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CONTENTS

1. BACKGROUND

A major requirement in the design of glass reinforced plastic (GRP) naval vessels is that they must withstand shock loads from possible non-contact underwater explosions of mines during the course of their service lives. The effect of these explosions on the GRP hull structure can be two-fold:

a) Primary shockwave effects leading to high accelerations, particularly at the ship's bottom, potentially causing delamination and debonding of attachments to the shell plating.

b) Hull girder whipping effects due to bubble pulse pressures, potentially causing catastrophic buckling of the ship's bottom and deck plating.

Whilst whipping analysis is now possible and forms a standard part of the design procedure for such ships, the impact of shockwave effects on the hull structure has only been studied through recourse to expensive and time- consuming shock tests. Hence there is a need to develop a theoretical, analysis model to supplement and reduce reliance on costly testing.

The hull structural response under the impact of a shockwave is transient dynamic and non-linear in nature. It is a highly complex phenomenon involving fluid-structure coupling and needs to be modelled in several stages. One important, initial aspect concerns the behaviour of the structure under large amplitude loading in a static context.

The areas of primary interest in the structure are joints (or secondary bonds). The joints could refer to connections either between two orthogonally placed plate panels as shown in Figure 1 or between the laminate of a (top-hat) stiffener and the plate panel as shown in Figure 2. Generally these joints are made by forming a boundary angle of varying dimensions over a flexible resin fillet which bridges between the two adjacent structural elements and provides a correctly radiused mould over which the boundary angle is laminated. In the case of a top hat stiffener this boundary angle is an integral part of the stiffener, which is moulded around a non structural foam core, again with a flexible resin fillet in the angle between the stiffener web and the stiffened structure.

Depending on their location in the ship these joints must be watertight, fire resistant, able to withstand shock loading, and retain their ability to transfer load from one structural element to another for the life of the hull, whilst remaining simple to fabricate. The correct design of these structural connections is considered a very important factor governing the performance of a hull of this kind when subjected to shock loading.

This report covers three aspects of work.

- A literature survey of work in the area of joint design, analysis and fabrication has been conducted. Preliminary conclusions with regard to their influence over the work in this project are drawn.

- A finite element based framework analysis has been conducted with a view to understanding the overall behaviour of the structure and its influence on boundary angle/joint loads.

- Based upon the above two aspects, future work directions as well as a programme of tasks have been outlined.

2. LITERATURE REVIEW

2.1 Design Synthesis Aspects - Marine Applications

One of the earliest approaches to GRP structure design is outlined in the Gibbs and Cox manual (1). This gives recommended arrangements of various joints and simple design examples. Typical material properties are listed for E-glass / polyester resin laminates with different weights. Section moduli and moments of inertia are tabulated for different geometries of top-hat stiffeners. The dimensions of the boundary angles, it is stated, "should be minimum consistent with strength requirements". However, no specific procedures concerning joint design are elaborated.

Early work in this country centred around the design of GRP minehunters (2-4). This covered aspects of structure design from a number of viewpoints such as strength, stiffness, stability and material failure modes. The philosophy of the design of frame-to-keel and shell-to-bulkhead connections is elaborated. Design, in this case, was based to a very large extent on the results of experimental tests. Criteria for assessing structural adequacy included static strength, fatigue and shock blast resistance.

Work on the minehunter programme formed the impetus to the drawing-up of naval engineering standards (5). These rules for naval ships are based on extensive experimental and analytical work carried out by the Admiralty Research Establishment, Ministry of Defence, Vosper Thornycroft and others. The standards prescribe minimum limits to various scantlings. Boundary angle thickness, for example, is specified to be at least half (and preferably two thirds) the thickness of the thinnest member. Flange overlap dimensions, the lay-up and stacking sequence are also specified.

With regard to merchant ships, yachts and small craft, design guidance is sought primarily from classification society rules. Lloyd's rules (6) state that where the length of a vessel exceeds 30 m, "the scantlings are to be determined by direct calculation". For smaller boats, the rules specify explicitly the flange width of top-hat stiffeners and weight of laminate forming the boundary angle.

American Bureau of Shipping rules (7) are applicable to GRP vessels under 61 m in length. Proportions of stiffeners are derived in terms of thicknesses of stiffener crown and webs, with these being dependent on section modulus requirements. Boundary angle thickness is again given as a function of the members at the joint.

Det Norske Veritas (8) has no formulae for direct derivation of scantlings but has tables of maximum allowable stresses from which stiffener moduli can be calculated. Design loads and modelling considerations (essentially simple beam theory) are also mentioned

A recent Ship Structures Committee report (9) uses diagrams from Reference 1 and states similarly that dimensions of boundary angles should be minimum consistent with strength requirements. It refers to classification society rules (7) for top-hat stiffener scantlings. It also outlines a simple procedure based on beam theory to determine section modulus and stiffness values for stiffeners.

2.2 Production Aspects and their Influence on Design

Descriptions of general production considerations are well documented in References 2 and 10-12. The influences of production on design are manifold. The more important factors, relevant in this context, are discussed below.

The most overriding production consideration is quality. This must be maintained if some assumptions used to design the ship are to hold true. This reflects directly back to design because the structure needs to be as simple as possible to ensure that quality can be maintained without requiring increased labour and, therefore, cost.

In a historical context, the major change in stiffener attachments to plating has been the replacement of bolted connections in case of early minesweepers (2) by bonded, flexible fillets in modern minehunters (11) - see Figure 3. The latter have proved to be cheaper both in terms of initial material cost as well as the labour required to fit them. Extensive testing (13-17) has shown that flexible fillets do have sufficient strength to sustain most static load conditions. From a design viewpoint, the studies have concluded that incorporation of flexible resin plies on the insides of the boundary angle is desirable - see Figure 4. However, production considerations such as cost and quality assurance have precluded its use.

Another design variable to be affected by production considerations is the radius of the fillet at the boundary angle. Sharp corners and areas in which rollers cannot get into need to be avoided. A bottom limit to corner radius owing to roller size does exist though this is strictly not a restriction because a minimum radius is required for structural reasons (18,19). From a production viewpoint, sharp corners are locations where undercutting caused by the action of the roller could cause resin starvation and, potentially, lead to delamination.

Resin levels are also problematical in areas of complex curavature and geometry such as at intersections of stiffeners and at stiffener endings (near hull-bulkhead connections, for example). A further factor needing to be considered is the stacking sequence and lay-up procedure. For example, chopped strand mat layers tend to be richer in resin than woven roving ones; with respect to through-thickness capabilities, it has observed that CSM layers fail at a load lower than that for woven roving layers. This demonstrates that the stackling sequence affects material properties, response patterns and consequently does need to be recognised at the design modelling stage.

A final production consideration that affects material performance is the manner in which the secondary bond, between the boundary angle and the base laminate, is affected. Secondary bonds require special attention; the base laminate needs to be prepared by careful wiping with a solvent such as triclorethane (20). Some problems have been encountered. In an earlier class of vessels, premature fatigue failure has been noticed (21). In another context, delamination of unidirectional fibres on the flange of a channel stiffener has been noticed (22). However, careful studies (23) have shown that correct preparation of the laminating surfaces does result in the through-thickness strength of the boundary angle (and secondary bond) to be the same as the base laminate. The principal reason for elaborating this is to ensure that this factor is considered within the analytical/numerical model.

2.3 Materials

There is an extensive body of literature concerning material types and their features with particular reference to marine applications (24-27). The two important features of particular interest in this context are the manner of calculation of mechanical properties and the prediction of failure modes and limits.

Analytical calculation of mechanical properties, starting from the simple "rule of mixtures" approach is again well documented. References 24 and 28-33 contain adequate information for the calculation of fundamental elastic properties such as Young's moduli, shear modulus, Poisson's ratios, etc. Micromechanics and laminate theories can then be used to deduce the five material stiffnesses in case of the two-dimensional, orthotropic laminate valid in a marine context. This could, theoretically, allow for variations in the properties for different plies; this is a production linked feature identified above. It is equally feasible to account for varying void contents. Obviously, the calculations will need to be correlated with practices prevalent in the industry here and consultations with Vosper Thornycroft will be required.

Failure criteria in laminates could be manifold (24,30,34,35) and include :

- longitudinal tensile mode through brittle failure, brittle failure with filament pullout or through debonding.

- longitudinal compressive mode through filament microbuckling, panel microbuckling, debonding or shear failure.
- transverse tensile mode through matrix tensile failure and constituent debonding.
- transverse compressive mode through matrix compression, shear or a combination of the two resulting in debonding.
- interlaminar shear through matrix shear, matrix shear through debonding or debonding by itself.

