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DETAILED RESEARCH REVIEW OF THE PERFORMANCE CHARACTERISTICS OF OUT-OF-PLANE JOINTS IN FRP MARINE STRUCTURES

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by

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1 Introduction

One of the areas of primary interest in marine structures made from FRP (Fibre Reinforced Plastics) is joints. The reasonable design of a joint is very important with a view to improving structural efficiency. Joints become necessary in a structure for three main reasons:

(1) Production/processing restrictions.

Large and complex structures can not be formed in one process thereby needing several components to be joined together to produce the completed structure. Considerations that limit process size include exotherm, resin working time, cloth size and 'drapability', and mould accessibility and release limitations.

(2) Splitting the load path (and hence the fibre path) around the structure.

This typically involves the addition of stiffeners and bulkheads. Generally these out-of-plane elements cannot be formed at the same time as the rest of the structure and so need to be joined to it.

(3) Access and repair considerations.

If components within the structure require regular servicing the structural elements that obstruct access need to be joined to the remaining structure in such a way as to allow them to be removed with reasonable ease. If the hidden components require only very occasional treatment (such as removal after a major breakdown) then the structure can be cut out as necessary and treated as a repair. Here the joining method can be considered to be permanent.

When compared to welded joints on steel ship, out-of-plane joints in an FRP ship, either frame-to-shell connections or bulkhead-to-shell connections as shown in Figure 1, are characterised by two important features. Firstly, they are relatively labour intensive, less suited to automation and hence expensive to fabricate. Secondly, in case of top-hat stiffened single skin FRP ships, boundary angles account for about 10% of total structural weight, whereas 2.5% is typically allowed for all weld metal in steel ship construction. In a similar vein, top-hat stiffeners account for around 30% of a (top-hat stiffened single skin) ship's structural weight. Hence they provide a focus for potential weight and cost savings.

Furthermore, FRP structures are particularly susceptible to failure at bonded connections, especially at out-of-plane joints. The weakness in this context is caused by the absence of load bearing fibres across bonded interfaces, by the relatively thin adhesive layer forming the bond and by the inevitable occurrence of stress concentrations associated with joint geometry and bond imperfections. Lack of bond ductility and the absence of crack-arresting action by fibres can result in a rapid propagation of small regions of debonding by a 'peeling' action until catastrophic failure occurs. So designers have to pay a careful attention to the structural components.

Joints in both single skin and sandwich structures can be divided into two basic types, namely in-plane joints and out-of-plane joints. In-plane joints have been the subject of attention of many researchers [1]

[2], with attention being focussed on analytical treatment [3] [4] [5], numerical analysis [6] [7], experimental studies [8] [9], failure criteria [10] [11] and materials aspects [12]. So, in this paper, we will focus on out-of-plane joints.

As far as necessity and importance of out-of-plane joints are concerned, comprehensive research work should have been conducted. Unfortunately, apart from some testing and analysis in the early days, and some theoretical calculations in recent years, relatively little attention has been paid to the design of joint details. In order to accelerate the research on out-of-plane joints it is necessary to study past work in this field and thereby establish a state of the art. The objective of this paper is to review the research work on out-of-plane joints. Specifically, attention will be focussed on design synthesis considerations, structural static response analysis, experimental research of static behaviour, and creep and fatigue characteristics.

2 Types of Joints

Out-of-plane joints used typically in marine structures are shown in Figure 2 and include: frame to shell, bulkhead to shell, stiffener ending, stiffener intersection, deck edge (tee), deck edge (knee), and so on. Among these connections the most important and most often used in ship structures are of two types. One is the connection between two orthogonally placed plate panels, such as deck-to-bulkhead, shell-to-bulkhead and floor-to-tanktop. In this connection, termed a laminated tee-joint, both elements are formed before the joint is fabricated, and access is available from both sides of the joint. Another is the frame-to-shell·connection. The connection mainly employs top-hat stiffeners which are usually moulded in situ onto the cured shell. However, increasingly, use is also being made of extruded/pultruded section stiffeners which are preformed and bonded in place using adhesives and/or boundary angles. We call this connection top-hat stiffener joint. Figures 1 a and b represent typical configurations for commercial and naval applications, respectivelly, and Figure 1 c are new and alternative configurations developed by Dodkins, Shenoi and Hawkins [13] and Shenoi and Hawkins [14].

