\Delta - 2\Delta sh)}{2}\, a_z \left(y_{sh} - \frac{B_{mh}}{2}\right) \text{ at point I,} \quad \text{kN.m}$$

$$M_{sps} = 9.81\, f_{serv}\, \frac{(\Delta - 2\Delta sh)}{2}\, a_z\, (y_{sh} - y_O) \text{ at point O,} \quad \text{kN.m}$$

Points I and O are shown in (FIG.20). Moment values at locations on the cross-deck between I and O are to be linearly interpolated.

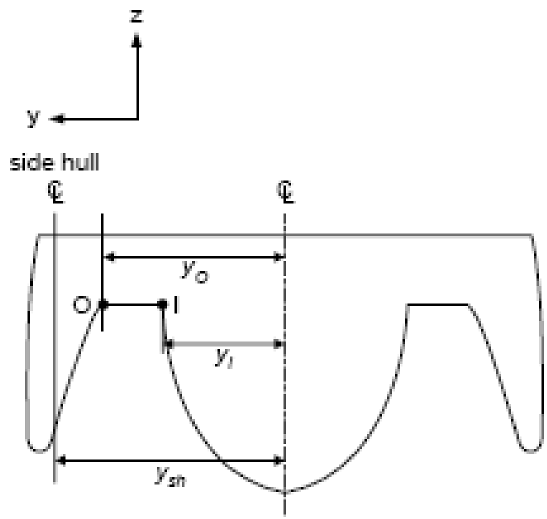


FIG.20 – I AND O LOCATIONS

The splitting shear force, Q_{sph}, corresponding to the hogging condition and Q_{sps}, corresponding to the sagging condition are uniform along the breadth of the cross-deck structure and are to be calculated as follows:

$$Q_{sph} = 9{,}81\, f_{serv}\, W_{sh}\, (1 + a_z) \qquad \text{kN}$$

$$Q_{sph} = \frac{9{,}81\, f_{serv}\, (\Delta - 2\Delta_{sh}) a_z}{2} \qquad \text{kN}$$

Rule formulae for the splitting moments were initially developed by using beam theory, coupling forces and accelerations and assumed load models. These were applied to a series of Trimaran designs to obtain the Rule splitting moment for each case, and then calibrated.

Transverse Torsional Bending Moment

The transverse torsional loads acting on the trimaran are of importance when undertaking the strength assessment of the cross-deck structure. The most severe heading for torsional loads is quartering (bow or stern) seas. The transverse torsional load may be considered to act about the transverse axis as shown in (FIG.21). The transverse torsional load, together with the splitting loads, drives the scantling design of the cross deck structure. The longitudinal positioning of the side hulls has the greatest effect on the magnitude of transverse torsional moment.

Traverse torsional moment is given by:

$$M_{tt} = 3.75 f_{serv} \rho (V_{sh} + V_{cd}) L_{sh} a_{heave} \qquad \text{kN.m}$$

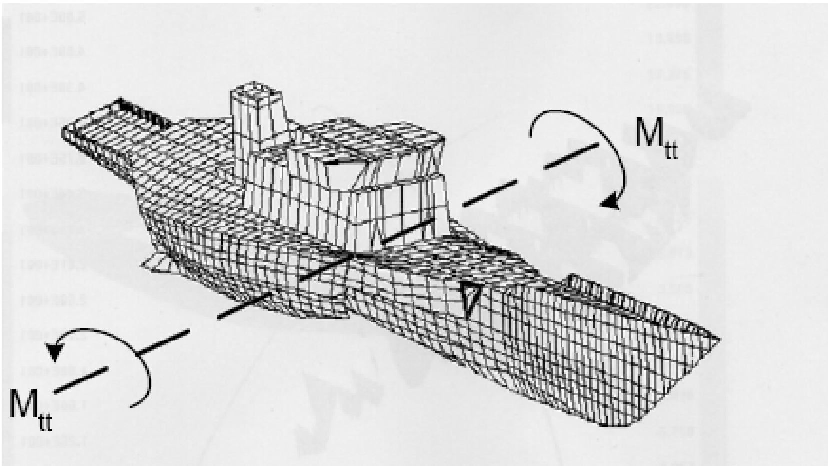


FIG.21 – TRANSVERSE TORSION DISPLACED SHAPE

Longitudinal Torsional Bending Moment

The longitudinal torsional load acts on the longitudinal hull girder, about the longitudinal axis, particularly the centre hull, (FIG.22). The most severe heading for longitudinal torsional loads is quartering (bow or stern) seas. In most cases, this load will not be of great significance as the hull girder is a closed section. However, in cases where the centre hull is extremely slender, the side hulls are relatively small and/or the transverse distance between the side hulls and centre hull is large, this load may become significant. Trimarans with the side hulls positioned aft generate the largest moments.

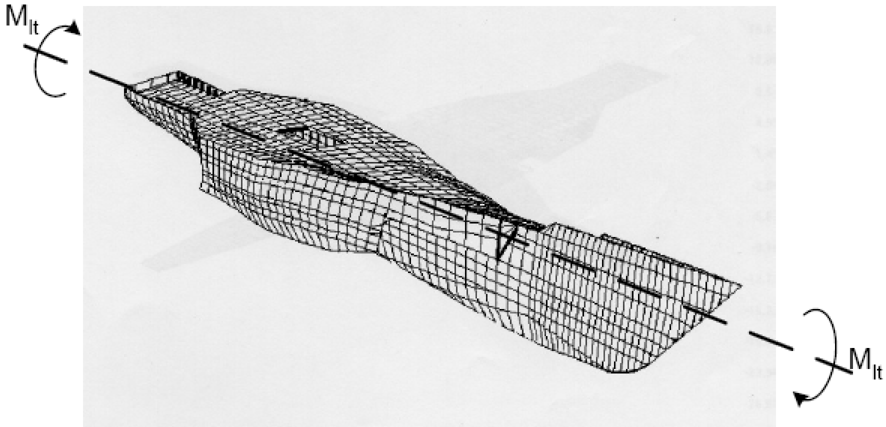


FIG.22 – LONGITUDINAL TORSION DISPLACED SHAPE

The distribution of the longitudinal torsional moment T_f was such that the maximum occurred at the mid length of the side hulls, reducing sharply forward and aft of the cross-deck intersections with the main hull.