Expressions for calculating ply strengths in the above cases are available in References 30,34 and 36-40. Most of these expressions have been derived (and verified) from a standpoint of laminates for aerospace applications. Consequently, as far as possible, the application of appropriate expressions in the marine context will have to be validated against some practical data from a marine viewpoint (41).

Another difficulty with regard to the above referred expressions is that they are applicable, in the main, to primary bond considerations. In the boundary angle and top-hat stiffener cases, secondary bonding is involved. It was noted in the previous section that if proper production procedures are followed, secondary bond features mirror those of the base laminate itself. Thus, this assumption may help justify use of the empirical equations in this context.

A final difficulty is that the expressions mentioned above refer to flat laminates. In this particular case, the laminate at the joint is curved. This curvature does need to be accounted for in an ideal case. It has been noticed from tests that failure at the joint occurs by delamination within the boundary angle, by failure within the fillet leading to peeling along (or close to) the secondary bond or by a combination of the two. Further investigations are required in this context. The observations recorded in References 18 and 19 will be particularly relevant here.

2.4 Analytical and Numerical Techniques

As in the above case, there is an extensive body of literature covering the analysis of anisotropic plates and laminates. Standard text-books (24,30,31,40,42) cover the basic elements, in a general context, with thoroughness. Complexities arise when addressing non-linearities: these can be due to two main causes, namely material dependence and geometry (or load) dependence. Both could have an impact in this project.

The nature of FRP, with differential properties for the fibres, reinforcements and the net laminates is such that the laminate could exhibit considerable reserve strength and stiffness after first ply failure (43). However, each successive localised failure does have an impact on the local and global properties - see Figure 5. The behaviour pattern in this case depends upon local stress fields, with stress concentrations having a significant impact (44,45). The behaviour pattern is also affected if the laminate considered is bimodular, with different tensile and compressive properties (46-48). Konishi (49) has attempted to relate deflection and stress fields to delamination growth in laminate panels.

The field of geometric non-linearity is treated most exhaustively by Chia (50) in contyext of both isotropic and anisotrpic plates and laminates. Special treatments of non-linearities in othotropic panels (51-53), transversely loaded (54) and under combined in-plane and transverse loads (55) are also available. In addition to such cases, which may or may not result in material failures, the behaviour is also affected by stability considerations; the post-buckled regime has considerable influence on the stiffness of orthotropic panels (56).

In terms of numerical techniques, the field of composite finite elements is again well researched. It has reached a stage where many commercial packages are available for standard computations (57). In this University, access is available to the ANSYS suite of programs (58).

In context of joints and top-hat stiffeners, specific and detailed studies are rare. An early approach (4), in a marine context, was through the use of plain strain finite elements. A detailed stress pattern was derived, though the elements used may not have permitted a full coverage of all composite material related properties. In Reference 59 the authors have attempted the modelling of a spar/wingspan joint in an aircraft. Failure criteria used included maximum stress, maximum shear strain in insert and Tsai-Wu. More recently, in reference 60, the authors have examined marine sandwich joints. The study has incorporated numerical and analytical approaches into the design. All three of these studies have been conducted in a small deflection, linear elastic analysis domain. The latter two studies have concluded that, at least first ply, failure can be detected through such analysis. However, as indicated in References 34,37 and 43 the behaviour pattern changes appreciably after this first failure. Therefore there is a need to model such changes which could introduce potential non-linearities into the system.

2.5 Summary

The principal conclusions that can be drawn from the review are listed below.

1) Joint and top-hat designs are synthesised using, at best, a beam theory approach. Detailed considerations of load, response limits and modelling, production and materials make-up are not always addressed in an explicit manner.

2) Some production considerations such as fillet radius, lay-up sequence and preparation of base laminate are especially important in joint design.

3) There are adequate sources of information to model most in-plane and through-thickness properties of laminates. However, some correlation in a marine context will be required insofar as the latter is concerned.

4) There is an extensive literature base for modelling non-linear behaviour of plates and lamintes. Their application to joint design, however, has been limited.

3. PRELIMINARY F.E. MODELLING

As indicated in the previous chapter and in the introduction the scope of the work includes development of an analytical tool and validation of results using both experimental and numerical models. The purpose of this chapter is to outline the capabilities of an existing F.E. package, within the University, ANSYS (58).

3.1 The Hull-Bulkhead Connection - Numerical Study

The first task carried out in this project has been to clarify the problem with respect to the boundary angle at a hull-bulkhead joint. Specifically, the intention has been to understand the magnitudes of loading at a joint under static (and psuedo-static) load regimes, implications of boundary conditions and geometric extent of a more detailed model for use at later stages.

For this, the structure in two adjacent compartments, as illustrated in Figure 6, has been represented through simple two-dimensional beam elements with three degrees of freedom at each node. Some assumptions regarding the modelling are outline below.

1) Geometric properties have been calculated on the basis of stiffeners being "lumped" together with the attached plating.

2) The bottom shell plating has been assumed to be flat and perpendicular to the bulkheads. In reality the shell is slightly curved and there is discernible deadrise.

3) The material in the beam is taken to be linear isotropic. The properties ascribed to the elements have been taken from a Vosper Thornycroft source (61).

4) The boundary angle is assumed to have similar properties as the laminate, inspite of there being flexible resin and CSM layers in the former.

5) The deck and adjacent bulkheads have been assumed to act as rigid constraints on the central bulkhead and shell respectively.

The model has then been loaded in several different ways as outlined below :

a) Peak over-pressure assuming a 200 kg mine exploding at a stand-off of 30 m (62).

b) Hydrostatic loading caused by a L/10 wave.

c) As (b) but with compartment flooded.

d) As (b) but with pin-ended joints at ends of the shell (rather than fully fixed ones).

e) At design draft and an 8 m head in compartment.

f) As (e) but with ship in drydock, i.e. no load on bottom shell due to hydrostatic pressure.

g) As (b) but with with very low modulus next to the joint.

The distribution of bending moment along the bottom shell for the above cases is shown in Figures 7(a)-(g). In cases of symmetrical loading, the bending moment diagrams are similar as illustrated in Figures 7(a)-(d) and 7(g). The magnitude of the peak bending moment at the join of the bottom shell and the central bulkhead obviously varies in proportion with the load. The peak value in case of the (static) shock load is about 80 MN.m while under other normal load conditions the maximum value of the peak bending moment is about 0.11 MN.m - in case (d). The region of zero bending moment is about 1.5 m on either side of the central bulkhead. In case of the compartment flooded - cases (e) and (f) above - the distribution of the bending moment is unsymmetrical about the central bulkhead. In this instance, the region of zero bending moment is about 0.5 m on the "loaded" side of the bulkhead.

Along the central bulkhead, there will be no bending moment in cases (a),(b),(d) and (g) because of symmetrical loading on the bottom, i.e. the bulkhead behaves as a strut. In cases (c) and (f), as shown in Figures 8 (a) and (b), the zero bending moment region is about 300 mm above the bottom shell. In case (e), with the compartment filled with water and an operational head of 8 m of water, the zero bending moment is about 500 mm above the bottom shell - see Figure 8(c). The maximum value of the bending moment at the join of the central bulkhead and bottom shell, along the bulkhead is 0.05 MN.m - case (f).

This simple analysis gives a limit to the geometric size of the joint. The maximum length of the flange, along the bottom shell and on one side of the bulkhead, is 1.5 m while the maximum length of the web is 0.5 m. This will be used in detailed models in later studies.

3.2 ANSYS F.E. Package - Composites Capability

The F.E. package available for general use in the University of Southampton is ANSYS (58). The choice of elements within this package is such that composite structures can be modelled in a number of ways. The three most recommended elements are layered shells/solids. Each of these is discussed briefly below.

STIF46 is an 8-node, isoparametric element designed to model layered thick shells or solids. The principal features are shown in Figure 9. The element is defined by eight nodal points, various relative layer thicknesses, layer material direction angles and orthotropic material properties. The output includes normal strains and stresses and shear stresses. The latter form the basis of interlaminar shear stress values. Upto 7 failure criteria can also be defined; all these are based on pre-defined maximum stress or strain values. One limitation is that there are only three (translational) degrees of freedom per node.

STIF91 is an 8-node isoparametric layerd shell element with six degrees of freedom per node and capable of taking in 16 different material layers - see Figure 10 for principal features. The input, apart from nodal locations, includes various layer thicknesses, material direction angles and material properties. Notably, each layer of the laminated shell may have a different thickness. The output includes all principal stress and strain components. No failure criteria, however, are defined.