A good understanding of the performance of the laminated tee-joint and top-hat stiffener joint and the methods used for investigating both the joints leads to an ability to analyze and design many other joints. For example, an angle joint is also sometimes encountered in marine structures. It too consists of two orthogonally placed plate panels like the laminated tee-joint. Using the same methods as those for laminated tee-joint and top-hat stiffener joint, it is not difficult to solve the design analysis problems related to angle joints. Similarly, with regard to a half-round stiffener joint, the design can be conducted in a way similar to that used for the top-hat stiffener joint. Therefore the review is generally limited to laminated tee-joint and top-hat stiffener joint.

3 Design Rules for Joints

The first boat made in glass reinforced plastic (GRP) was built in America in 1944. This fifty years design, production and service experience has resulted in the publication of a number of design codes and rules for the fabrication of marine structures made from GRP/FRP.

3.1 Marine Design Manual for GRP [15]

One of the earliest approaches to GRP structure design is outlined in the Gibbs and Cox manual [15]. This manual gives recommended arrangements of various joints and simple design examples. For top-hat stiffener joint, design graphs of section modulus and moment of inertia are given in order to expedite the work of designer. These stiffener configurations give the designer the opportunity to vary the stiffness and strength of the section simply by changing the cross-sectional dimensions instead of changing the thickness of the fibreglass laminate. For maximum strength and rigidity the reinforcement preferred for the stiffener laminate is woven roving, with the thickness approximately equal to that of the shell laminate. The dimensions of laminated tee-joint shown in Figure 1, it is stated, 'should be minimum consistent with strength requirements; 2″ minimum recommended'. In secondary bonded joints, i.e. joints using an adhesive to connect cured laminates, subjected to shear tending to slide one laminate past the other, it is recommended that a layer of resin impregnated fibreglass mat reinforcement be placed between the joining surfaces since it acts as the adhesive carrier and as a reinforcement. The manual explained that fibreglass mat is preferred since it retains the adhesive and its random distribution of glass fibres provides reinforcement in the joint opposing the shear force.

3.2 Naval Engineering Standards [16]

It is well known that the first GRP minehunter in the world was built in U.K. in 1973. So early work in U.K. centred around the design of GRP minehunters [17] [18] [19]. This covered aspects of structure design from a number of viewpoints such as strength, stiffness, stability and material failure modes. 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 conducted in U.K. formed the impetus to the drawing-up of naval engineering standards [16]. 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 covered the methods of joining different structural members and design of structural intersections including tolerances. With respect to laminated tee-joint, it is stated that, the thickness of both boundary angles shall be $k(t-n)$ mm of CSM (chopped strand mat) + n plies of woven roving, where t is the thickness of mm of the leg of the tee, $n = 4$ for machinery spaces and 2 elsewhere as a fire barrier and $k = \frac{1}{2}$ minimum and desirably $\frac{2}{3}$. The extent of the boundary angle along both the leg and top of the tee should be 100 mm minimum and 150 mm preferably and always in tanks, irrespective of thickness of structure, although when joining minor structure 75 mm boundary angles may be permitted. Some suggested values and tolerances for gap sizes are also given.

3.3 Lloyd's Register of Shipping Rules [20]

With regard to merchant ships, yachts and small craft, design guidance is sought primarily from classification society rules. Lloyd's rules [20] state that, for top-hat stiffeners, the 'width of the flange connection to the plate laminate is to be 25 $mm + 12$ mm per 600 g/m^2 of reinforcement in the stiffener webs, or 50 mm whichever is the greater'. For connection of floors, bulkheads, etc., to adjacent structure in a tee-form, the width of each flange needs to be '50 mm + 25 mm per 600 g/m^2 of reinforcement'. Furthermore, the weight of the laminate forming each angle is required to be at least 50% of the weight of the lighter member being connected.