Longitudinal torsional moment is given by:

$$M_{lt} = 7.5 T_f f_{serv} \rho \left(V_{sh} + V_{cd} + \frac{V_{mhs}}{2} \right) y_{cs} a_{heave} \quad \text{kNm}$$

Load Combinations

When deriving design loading conditions to be applied to the hull structure, the individual responses, such as longitudinal and transverse bending moments need to be considered together with their correlating factors, because at a given instant the structure will be experiencing a combination of these responses. The concept of load combinations will be explored further in subsequent sections of this paper.

Local Load Components

During the development of these Rules, efforts have been concentrated on investigation of global loads, in particular the influence and effects of the side hulls and cross deck. Local loads to derive plating and stiffener scantling are based on our experience of monohulls.

For the local loading, slamming on the leading edge of the cross-deck and the wet-deck was investigated, and limited model testing[27] and sea trial results[22] suggested that the incidence and magnitude of slamming loads in these areas are probably not of great concern. This appears to be reasonable because the side hulls usually reside well aft of the main hull bow, the bow area normally expecting impact loads due to pitch and heave motion combinations. The worst slamming normally occurs in head seas, and because of the similarity in pitch and heave behaviour at this heading between the trimarans and monohull, slamming formulae are taken from the Complementary Rules.

Compared to an equivalent monohull, the Trimaran arrangement intended for these rules would only exhibit small differences in both heave and pitch acceleration magnitudes. As heave and pitch accelerations form the basis for local pressure distributions for monohulls, trimarans shell envelope pressure distributions are therefore taken from existing monohull requirements and adapted for easy application to a trimaran configuration.

DEVELOPMENT OF GLOBAL STRENGTH REQUIREMENTS

Structural Analysis

A finite element analysis study[28] was carried out on Triton subjected to the anticipated global loads. This enabled quantification of both the level and distribution of stress in the structure. For classification the rules require a detailed structural analysis as mandatory. As a first estimate of the stress in the structure prescriptive requirements to measure structural capacity have been developed using this study.

Longitudinal Strength

Stresses Due to Longitudinal Bending Moment

One of the main issues to consider was the structural effectiveness of the side hulls and cross-deck under longitudinal bending. The study[28] showed the main hull to be fully effective in bending, whilst the side hull effectiveness varied, with values as low as 55% at the base of the outrigger side shell.

High stresses were recorded with the keel in tension. This can be attributed to the position of the transverse section horizontal neutral axis being much closer to the main deck, this 'upper flange' being much larger than the 'lower' flange resulting in bottom plating of substantial thickness. Acceptance stress criteria for longitudinal strength are taken from the Complementary Rules.

Transverse Strength

Transverse loads on the side hulls manifest themselves as transverse stresses, usually concentrated in way of transverse bulkheads with deck and side shell functioning in simplified terms as a flange. In addition, Qinetiq's FE analysis suggested that for the splitting moments, the peak stresses were most likely to be found in way of the transverse bulkheads connecting the hulls to the main hull. The Trimaran Rules also provide guidance on the structural design and integration of the cross-deck structure.

Stresses Due to Splitting Bending Moment

Stress induced by the splitting moment can be calculated directly from the hogging and sagging splitting moments and shear forces and the section modules of the considered longitudinal cross-deck section as shown in (FIG.23).

Stresses Due to Transverse Torsional Bending Moment

Transverse torsional stress distributions can be quite complex when considering multi-cell shear flow, because in a section containing *n* thin-walled cells, each cell comprises a closed cell and so the overall shear flow consists of the superposition of *n* separate circulating shear flows, one in each cell, each of which is constant around the perimeter. Clearly, this can result in a somewhat complex analysis for a cross deck structure which may comprise several deck and transverse bulkheads. Hence the Rules provide a means of assessment with simplified assumptions about the cross-deck structural arrangement to give a rapid estimation of its structural capacity.

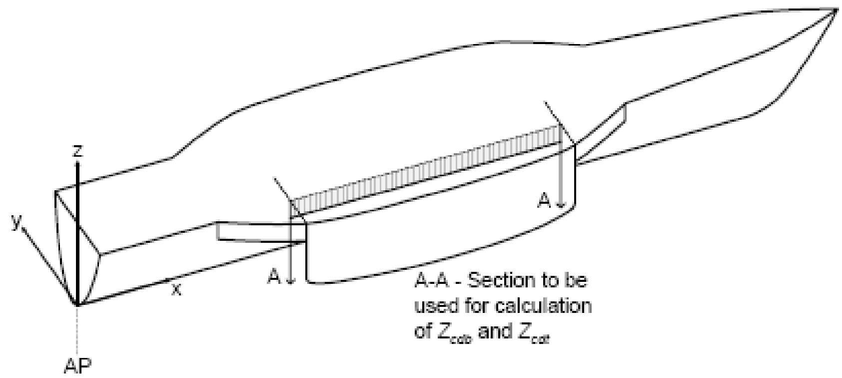


FIG.23 – LONGITUDINAL SECTION FOR TRANSVERSE STRENGTH

For the derivation of stresses in the cross-deck due to transverse torsion the following simplifying assumptions are necessary:

- Hulls infinitely stiff (greater stiffness than cross-deck);
- *'n'* transverse bulkheads, equally spaced;
- Structure is a series of closed cell 'torsion boxes' formed by bulkheads and decks;
- Cross-deck torsion boxes have 100% fixity;
- Each torsion box has equal stiffness;
- Longitudinal position of transverse torsional centre ≈ LCG of cross-deck ≈ geometrical centre of cross-deck ≈ 0.5 x L_{sh};

- Forward and aftermost members will be subjected to the most severe stresses;
- Stresses are due to a combination of vertical bending stress and shear.

The approach adopted is to consider the stresses consisting of 3 components:

- Bending stress σ_{tt} in the cross-deck torsion boxes due to the vertical displacement of the side hulls;
- Shear stress τ_{tt1} in the cross-deck torsion boxed due to the vertical displacement of the side hulls;
- Torsional shear stress τ_{tt2} in the cross-deck torsion boxes due to the twisting of the side hulls about the longitudinal torsion centre.

Load Combinations for Transverse Strength

From the work carried out on load combinations a significant splitting moment occurred simultaneously when transverse torsional moment was maximum in oblique seas, although the same could not be said for transverse torsional moment when the slitting moment was maximum in beam seas. For this reason the cross-deck structure needs to be assessed with a combination of 60% of the splitting moment stresses plus 100% of the transverse torsional stresses.