STIF99, whose major features are shown in Figure 11, is an extension of STIF91 in which upto 100 different material layers are permitted. The major advantage this offers over STIF91 is in terms of failure criteria. The user has a choice. Incorporated within the program are three pre-defined criteria namely maximum strain, maximum stress and Tsai-Wu. Alternatively, the user can define upto six other criteria. The output is similar to the previous case.

A preliminary study has been made comparing the three element options. At this initial stage, it has been decided to opt for STIF46 for a detailed modelling exercise because of one major reason. This concerns the ability to model a joint between the base laminate and a boundary angle. In case of STIF91 and STIF99, the nodes are located at mid-thickness. Consequently, it is difficult to specify a linkage between a mid-thickness node in the base laminate and the corresponding node in a boundary angle. STIF46, on the other hand, offers no such disadvantage. Hence, it has been chosen for the modelling of detail in the boundary angle region.

3.3 Detailed Model of the Boundary Angle Joint

Using STIF46, a detailed test model of a joint has been created. The details of the geometry are shown in Figure 12. The material properties used in this case are as per standard Vosper Thornycroft practice for the minehunters. The load has been of a direct, tensile nature applied at an orientation of 45 at the tip of the web. The model has been constrained at the flanges: the edge on one side of the flange has been fully fixed while the edge on the other side has been constrained in the vertical direction.

A typical deflected shape is shown in Figure 13. The load-displacement relationship is shown in Figure 14. A stress contour plot is shown in Figure 15. These are preliminary results. The detailed model used here is crude. It requires refinement in the boundary angle region and with respect to the base laminate especially in the vicinity of the boundary angle. Neverthless, this crude, test model does indicate that the approach adopted for this numerical part is valid and worthy of further attention and study.

4. PROPOSED CONTINUING WORK PROGRAMME

Further work in this regard concerns three aspects and each of these is briefly dealt with below.

4.1 Analytical Approach

Based on the literature review, it has been decided to investigate two principal ways of modelling joints. The first is through the use of an analogy of the boundary angle to a curved beam. A simple model on the basis of an earlier study (63) of wooden and fibre reinforced composite bends is currently being investigated. Initially it is proposed to create a strength model. On successful evaluation of this, various failure criteria (34,37,38,39) can be incorporated. If results from this are realistic (in comparison with numerical models), then the possibilities of extending the theory into non-linear domains (50,53) will be studied.

The second approach is to treat the boundary angle as a segment of a cylinderical shell. Classical, laminated, anisotropic shell theory (64,65) and its relevance can be tested. The extent and depth to which this study is conducted will depend upon progress in the above curved beam case and the consequent remaining time/resources in the project.

In either case, the ultimate goal is to create a set of algorithms which will be able to predict boundary angle behaviour. The approach will be compared and checked against numerical and experimental studies outlined below.

4.2 Numerical Approach

This will be continued, initially, on the basis of the detailed model outlined in Section 3.3. The tasks will include the following.

- Creation of a more refined mesh in the region of the joint. The element sizes will be optimised both in the boundary angle itself and in the base laminate.

- Validation of the model against existing experimental data in a ship context (18,19). The data, in this case concerns primarily displacements; hence this will provide only an initial check on the accuracy of the model.

- Further validation of the model against more detailed studies involving stress measurements and large displacements as well (66). It is further anticipated that this model will be compared against tests carried out specifically for this project (see Section 4.3 below).

- Finally, use of the validated model to look at behaviour patterns in the post-initial failure domain. This will require material properties in the non-linear region (43-46), full validation of which may not be feasible in context of this project. However, use will be made of existing data (41).

The "validated" F.E. model can then be used as a basis to investigate variations in geometry, material lay-up and load patterns. The results will also provide a benchmark against which the analytical model can be compared.

4.3 Experimental Work

The resources available for experimentation in this project, in terms of technician support, equipment budget and material samples, are limited. Consequently, the test programme needs to be carefully planned to tie-in with the work mentioned in Section 4.2 above. It is anticipated that the testing will involve Tee joints of (relatively simple) variable lay-up and geometry configurations. The samples will be tested in a manner similar to earlier studies (18,19). It is expected that measurements will include load, displacement (at appropriate location) and, perhaps, surface strain at one important location.

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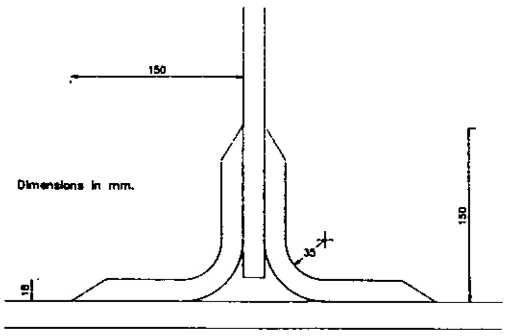