3.4 American Bureau of Shipping Rules[21]

American Bureau of Shipping rules (1978) [21] are applicable to GRP vessels under $61m$ in length. For top-hat stiffener joint, proportions of the stiffener are derived in terms of thicknesses of stiffener crown and webs, with these being dependent on section modulus requirements. For a laminated tee-joint, the thickness of each boundary angle is to be not less than obtained from the following:

a Single-skin to single-skin

One-half the thickness of the thinner of the two laminates being joined.

b Sandwich to sandwich

The thickness of one skin of the thinner of the sandwich panel.

c Sandwich to single skin

Either one-half the thickness of the single-skin laminate or the thickness of one skin of sandwich panel, whichever is less.

The width of each flange including end taper is to be not less than 15 times the thickness obtained above. American Bureau of Shipping rules specify that in secondary bonds the final ply of laminate along the bond line of the cured laminate is preferably to be chopped-strand mat. The first ply of the secondary layup is to be chopped-strand mat.

3.5 Det Norske Veritas Rules [22] [23]

Det Norske Veritas (1985) [22] 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.

Det Norske Veritas (1991) [23] rules specify first principles-based calculation procedures to determine section moduli of top-hat sections and scantlings of plate laminates.

3.6 American Ship Structures Committee report [24]

Ship Structures Committee report (1990) [24] uses diagrams from Reference [15] and states similarly that dimensions of boundary angles should be minimum consistent with strength requirements. It refers to classification society rules [21] for top-hat stiffener scantlings. It also outlines a simple procedure based on beam theory to determine section modulus and stiffness values for stiffeners.

Apart from the design rules mentioned above other several classification societies have still published rules for the design, construction and testing of FRP ships [25] [26] [27] [28]. Recently, many of these Societies, as a consequence of their years of experience, have revised and updated these rules or issued new rules for unusual and high speed craft [29].

It is noteworthy that no specific procedures concerning joint design are elaborated in the design rules above. Another common characteristic of all the rules issued by the different classification societies is that of maintaining a high level of safety in the scantlings of the structures through the application of high safety factors. The principal thrust of the rules seems to be the provision of high stiffness in the region of the joint. Implicitly the rules are geared to result in adequate in-plane strength of boundary angle laminates. Significantly, the weak area, i.e. the out-of-plane properties is not addressed in an explicit manner.

4 Modelling of Static Structural Response–Finite Element Analysis

Efficient design of structural connections requires a clear understanding of the distribution and magnitude of the stresses at the connections. Early work on the stress analysis associated with out-of-plane joints was directed towards solving a particular design problem and not to developing a design method. So, specific and detailed analytical studies in context of out-of-plane joints are surprisingly rare. It is clear that classical analytical methods cannot accommodate such complex geometries. Therefore, it is necessary to utilize numerical techniques such as the finite element method.

4.1 Single Skin Tee–Joints

Smith [19] [30] carried out stress analysis on a laminated tee-joint i.e. a bulkhead to shell joint incorporating a single boundary angle by means of a plane strain approach. The joint was subjected to shell 'pull-off' and flooding load. The computed distributions of direct and shear stress across bonded interfaces are shown in Figure 3. Although he derived a detailed stress patterns on the tee-joint, the elements used may not have permited a full coverage of all composite material-related properties.

Experimental evidence indicates that the incorporation of an insert increases out-of-plane load transfer capability of an integral composite joint [31] [32]. Gillespie and Byron-Pieps [33] conducted a detailed analysis of a spar-to-wingspan joint in an aircraft using structural analysis program 5 (SAP5). They utilized two-dimensional plane stress element with temperature-dependent and orthotropic material properties and did linear-elastic stress analysis. Analytical results consist of design curves where the interlaminar normal stress σ_3, interlaminar shear stress σ_{23} and the inplane bending stress component σ_2

are presented as a function of various geometric parameters. This enables designers to employ any failure criterion, for example, maximum stress failure criterion, Tsai-Wu failure criterion, etc., and material strength estimations for prediction of integral joint strength.