Local Scantling Requirements

Main development effort has been concentrated on the estimation of global strength capacity. Requirements for local scantling are derived from LR's existing requirements for monohulls and multihulls. If the load values are obtained at the same probability levels as those for monohulls and multihulls, then the structural capacity methods and acceptance criteria for local scantlings are the same.

DEVELOPMENT OF DIRECT CALCULATION PROCEDURE

Background

It is important to quantify the need for direct calculations, because to embark on such an avenue of investigation can be costly, both in time and materials. At this point it is perhaps worth reminding ourselves of the attributes of rules and the two main categories of direct calculations, namely structural response and structural capacity.

Prescriptive rule formulae to calculate the structural response to loads arising from the operational environment are usually of simple empirical format and provide envelope loadings for an anticipate size and configuration range of trimarans. The formulae developed for the Trimaran Rules have been based on a large amount of data, however compared to monohulls the sample size is small. As a result of this the estimate of structural response must include inherent conservativism. As more trimarans are built and further in-service data is gathered, these formulae can be refined and enhanced and the level of confidence in their values increased.

When assessing the structural capacity of the trimaran to withstand loading the hull structure is idealised as a beam, the simplifying assumption then being that the hull girder behaves in accordance with simple beam theory. This can be a fairly drastic simplification although there are some corrections that can be applied to account for effects such as shear lag. The principal assumptions that remain are that the hull girder is elastic, its deflections are small, there is no in-plane distortion and the strain due to bending varies linearly over the cross section about the neutral axis. Rule of thumb methods exist for estimates of the effectiveness of a structural component i.e. whether it is of sufficient length to be effective; this is particularly important when considering the effectiveness of the side hulls in the longitudinal hull girder bending assessment. Hull and superstructure interaction can only be accurately modelled three-dimensionally by using 3D Finite Element Analysis.

From the discussions above, it is clear that there may be great benefit in a direct calculation of both response and capacity to verify the design and ensure the design stresses and buckling criteria are within acceptable levels.

The direct calculation procedure consists of two main parts; Structural Strength Analysis and Verification, and Load Development. The first is mandatory for all trimarans and sets out a method for detailed stress analysis using the Finite Element Method and the process is illustrated in (FIG.24). Load combinations to be applied to the model are a practical attempt to reduce the number of load cases to a reasonable number rather than for a specific sea environment. The second part deals with load development as an alternative to the rule loadings, these loads may be used instead of the rule in most aspects of the structural verification, and may be determined theoretically or experimentally through model tests.

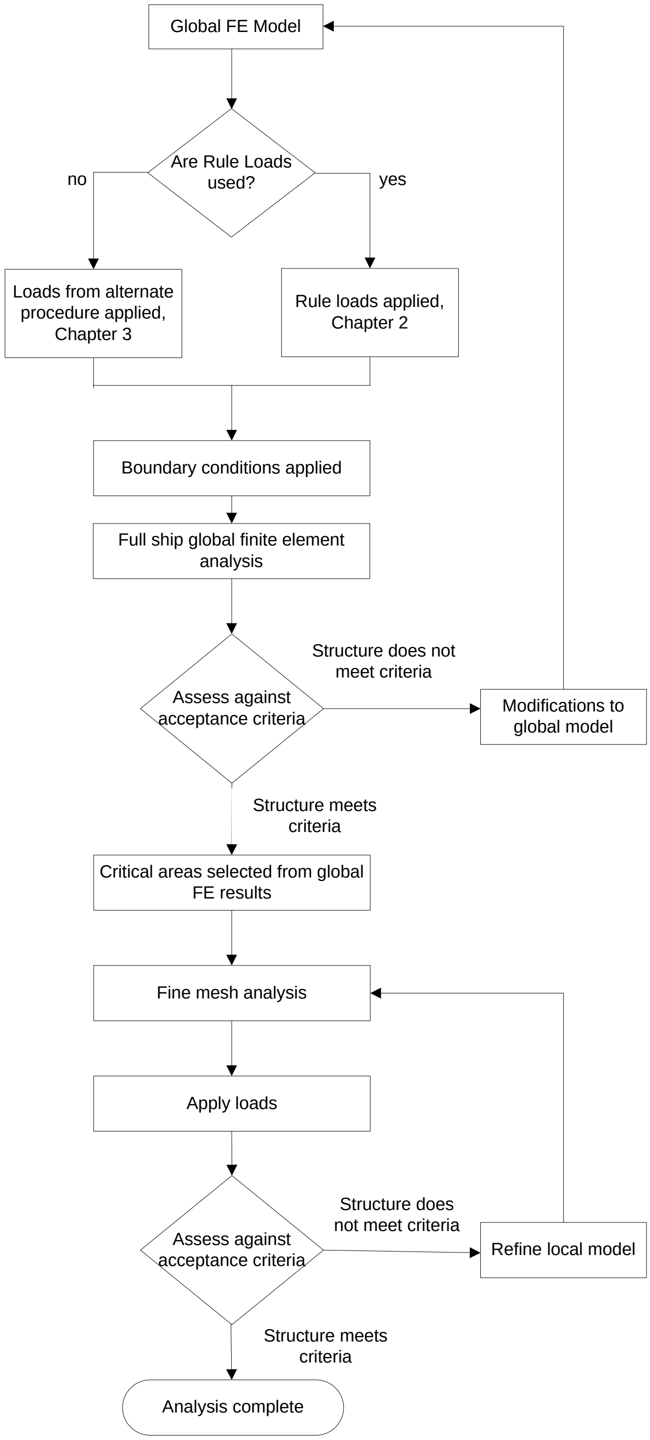


FIG.24 - STRUCTURAL ANALYSIS AND VERIFICATION

Load Combination Factors

In operation, trimarans will experience several load components simultaneously. Some load components may be more dominant than others under certain conditions. When deriving suitable design conditions for trimarans the individual load components, including static and dynamic loads, need to be combined. The concept of load combination factors was introduced in the Trimaran Rules to produce a table of factors (see Table 1) combining the simultaneously acting load components in a rational and simplified way. This is also a practical attempt to reduce the number of load cases to a reasonable number.

Briefly, to calculate the load combination factors, a few representative load cases can be chosen to capture the worst scenarios. For each case, the dynamic load combination factors are calculated by maximising one critical load component and taking a 'snap-shot' of other simultaneously acting dynamic loads. The ratios between the values of these dynamic loads at the 'snap-shot' and their respective long-term envelope values give the dynamic load combination factors.

Methodology

The design approach of using load combination factors is based on the Equivalent Design Wave (EDW) concept. An EDW is defined as the regular wave giving the same response level as the reference design value, and it is described by a wave period and wave amplitude.