Figure 1: A Typical Shell/Bulkhead Connection

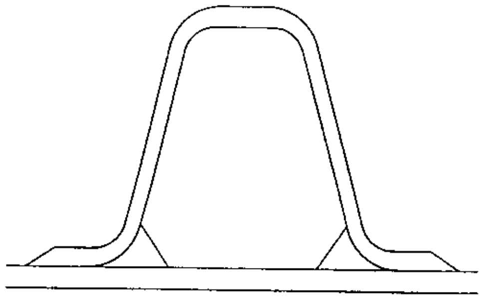


Figure 2: A Typical Top-Hat Stiffener

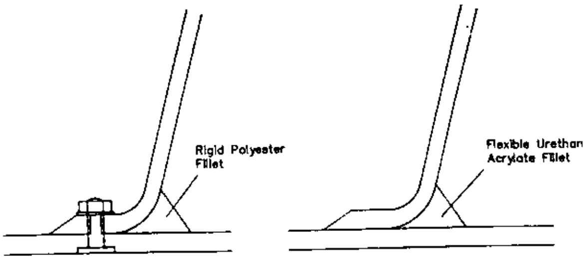


Figure 3: Alternative Boundary Angle Attachments

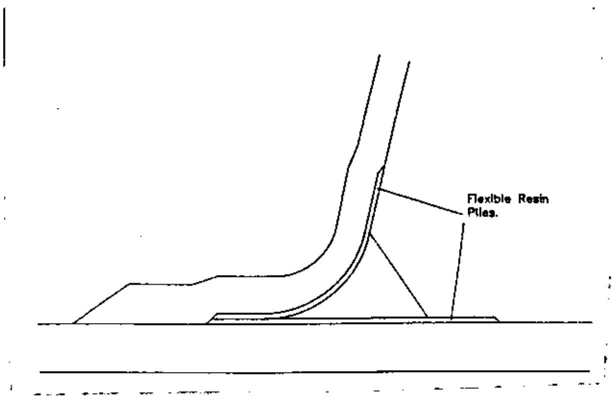


Figure 4: Incorporation of Flexible Resin Plies in Boundary Angle

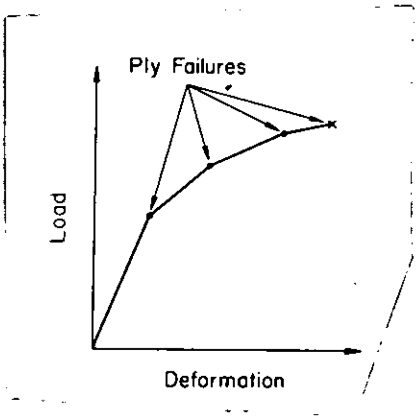


Figure 5: Variation in Stiffness Characteristics after Successive Ply Failures

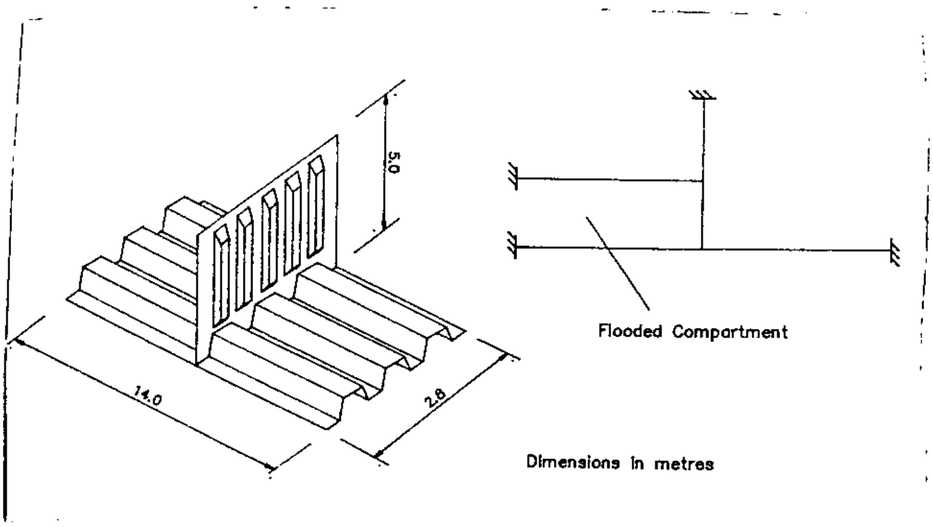


Figure 6: A Simple 2D Framework Model

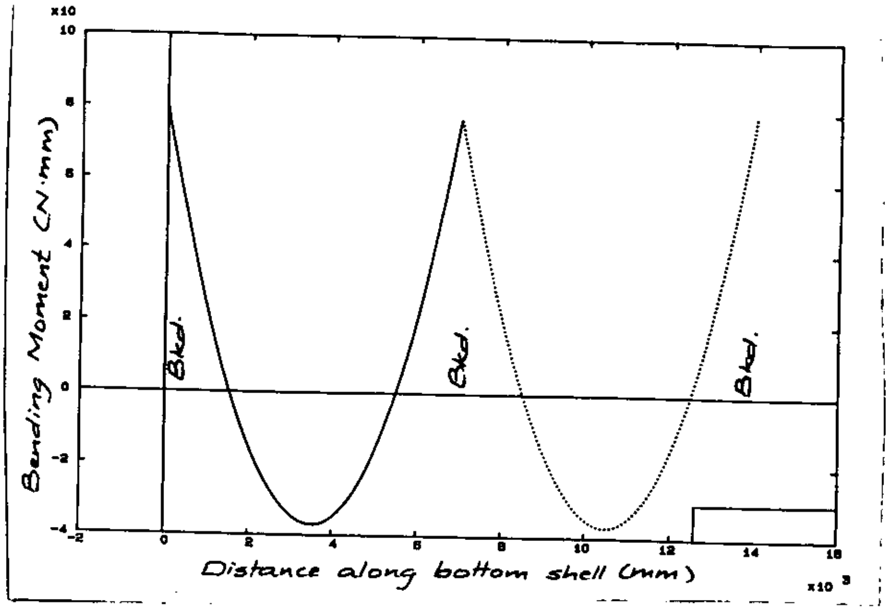


Figure 7a: B.M. Diagram - Shock Load

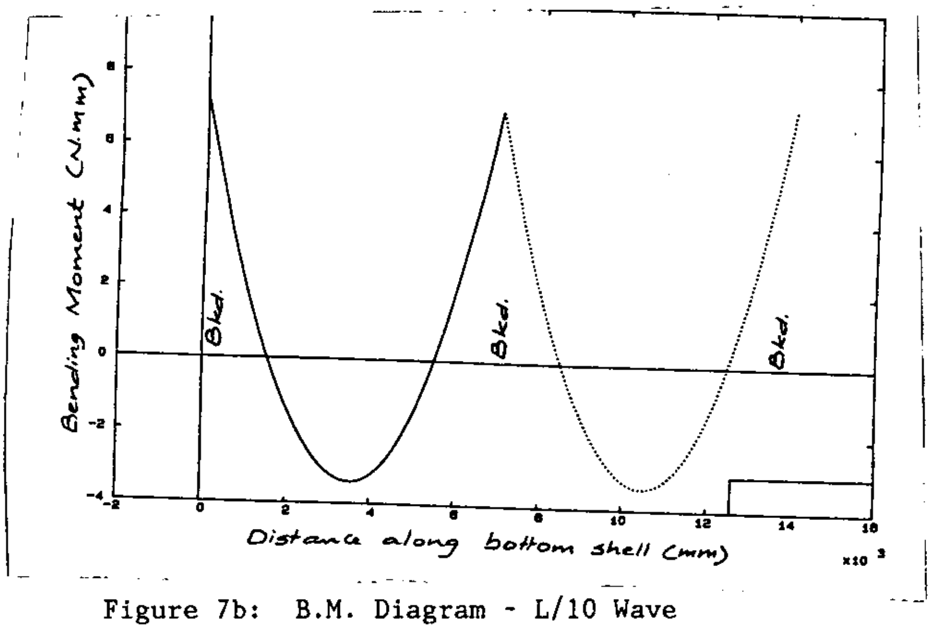


Figure 7b: B.M. Diagram - L/10 Wave

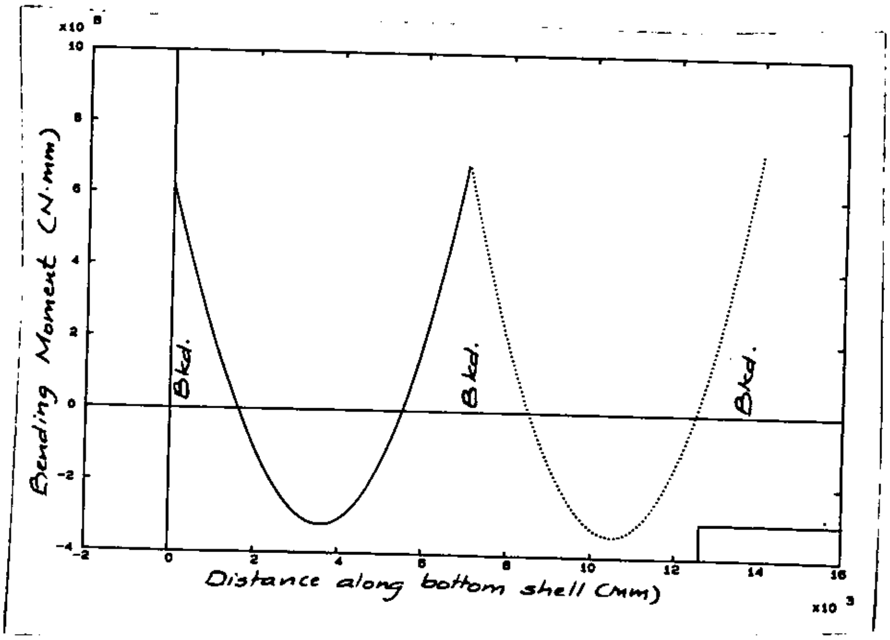


Figure 7c: B.M. Diagram - L/10 Wave and Compartments Flooded

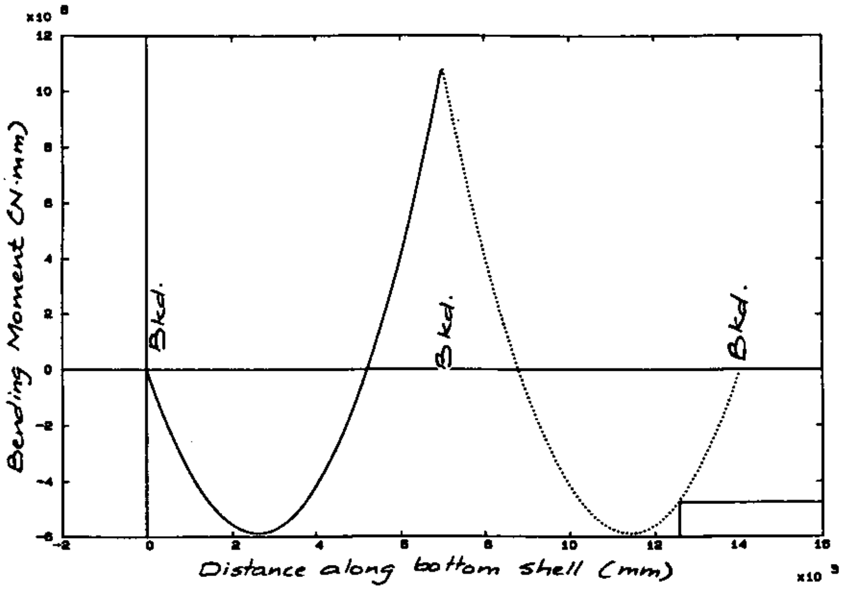


Figure 7d: B.M. Diagram - L/10 Wave with Attachments at Remote Bulkheads Assumed Pin-Jointed