As described above, traditional analytical methods are currently incapable of modelling joint behaviour due primarily to complexities in geometry and material make-up. Shenoi and Hawkins [14] investigated comprehensively the impact of systematically-generated design variations, as shown in Figure 4, on the behaviour of laminated tee-joint in FRP ships using the ANSYS finite element analysis package. The package has a composite capability in that it provides options from three layered elements comprising two eight-nodes shell and one eight-noded solid. The geometry of the boundary angle problem precluded the use of shell elements as some stacking was required. The solid element assumes a linear variation of stresses through the element thickness of a laminate so several elements were required through the thickness of a laminate to illustrate the stress distributions within it. The flexible resin fillet was represented using an isoparametric three-dimensional solid element with large deflection and plasticity capabilities. Figure 5 shows a typical finite element idealization. The model is made up of two elements across the width of the joint with one element through the thickness of the base plate and two elements through the thickness of the boundary angles.

The model were subjected to a displacement controlled pull-off of the web at 45^0 to the base (flange) plate and restrained on the flanges at the edges of the overlaminates as shown in Figure 6. This simulates the loading applied to a boundary angle within a tank structure subjected to both vertical tensile loading and horizontal bending due to hydrostatic pressure.

In order to have a fuller range of geometry and material variants, three designs, viz. B, C and F, out of the eleven joint configurations shown in Table 1 were numerically analyzed by Shenoi and Hawkins using four combinations of linear and non-linear material properties and small and large deflection analysises. Figure 7, 8 and 9 give stress contours within laminated tee-joint for three sample sets–B, C and F. Sample B is a 'traditional' fillet with a thick overlaminate, sample C is a pure fillet and sample F is a new design with a very thin overlaminate, which represent the possible types of joint in a production context. To model the behaviour of the joint after initial delamination, gap elements, with suitable properties, were inserted between the inner and outer laminates around the radius on the tension side of the joint. Figure 7 shows the stress contour plots with gap elements included. The results obtained from their numerical modelling can be summarized in terms of three main conclusions:

- The behaviour of laminated tee-joint is very dependent on geometry and material make-up. The numerical studies have been shown that increasing the thickness of overlamination, traditionally the criterion used for design, has a detrimental effect on joint performance, and that the radius of the fillet, traditionally given little or no consideration by current design methods, is critical to the performance of the joint. In general, increasing fillet radius enables the joint to withstand high loads. The gap between the panels and the edge preparation of the tee piece also influence joint behaviour but to a lesser degree. The choice of overlaminating resin has been shown to be an important material-based vaviable. The impact of these geometry and material variations on stresses is summarized in Table 2.

- Failure modes too are dependent on material and geometry configurations within the joint. The prediction of failure values using finite element modelling has been of limited value because of the inadequate and incomplete nature of basis material data, especially those pertaining to resin strength and modulus and laminate interlaminar tensile strength values. So it is necessary to do accurate measurement about the basis material data.

- With regard to the efficiency of laminated tee joint, which is defined as the ability of a design configuration to withstand as large a load and as high a deflection with as low internal stresses as possible, it is evident from Figure 10 and 11, which are plots of peak joint stresses versus load and deflection for the different joint configurations, that the most efficient joint is sample F using a large radius flexible resin fillet with an overlaminate of minimal thickness, just sufficient to withstand the membrane tensile loads.