The wave period of EDW is selected by finding the peak of the transfer-function of the maximised response. (FIG.25) shows as example of the Response Amplitude Operator (RAO) amplitudes generated for vertical bending moment and splitting moment for the same load condition, ship speed and heading. RAO is the amplitude of response to a unit wave amplitude for a range of wave frequencies which are complex numbers normally presented by absolute amplitude values and phase angles. Amplitude a_{max} and phase angle ε_{max} are obtained for the maximum response, in this case vertical bending moment. At the same time instant amplitude a_i and phase angle ε_i are obtained for the simultaneously acting response which in this case is splitting bending moment.

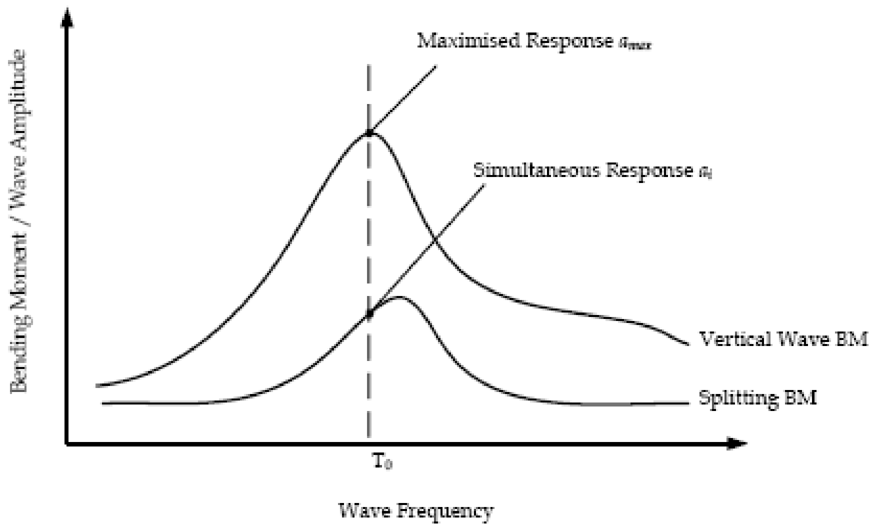


FIG.25 - SKETCH OF THE RAO AMPLITUDES OF CHOSEN RESPONSES

The phase angle represents the time difference between the maximum wave amplitude and maximum excitation (response) amplitude, as shown in (FIG.26).

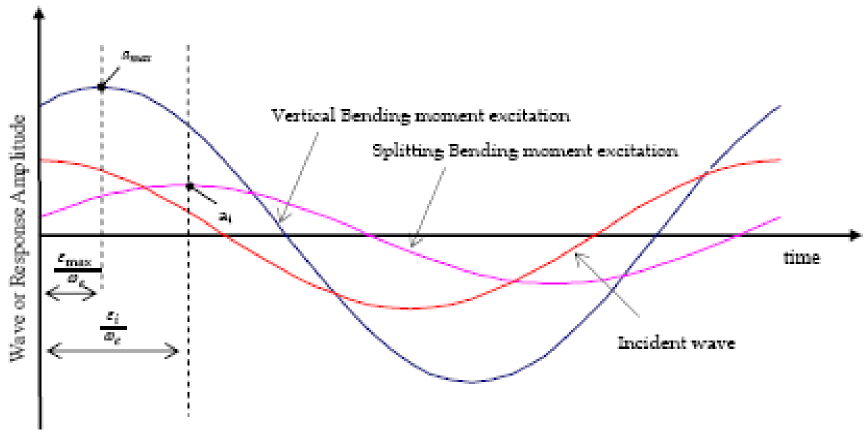


FIG.26 - RELATIONSHIP BETWEEN EXCITATION (RESPONSES) AND INCIDENT WAVE

The load combination factors themselves are calculated as follows:

- Choose response to be maximised;
- Review RAO of chosen response for all headings to find the heading with maximum RAO;
- Define the design value R_{max} for the chosen response. R_{max} is the rule design value or a directly calculated long-term value;

- From the RAO curve determine the maximum RAO a_{max} and corresponding phase value ε_{max} of the chosen response, see (FIG 25). T_0 is the period of the EDW;

- Determine the time instant t_{max} at which the chosen response is a maximum, given as:

$$\cos(\omega_e t_{max} + \varepsilon_{max}) = \pm 1 \qquad rads$$

Where w_e is the wave encounter frequency,

$$\omega_e = (1 - \frac{\omega V}{g}\cos\psi)\omega$$

Where w is the frequency of the incident wave, V is the forward speed of the trimaran in *m/s*, and ψ is the wave heading to the ship. Here ψ = 180 degrees means head sea;

- The wave height of the EDW is given as:

$$h_{max} = \frac{R_{max}}{a_{max}} \qquad m$$

- At t_{max} for the same operation profile obtain the amplitudes a_i and phase angles ε_i of all the other simultaneously acting responses at T_0 (See FIGs. 25 and 26);

- The magnitude P_i of the simultaneously acting response at t_{max} is given by:

$$P_i = h_{max} a_i \cos(\omega_e t_{max} + \varepsilon_i)$$

- The proportion of the simultaneously acting response at t_{max} compared to its design value $R_{max\text{-}i}$ gives the load combination factor (LCF) as follows:

$$LCF = \frac{P_i}{R_{max-i}}$$

Calculation of Rule Load Combination Factors

From the results of the ship motion and load predictions, the ship speed, heading and the location of maximum response can be identified for each of the load components. This had been done for all the trimaran variants by reviewing the longitudinal and transverse distributions of the predicted design values[26]. Following the approach described previously, the response components were then maximised one at a time for each chosen load case[29]. Having established the time instant and wave condition in which a maximum response occurs, the responses of the other components were calculated. Table 1 shows the load combination factors developed for the Rules, for example if we take Load Case 1 this would give a total bending moment M_{tot} resulting from static bending moment M_{st} and dynamic bending moment M_{dy}.

$$M_{tot} = M_{st} + M_{dy}$$

Where

$$M_{st} = M_{swh}$$

$$M_{dy} = M_{wh} + 0.3M_{sph} - 0.2M_{tt}$$

The proportion of M_{tt} occurring is negative, this is because the component is considered reversible, depending on its phase relative to the maximised response. For responses where there are both hogging and sagging components the amplitude (and hence the LCF) can be represented as hogging or sagging rather than positive or negative.