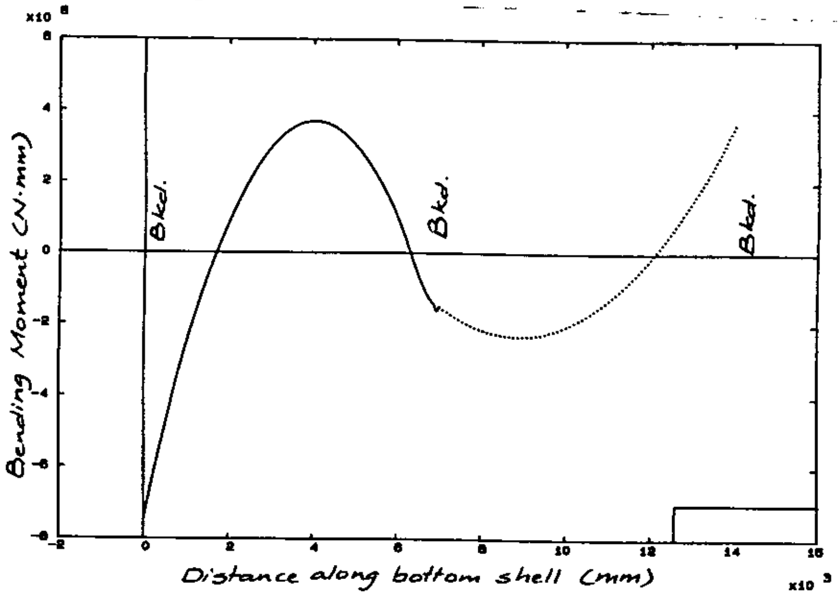


Figure 7e: B.M. Diagram - Design Draft and Tank Flooded

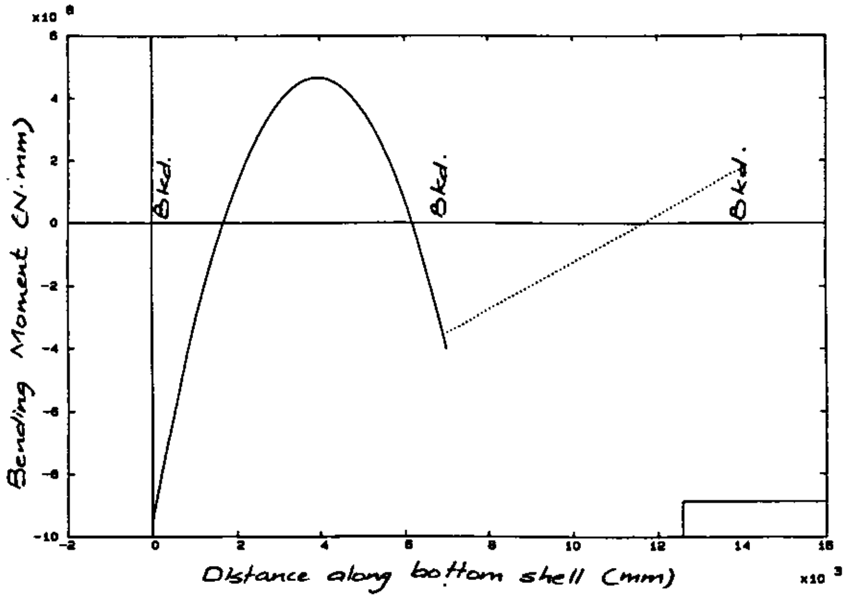


Figure 7f: B.M. Diagram - Drydock Condition

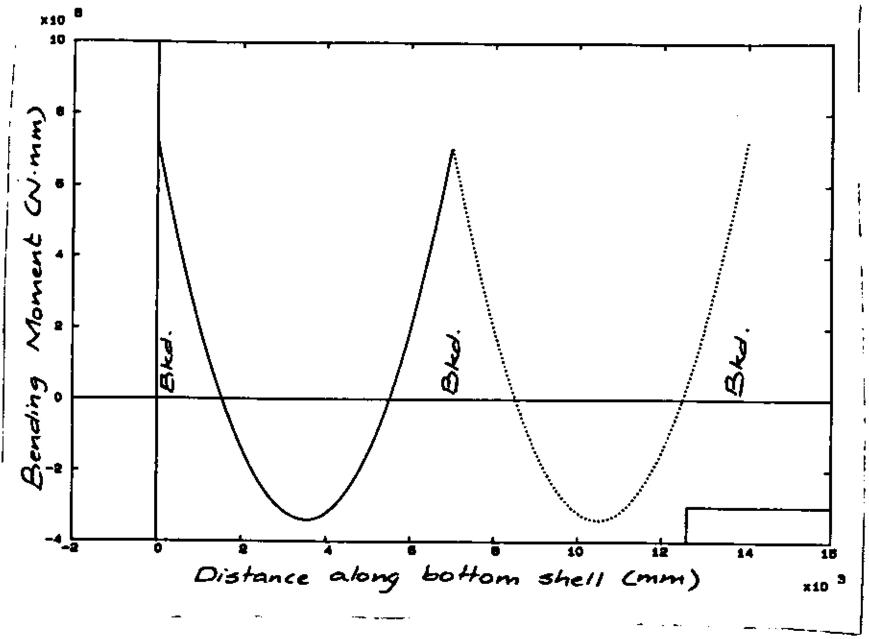


Figure 7g: B.M. Diagram - L/10 Wave with Low Modulus at Attachments

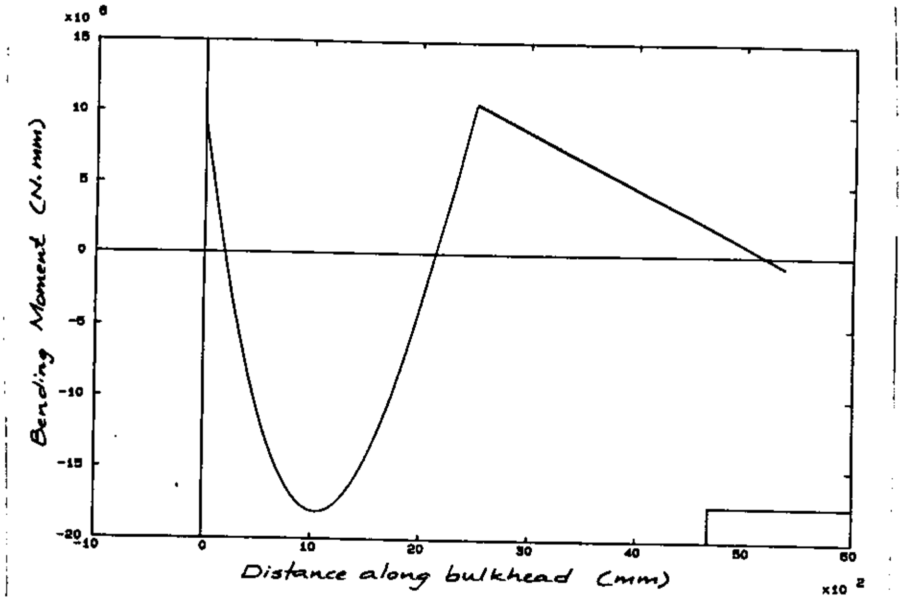


Figure 8a: B.M. Along Bulkhead - 1/10 Wave and Compartment Flooded

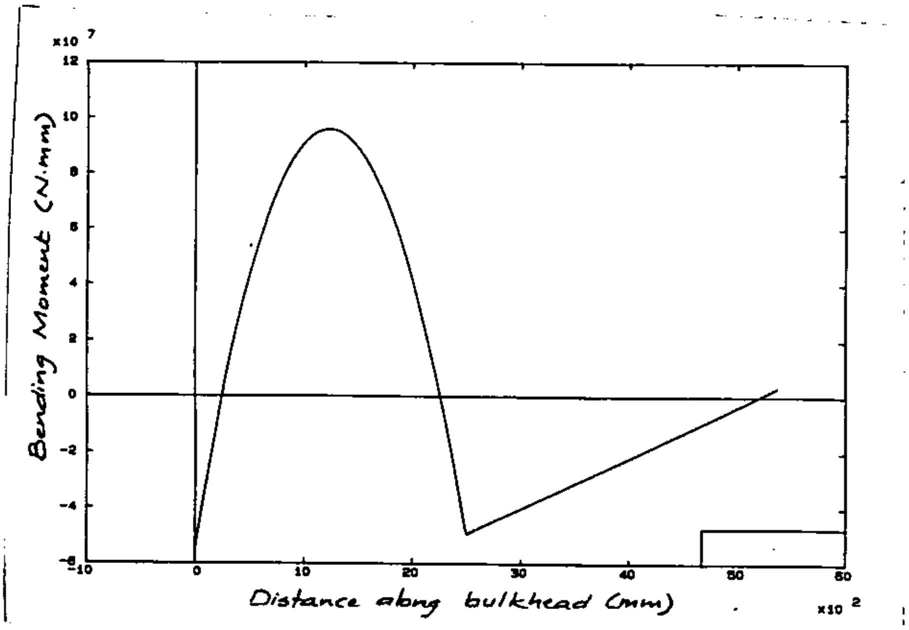


Figure 8b: B.M. Along Bulkhead - Drydock Condition

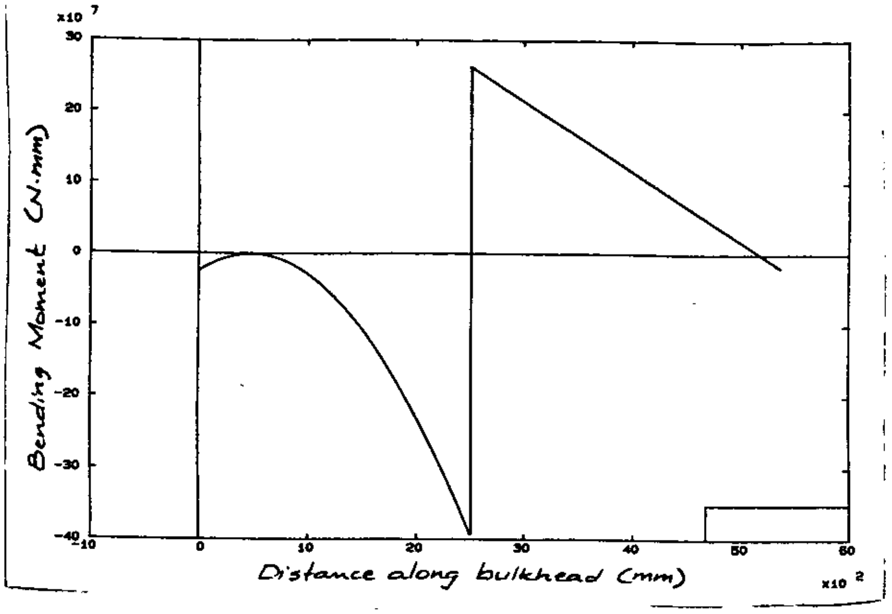
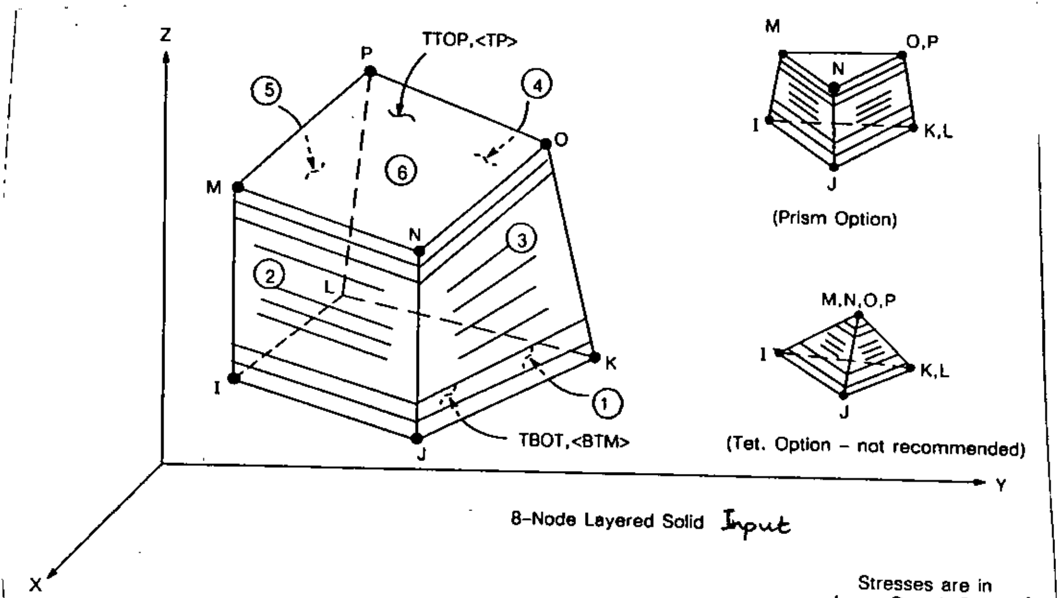
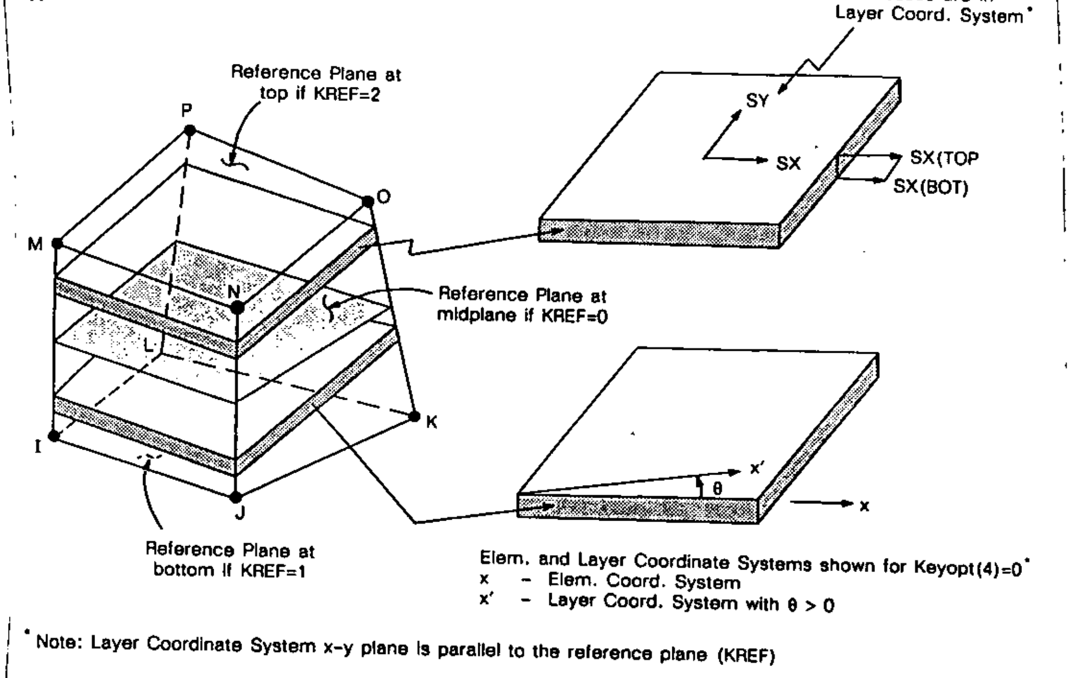


Figure 8c: B.M. Along Bulkhead - Design Draft and Tank Flooded



8-Node Layered Solid Input



Elem. and Layer Coordinate Systems shown for Keyopt(4)=0*
x - Elem. Coord. System
x' - Layer Coord. System with θ > 0

* Note: Layer Coordinate System x-y plane is parallel to the reference plane (KREF)