TABLE 1 - *Load Combination Factors*

Wave Detection	Maximised Response Component	Load Case No.	Load components									
			Static Loads		Dynamic loads							
			M_{swh}	M_{sws}	M_{wh}	M_{ws}	M_h	M_{sph}	M_{sps}	M_{lt}	M_{tt}	Θ_{max}
Head Seas	Hogging Vertical Wave BM	1)	1.0	0.0	1.0	0.0	0.0	0.3	0.0	0.0	-0.2	0.0
	Sagging Vertical Wave BM	2)	0.0	1.0	0.0	1.0	0.0	0.0	0.3	0.0	-0.2	0.0
Beam Seas	Hogging Splitting BM	3)	1.0	0.0	0.1	0.0	0.0	1.0	0.0	0.2	0.0	0.0
	Sagging Splitting BM	4)	0.0	1.0	0.0	0.1	0.0	0.0	1.0	0.2	0.0	0.0
Oblique Seas	Longitudinal Torsional BM	5)	whichever results in the highest global stress		0.0	0.0	-0.3	0.4	0.0	1.0	0.3	0.0
	Horizontal BM	6)	whichever results in the highest global stress		0.0	0.0	1.0	0.4	0.0	0.0	-0.2	0.0
	Transverse Torsional BM	7)	1.0	0.0	0.0	0.2	-0.2	0.6	0.0	0.0	1.0	0.0
	Static Roll	8)	0.0	1.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	1.0

Structural Modelling

The structural modelling and boundary conditions have been developed from experience of three-dimensional modelling for various ship types. The structural modelling requirements aim to give the designer some flexibility and in view of the relatively limited application experience, recommendations and guidance are provided rather than prescriptive requirements.

A full breadth model is recommended as this will allow for any conditions of asymmetry and will also simplify loading and boundary conditions. Finite element analysis of ships is divided into following categories:

- Global or whole ship model; where the ship is modelled using a coarse mesh. This is to ensure that the global hull stresses comply with allowable stresses;

- Local model; where a refined mesh of a particular part of the ship's structure, such as a half-width section of the hull and / or superstructure. This is modelling using a detailed mesh and boundary conditions supplied by the global analysis. This can be included in global model;

- Stress concentration analysis; where a very detailed analysis of a minor structure such as a hatchway. This is carried out to determine the geometrical stress concentration factor for fatigue analysis calculations in either a separate model or global model.

The following potential areas of high stress have been identified based on Triton's configuration:

- In way of the intersection between the cross-deck structure and the main hull (horizontal bending moment);

- The most highly stressed transverse bulkhead and its deck interface (splitting shear force);

- Transverse bulkheads in way of midships (longitudinal torsional moment).

In general, the procedure for FE analysis adopts a similar approach to that given in LR's *ShipRight Design Assessment Procedure*.

If Rule loads are being applied to the model the standard load cases developed with the load combination factors (LCF) (Table 1) are to be considered. The Rule loads are to be applied in turn to the FE model using the boundary conditions described in the Structural Strength Analysis and Verification section of the Rules. Stresses obtained for each individual Rule load can then be combined (for stresses in the same plane) using the LCFs to give the total stress.

For loads calculated directly the simultaneously acting loads are to be derived and superimposed.

It is recommended that the still water case be modelled first, and separately, so that the correct weight distribution i.e. buoyancy and mass is obtained for subsequent analyses. It will usually be necessary to apply an interactive procedure to balance the mass and buoyancy forces at the correct trim by adjusting the weight distribution as necessary. The wave-induced load cases are simple and extension of the still water case. Using the still water weight distribution, the aim is to balance the trimaran on the wave profile.

A finate element analysis was carried out on a 90m trimaran applying rule load cases and boundary conditions. Sample results are as shown in (FIGs 27 and 28) (superstructure removed for convenience).

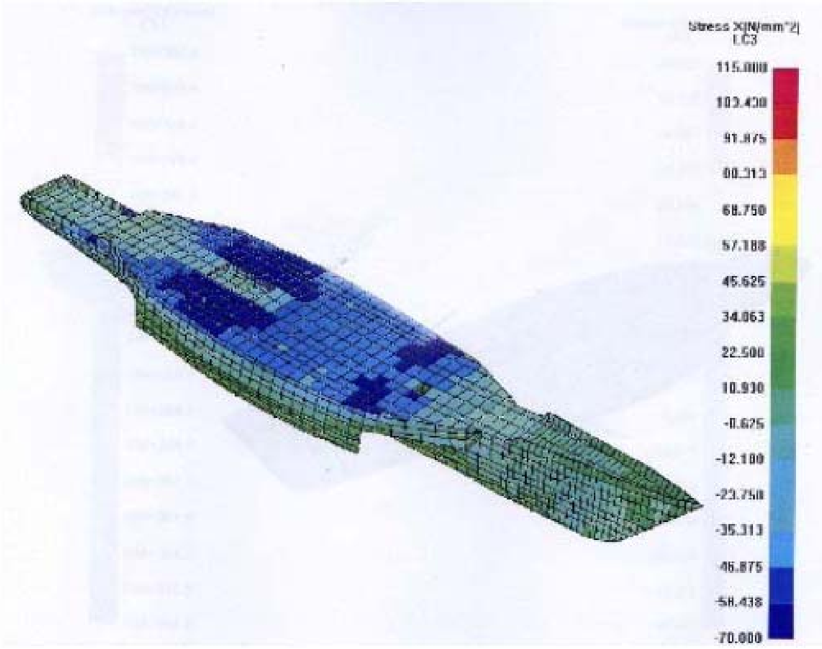


FIG.27 - LONGITUDINAL BENDING SAGGING - LONGITUDINAL STRESS DISTRIBUTION AT DECK (COURTESY OF QINETIQ)