8-Node Layered Solid Output

Figure 9: STIF46 Characteristics

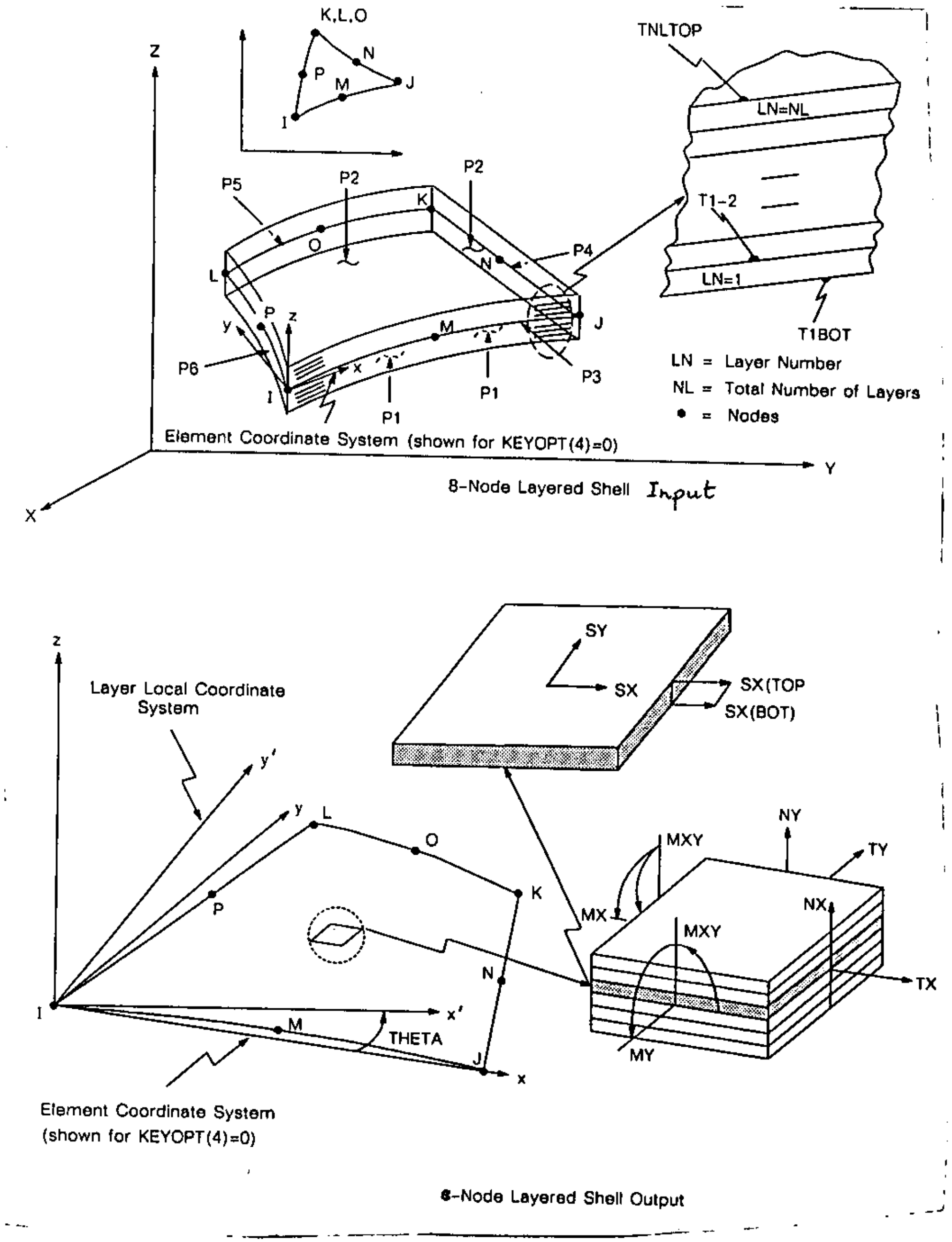


Figure 10: STIF91 Characteristics

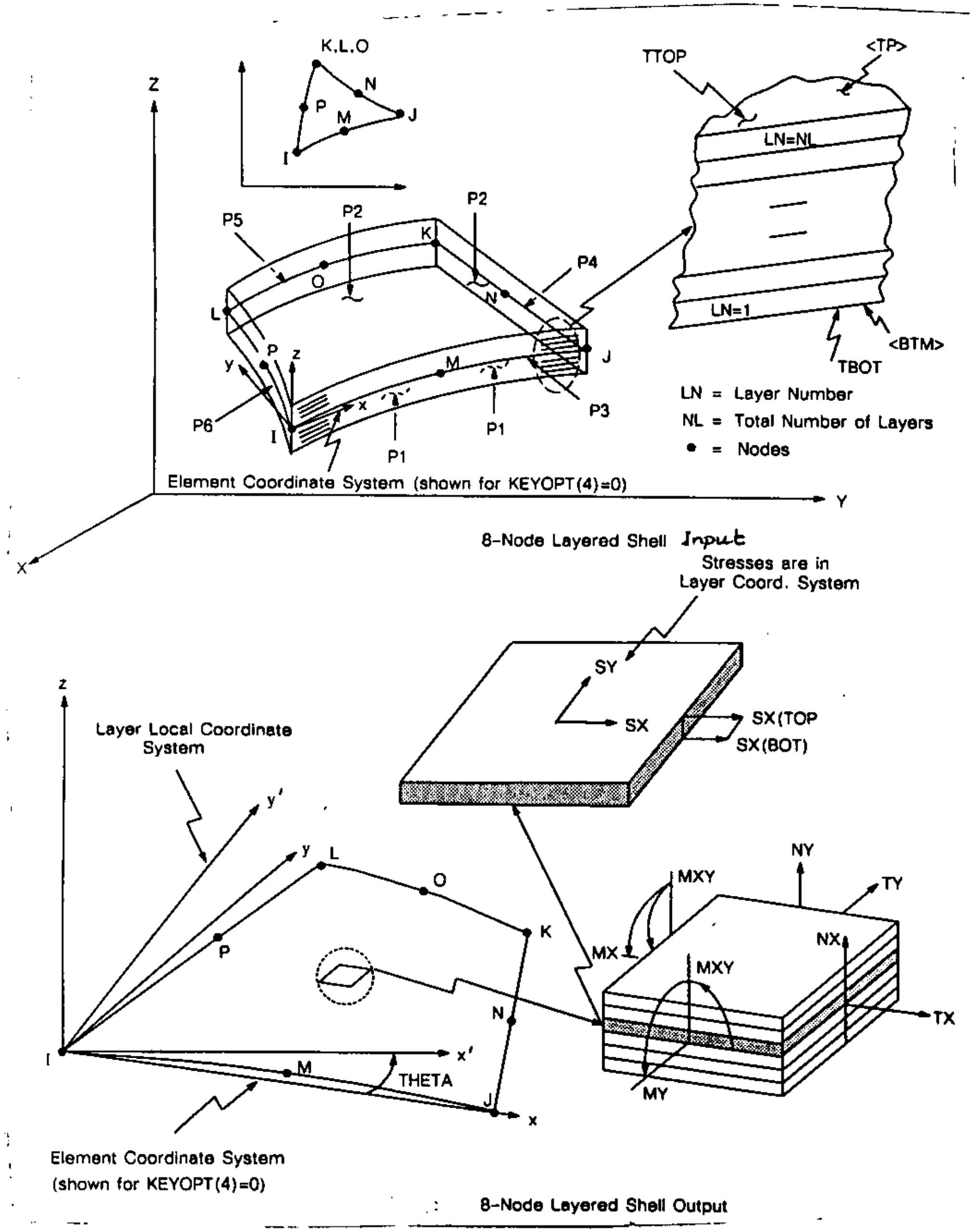


Figure 11: STIF99 Characteristics

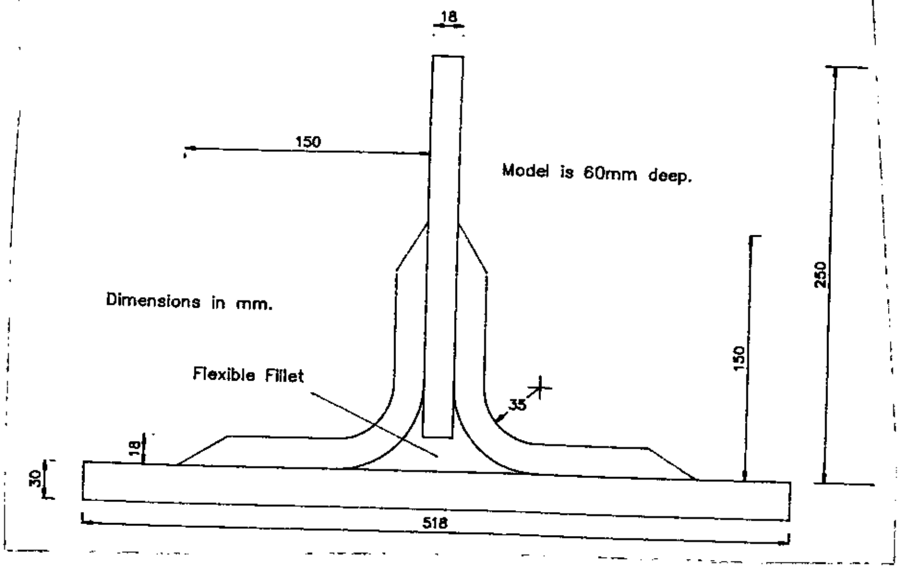