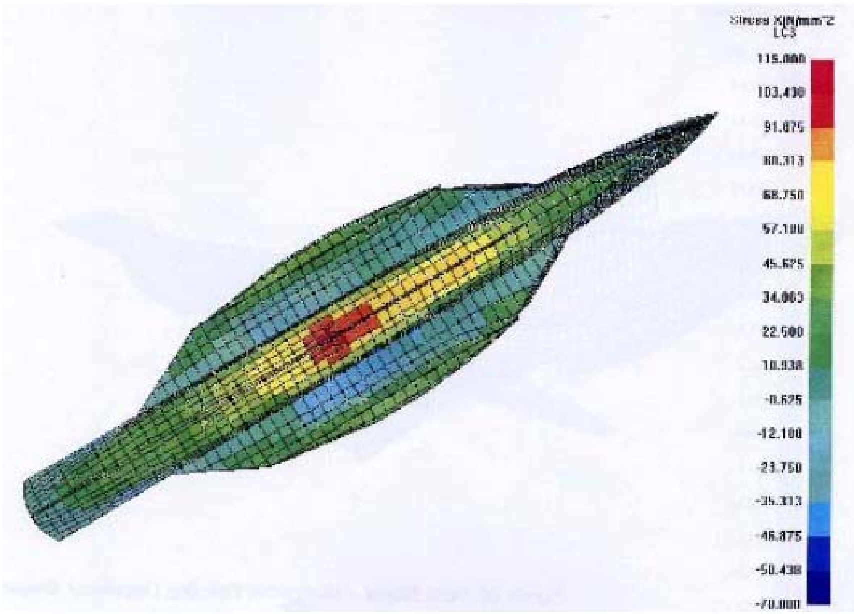


FIG.28 - LONGITUDINAL BENDING SAGGING - LONGITUDINAL STRESS DISTRIBUTION AT KEEL (COURTESY OF QINETIQ)

Load Development

As an alternative to the Rule Loads, a load development procedure based on a direct calculation method is provided. These loads may be used instead of the Rule loads in some aspects of structural assessment to derive the design loads provided that the proposed loads are submitted and approved by LR.

Limited experience with trimarans compared with monohulls makes theoretical and experimental verification of design loads important. The reliability of theoretical methods for both global and local load prediction often makes the experimental verification of design loads imperative.

Theoretical Analysis

The determination of dynamic loads is to be based on theoretical analysis of ship load and motions using long term motion and load prediction methods.

For a rigid body analysis the trimaran is regarded as an unrestrained rigid body with six degrees of freedom. The level of complexity for analysis of the interaction between the waves and the ship's hull can largely be determined by choosing between 2D and 3D and also between linear and non-linear approaches. All the approaches discussed below are based on velocity potential theory which means that the effects of flow separation and viscosity are neglected. This approximation has approved to be appropriate in sea-keeping analysis for ships in terms of accuracy and computational cost.

A traditional 2D or 'strip theory' investigation is based on the simplifying assumptions that the ship is subdivided into a number of prismatic segments and the forces are calculated separately for each segment using 2D flow theory, neglecting longitudinal flow and interaction between adjacent segments. For conventional slender ships, strip theory works reasonably well for global motion and load prediction for engineering purposes. In terms of response predictions for a trimaran, one drawback of 2D theory is that it cannot account for the hull interaction between side hulls properly.

3D method based on panelization has become more popular benefiting from the development of IT technology. The basic idea of such a method is to use a number of flat panels to describe the hull surface. Based on the velocity potential assumption, each panel can be represented by a source and a set of boundary condition functions are set up to derive the intensity of the source on the panels and hence the motion and load response of the ship. Amongst various ways to solve the sources mathematically, Green function method[9] is commonly used as it is well developed and results are relatively stable.

Compared with the 2D method, a 3D method can better account for effects such as forward speed and interaction between hulls. It is therefore more suitable for investigation of wave induced loads on trimarans. Another benefit is that the 3D panel method can predict the hydrodynamic pressure at the panels around the hull which can be applied to FE panels directly, though mapping between the two meshes may be required. An example hullform of the 3D panel method software PRECAL is shown in (FIG.29).

Unlike rigid body techniques, hydroelasticity is a structures based approach where the inherent flexibility of the ship's structure is included in the calculation of the wave induced loads. The model force distributions for the principal free vibration modes are calculated using an FE model. The fluid-structure interaction problem is then solved to calculate the principal coordinates associated with each of these modes. (Principal coordinates can be thought of as 'weighting factors' for each of the modes). The total load in calculated by superposition of the contributions from each of the modes. For an introduction to the applicability and background to hydroelasticity the reader should refer to the recent LRTA paper by Hirdaris and Ge[30].

The hydroelasitic approach represents a more accurate formulation of the physics of the fluid-structure interaction than rigid-body methods. Furthermore it allows the global stresses in the structure to be calculated directly from the hydrodynamic analysis, as well as the ability to include the effects of the whipping response of the body after a slam occurrence. Using rigid body methods, these would both require reapplication of loads to an FE model of the structure. In the case of whipping, it may directly affect the calculation of the design loads. However hydroelastic calculation tools need validation on a larger scale compared with better established strip theory or 3D hydrodynamic tools.

As discussed previously, there is always the consideration of whether and how to include non-linear factors and whether to solve the problem in time or frequency domain. In reality, the wave-ship interaction is non-linear, especially in severe sea states. Ideally we should operate calculations in time-domain and account for all the nonlinear factors. Considerations are required for predictions of added mass, damping, scenarios of high speed operation, slamming as well as the departure from wall-sidedness to include non-linearities arising from hull shape, as mentioned previously. However, we have to be realistic with the capabilities of our currently available computational tools. Past experience has showed that the above mentioned rigid body hydrodynamic tools are sufficient for carrying out linear and frequency domain, or frequency domain based time domain analysis. The application of non-linear effects in the time domain requires further development and large scale validation.

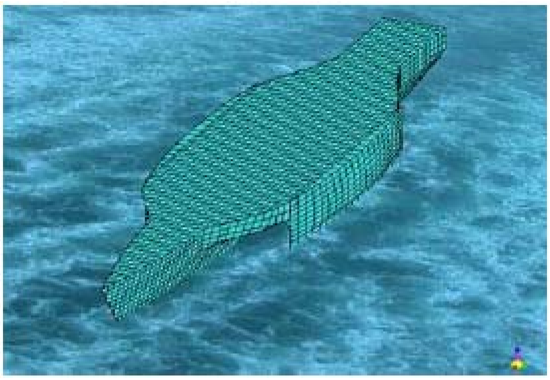


FIG.29 - PRECAL TRIMARAN HULLFORM

Model Experiments

A tank test is performed in a model basin. The waves can be either regular or irregular, and the models free running or towed. There are two different types of model basins usually for motion and load tests, a conventional towing tank which is long and narrow, or a seakeeping basin which is considerably wider and shorter.

Open sea tests are carried out at sea. The waves encountered can only be irregular since regular waves hardly exist. The model can be free running or towed.

The types of model used for the testing can be described as rigid, rigid segmented or elastic[31]. Rigid models are the simplest and measure motion response, as well as resistance and powering. For the rigid model, only the global mass properties such as mass, inertia and centre of gravity need be modelled correctly, not the mass distribution.

FIG.30 - SEGMENTED MODEL (COURTESY OF QINETIQ)

A rigid segmented model measures different components of global loads and consists of a number of rigid segments, depending on the load components to be measured. Unlike the rigid model the global mass distribution needs to be modelled. For a trimaran the main hull and cross deck nee to be segmented at the expected critical locations to measure longitudinal and transverse loads respectively. An example of a rigid segmented model is shown in (FIG.30). The model consists of 6 sections connected to a central beam with two side hulls connected to the main hull by two cross structure beams. Each side hull has a separate longitudinal beam.

An elastic model is the more sophisticated type of segmented model which additionally models the hull stiffness, in practice this is more difficult to achieve. Some institutions modelled the flexibility of the hulls by cutting the hull bodies in several sections connected by springs or steels between them, in order to investigate the springing or whipping effect[32].

Load Application to FE Model

This sub-section discusses how to apply the load responses from the direct calculation directly to the FE model.

The design value (vertical wave bending moment etc.) is intended as the extreme value expected during a 20 year operating life in the North Atlantic. This is assumed to correspond to 10^8 wave encounters or a long term probability of 10^{-8}.

Having established from either theoretical or model experiments the magnitude of the design loading on the trimaran, we can then follow a procedure similar to that described earlier to determine a characteristic wave condition which will result in the design value and then may be applied to an FE model. An outline of the stages in this process are given in (FIG.31). The only difference between the Rule load and LCF approach is that we do not calculate the LCF explicitly for each load case, instead using the time instant 't' to derive the simultaneously occurring responses for the same time instant. The simultaneously occurring responses are then superimposed for application to the FE model.

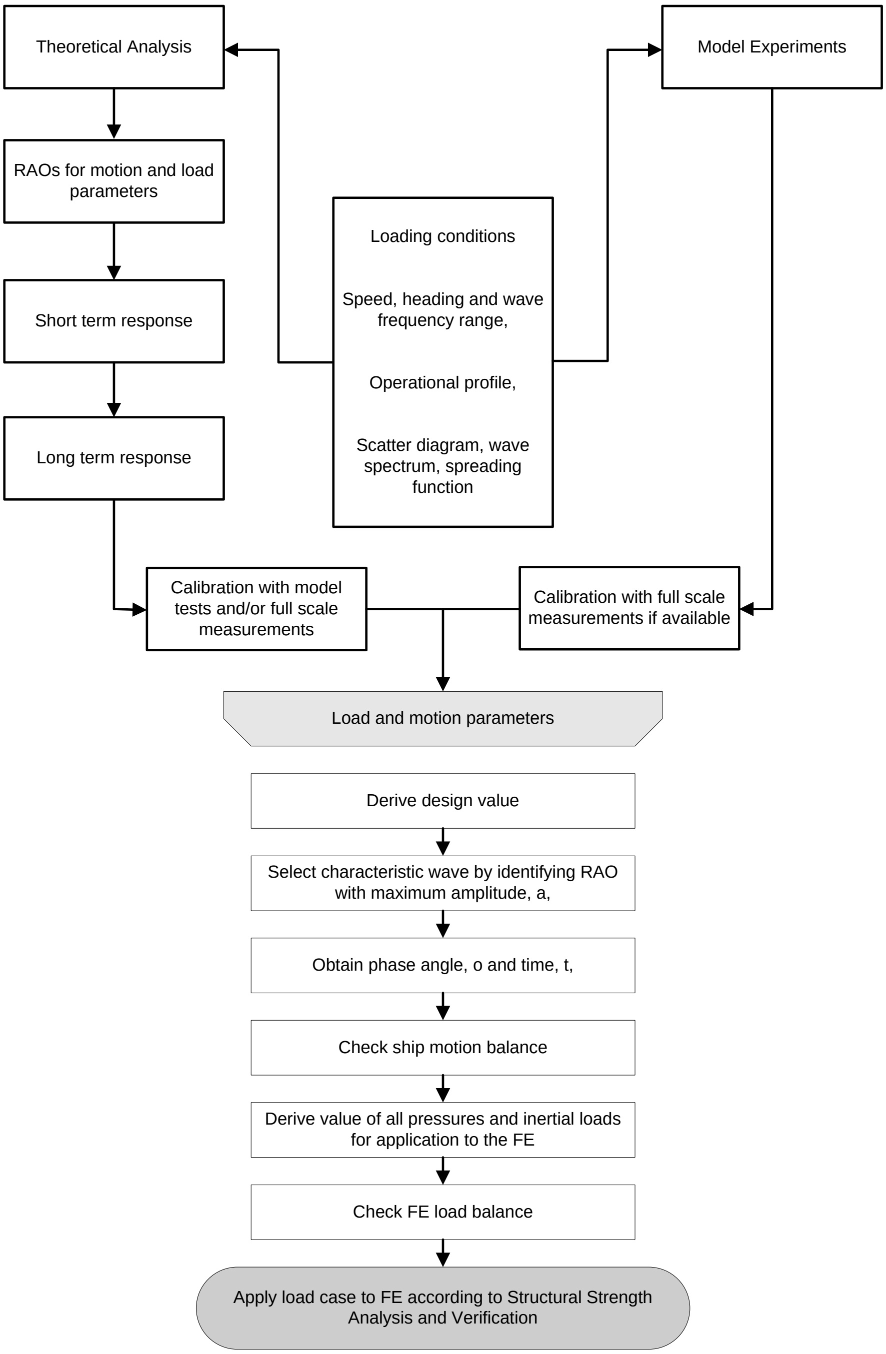


FIG.31 - DIRECT LOAD CALCULATION STAGES

CONCLUSION AND DISCUSSION

The trimaran platform has already become the basis of a large high speed vehicle ferry and a surface combatant project. Trimarans Rules developed by LR have provided in particular, Rule formulae and procedures to estimate loads applicable to this unique hullform configuration.

It has been necessary to develop loadings for the Trimaran Rules using a variety of data sources which include full scale measurements, model experiments, theoretical methods, parametric studies, service experience and existing data for both multihulls and other ship types. After calibration of these data sources parametric rule formulae have been developed to provide unique 'envelope' loadings for the anticipated size range of Trimarans. As an alternative to the Rule loads a direct calculation procedure has been developed which will enable a higher level of analysis.