Figure 12: Geometry of the Preliminary Hull-Bulkhead Model

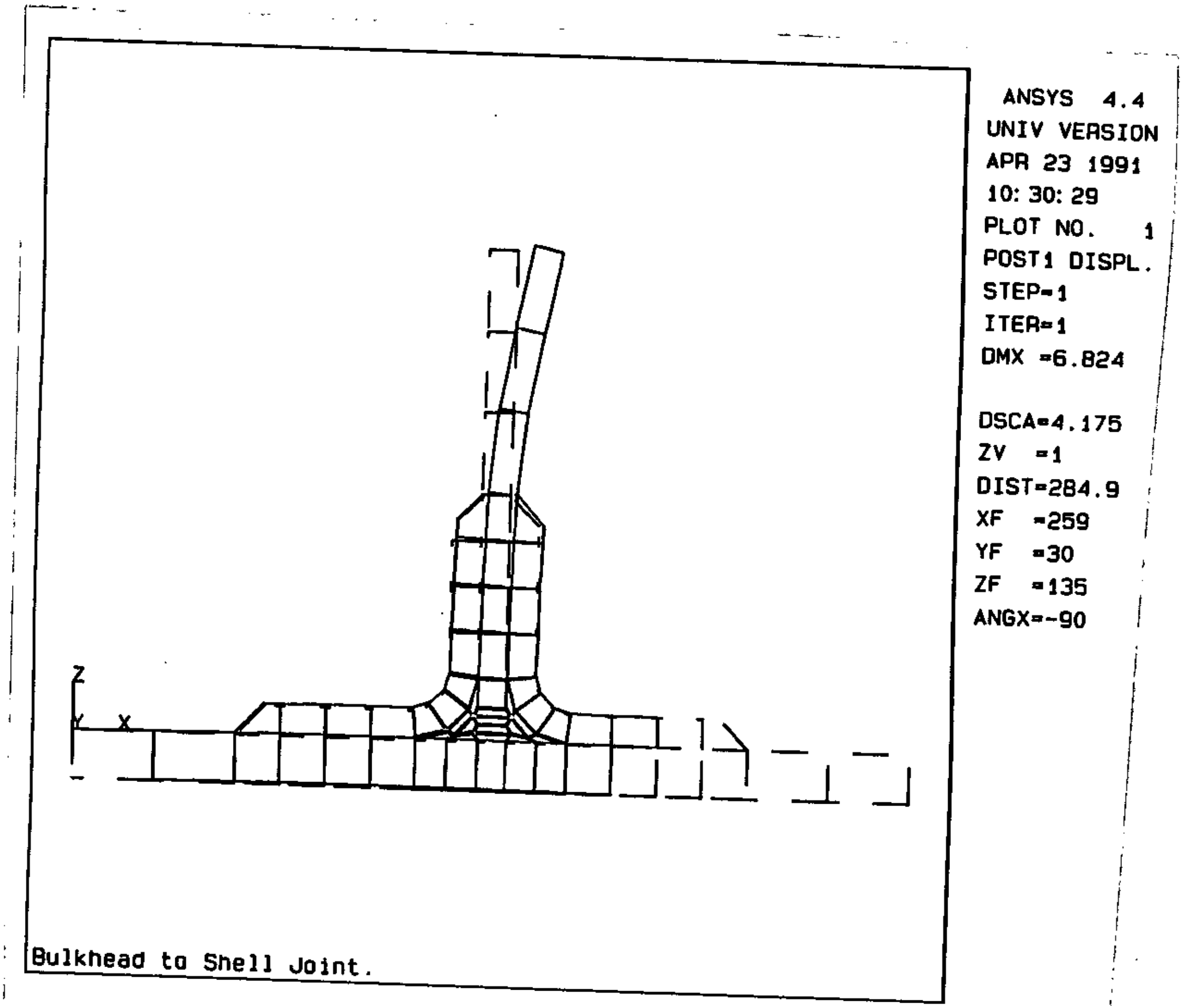


Figure 13: A Typical Displaced Contour of Joint

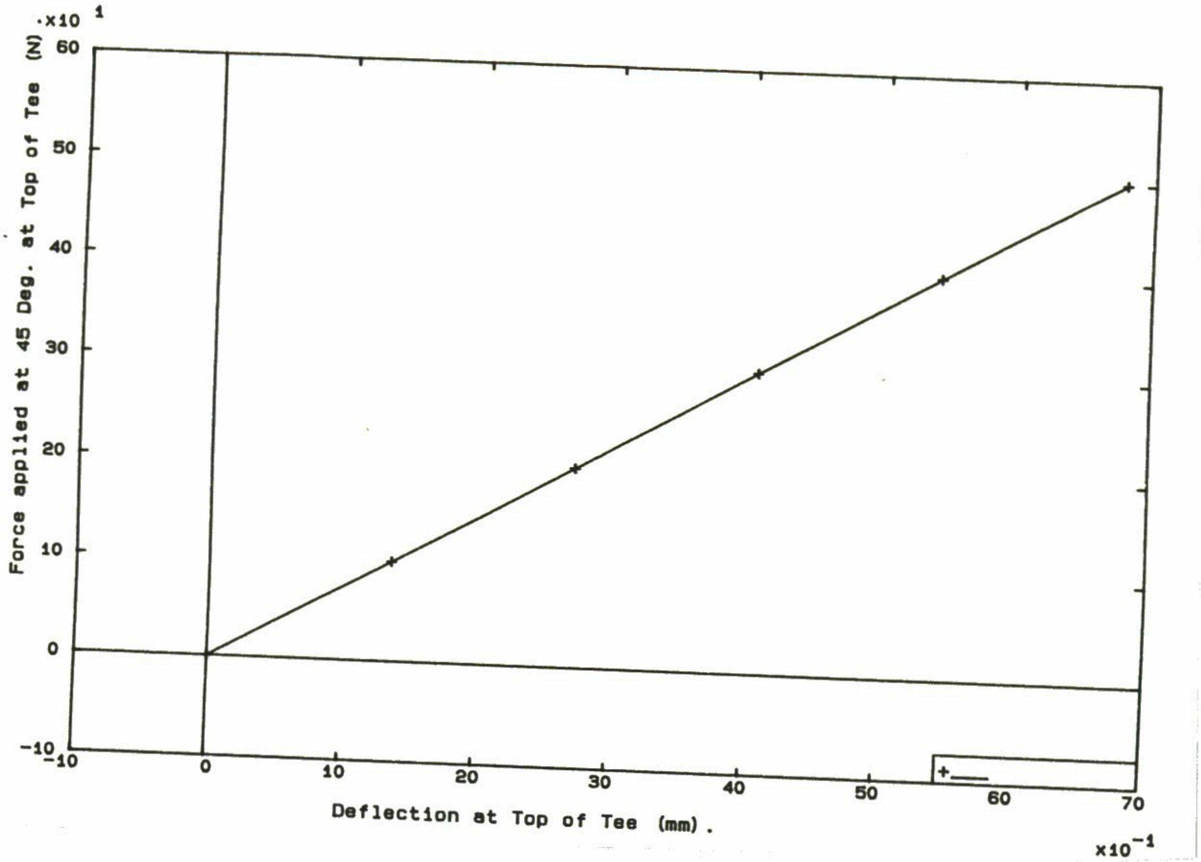


Figure 14: Load-Displacement Relationship

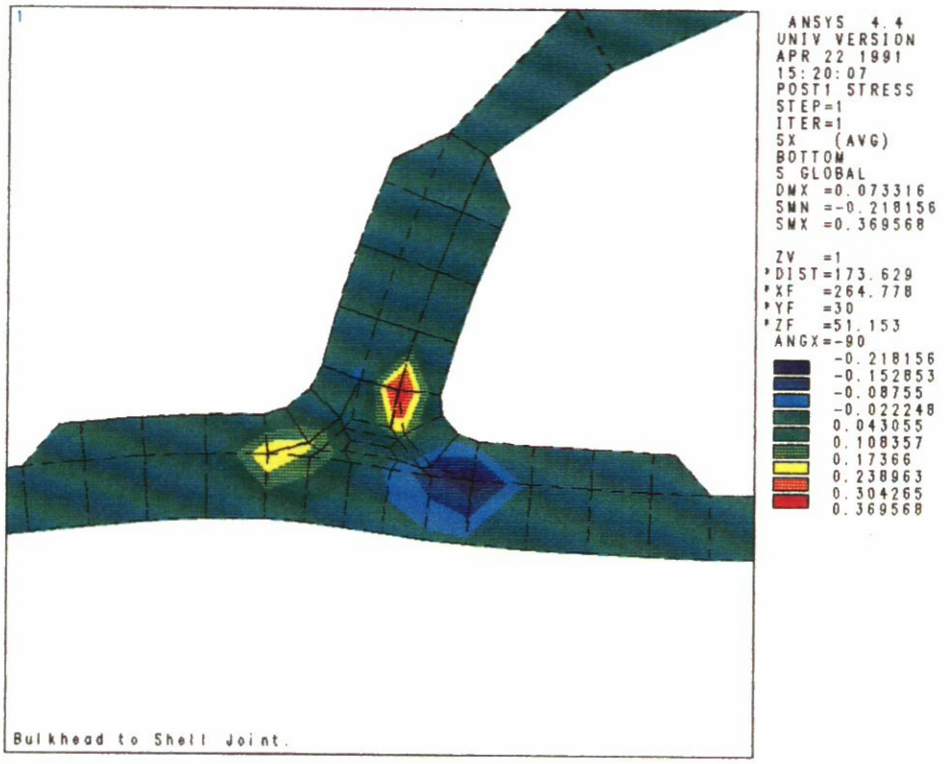


Figure 15: A Typical Stress Contour Plot