UCL studies show scantlings derived from the Trimaran Rules commensurate with what would be expected for typical trimaran designs of a frigate and containership. Further in-service experience will make it possible to develop these Rules further, to reinforce or relieve the reliance on monohull and catamaran based requirements, and also investigate other areas such as fatigue design. These are challenges not just for the Rules, but also to the trimaran structures community.

ACKNOWLEDGEMENTS

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Symbols

Symbol	Definition
a_{heave}	heave acceleration, in g
a_i	RAO value of the simultaneous response at the Equivalent Design Wave period T_0
a_{max}	maximum RAO for the chosen maximised response
a_z	vertical acceleration due to heave and roll, in g
B_{mth}	main hull breath
B_{wl}	waterline breadth is the total moulded breadth of the three hulls on a waterline at the design draught, excluding tunnels
C_b	block coefficient, is the block coefficient at draught T correspondence to the waterline at the design draught, based on Rule length L_R
D	depth is measured, in metres, at amidships, from the top of the main hull keel plate to the moulded deck line of the uppermost continuous deck, at the side of the side hull. This uppermost deck is to be a deck shared with the side hull s and cross-deck structure
D_f	longitudinal distribution factor

F_f	hogging or sagging correction factor based on the amount of bow flare, stern flare, length and effective buoyancy of the after portion end of the ship above the waterline
F_{fh}	hogging correction factor
F_{fs}	sagging correction factor
F_{serv}	service group factor, based on wave height
H_f	horizontal bending moment distribution factor
H_{max}	wave height of the Equivalent Design Wave, in meters
L_f	wave coefficient
L_R	Rule length, the distance, in metres, on a waterline of the main hull, at the design draught from the forward side of the stem to the after side of the rudder post or to the centre of the rudder stock if there is no rudder post. L_R is to be not less than 96 per cent, and need not be greater than 97 per cent, of the extreme length on a waterline of the main hull at the design draught. In vessels without rudders, the Rule length, L_R, is to be taken as 97 per cent of the extreme length on a waterline at the design draught. In vessels with unusual stem or stern arrangements, the Rule length will be specially considered
L_{sh}	side hull length, L_{sh}, is the distance, in metres, measured on a waterline at the design draught, from the fore side of the stem to the after side of the stern or transom of the side hull. For stepped side hulls above the waterline, and where the side hull extends significantly forward of the stem, or aft of the stern or transom, the side hull length will be specially considered
M_{dy}	dynamic bending moment, in kNm
M_h	horizontal wave bending moment, in kNm
M_{lt}	Longitudinal torsional moment, in kNm
M_o	a wave bending moment, in kNm
M_{sph}	hogging splitting moment, in kNm
M_{sps}	sagging splitting moment, in kNm
M_{sw}	still water bending moment, in kNm
M_{swh}	still water hogging bending moment, in kNm
M_{sws}	still water sagging bending moment, in kNm
M_{tot}	total bending moment, in kNm
M_{tt}	transverse torsional moment, in kNm
M_w	vertical wave bending moment, in kNm
M_{wh}	hogging vertical wave bending moment, in kNm
M_{ws}	sagging vertical wave bending moment, in kNm
M_{st}	static bending moment, in kNm
P_i	the magnitude of the simultaneously acting response at t_{max}
Q_{sph}	splitting shear force, in kN, in the hogging condition
Q_{sps}	splitting shear force, in kN, in the sagging condition
R_{max}	rule design value or directly calculated long-term value for the chosen maximised response
R_{max-i}	rule design value or directly calculated long-term value for the simultaneously acting response
T	draught, in metres
T_0	period of the Equivalent Design Wave, in seconds
T_f	longitudinal torsional bending moment factor

t_{max}	the time instant when the maximised response have the maximum RAO value, in seconds
V	forward speed of the trimaran, in m/s
V_{cd}	volume on the cross-deck structure, in m^3, on one side of the ship. The inside and outside boundaries of the cross-deck structure are to be taken as the vertical lines extending upward from points O and I
V_{mhs}	volume of the main hull, in m^3, which extends the length of the side hull. The outside boundary of the main hull is to be taken as a vertical line extending upwards from point O
V_{sh}	volume of one side hull, in m^3. The inside boundary of the side hull is to be taken as a vertical line extending upward from the point O
W_{sh}	total weight of one side hull, in tonnes, including lightship weight and deadweight. The inside boundary of the side hull is to be taken as a vertical line extending upward from the point O
x_{sh}	longitudinal distance, in metres, from mid-length of the side hull to mid-length of the main hull where distance is positive for side hull mid-length aft of the main hull mid-length
Y_{cs}	transverse distance from centreline to the centre of area of a cross-section A_{cs} taken at mid-length of the side hull
Y_o	distance between the centreline of the main hull and point O
Y_i	distance between the centreline of the main hull and point I
Y_{sh}	distance, in metres between the centreline of the main hull and the transverse centre of area of the side hull
Z_{cdb}	section modulus at the bottom, or wet deck, of a longitudinal section of the cross-deck structure, extending only the length of the side hull, in cm^3
Z_{cdt}	section modulus at the top, typically the main deck, of a longitudinal section of the cross-deck structure, extending only the length of the side hull, in cm^3
Δ	displacement is the total displacement of the side hull and the main hull in tonnes
Δ_{sh}	displacement is the total displacement of one side hull in tonnes
ε_i	phase value of the simultaneous response, in degrees or radius
ε_{max}	phase value of the chosen maximised response, in degrees or radius
P	water density in t/m^3
W	wave frequency, in rad/s
W_e	encounter frequency, in rad/s
Ψ	wave heading to the ship, in degrees

Authors' Biographies

Tim Blanchard graduated from Newcastle University in 1995 and joined Lloyd's Register Ship Emergency Response Services. In September 1997 he transferred to the Graduate Training Scheme and following completion of his training spent three years in the Passenger Ship and Special Service Craft Hull Structures Approval Group. Tim is now working as a Senior Project Engineer in the

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Chunhua Ge gained her Bachelor Degree in Naval architecture and Ocean Engineering in 1995, and her Master degree in Mechanics of Ship structures in 1998 from Shanghai Jiao Tong University. She was awarded her Ph.D degree in 2002 for her research in slamming-induced hydro-elastic responses for catamarans in Norwegian University of Science and Technology. In 2002 she joined Lloyd's Register as a Hydrodynamicist in the Research and Development Department, specialising in hydrodynamic related issues and rule development for different types of ships.