4.2 Sandwich Tee–Joints

For a laminated tee-joint in the sandwich ship structure, Violette [34] derived a suite of computer algorithms to predict failure loads of the joint on the basis of simple beam theory. In the theoretical analysis, the hull bulkhead assembly has been modelled as a beam with variable cross section and engineering properties. Further using finite element techniques, Shenoi and Violette [35] examined the influence of joint geometry in the sandwich ship structure, as shown in Figure 12, on the ability to transfer out-of-plane loads. Figure 13 gives the finite element deflection plots. Figure 14 shows a comparison of the slope along the span in the F.E. model and beam theory model. It can be concluded that the beam theory-based analytical model is sufficiently accurate for preliminary design configurations. Shenoi and Violette presented the following performance indices:

1. weight efficiency (weight to strength ratio, weight to deflection ratio)

2. production efficiency

3. material cost implication

which can be used to determine the best overall joint configuration. When the tee-joint is limited to the transfer of compressive load, the filler fillet of radius 10 mm has achieved the best overall performance followed by the foam pad joint according to the performance indices above. As a rule of thumb, optimum filler fillet radius should be chosen as approximately the thickness of the sandwich panel, to correlate with empirical formulae used in wood epoxy construction.

4.3 Single Skin Top-Hat Stiffener Joints

As in the case of laminated tee-joints, a key characteristic of top-hat stiffener joints is that, because of a lack of continuity of reinforcing fibres across the joint, it is susceptible to fail by peel or delamination well before the ultimate in-plane material stress is reached. Furthermore, its dependence on interlaminar properties makes the joint somewhat sensitive to material imperfections such as voids and to minute changes in geometry in the laminate.

Smith [30] presented some typical arrangement of top-hat stiffener joint illustrated in Figure 15. It is well known that the main purpose of this connection is to transmit shear stresses between the shell and frame flanges under local bending caused by lateral pressure or concentrated lateral loads. Design of the connection requires evaluation of an envelop of maximum shear forces in each frame from the shear force distributions obtained by the finite element analysis of a hull compartment. From the shear force S acting on a frame cross-section the average shear stress τ across the bonded interface can be obtained by beam theory

$$\tau = \frac{S}{b_j D_V} \int_{z_j'}^{z_0'} bz' E dz \qquad (1)$$

in which D_V is the flexural rigidity of the section, b is the total width of material resisting vertical shear at a distance z' from the netural axis, E is the local Young's modulus and b_j is the value of b at position j; integration is taken from z_j' to the outer fibre of the section at z_0'.

In order to identify key variables that control and govern the transfer of load from the panel to the stiffener and vice-versa, Shenoi and Hawkins [36] and Dodkins, Shenoi and Hawkins [13] carried out the study of the problem of top-hat stiffener joint shown in Figure 1. The variations considered by them,as shown in Figure 16, are:

- radius of fillet ($25 - 125$ mm);
- thickness of overlaminate ($1 - 12$ $laminates$);
- gap between base panel and stiffener ($10 - 50$ mm);
- the fillet backfill angle inside the stiffener ($0 - 45^0$).

The boundary conditions applied to the models are shown in Figure 17. Centre-clamp loading is the most severe in terms of minimum load at failure, as this mode places direct tensile loading on to the fillet. However, this form of loading does not result in the initial delamination seen in practice, so two-clamp loading has also been considered.

All the modelling was done using the ANSYS suite of programs. The F.E. analysis was conducted taking into account possible non-linearities in the material properties as well as those due to structural geometry. The results of the study are summarised in Tables 3 and 4. Four interesting features that arise are outlined below:

- For both loading modes, the through-thickness stress in the overlaminate shows a minimum value with a thickness of two laminations, rapidly increases as laminations are added, then remains reasonably constant at thicknesses above five laminations.

- For two-clamp loading, the stress in the fillet increases slightly as overlaminations are increased up to four laminations, and then falls rapidly away as more laminations are added and the joint becomes stiffer. This reduction in stress does not occur in single-clamp loading because the joint stiffness is little affected by an increase in overlaminations.
- For both loading methods, the stress in the fillet shows a minimum at a gap of $30mm$, although for single-clamp loading the stress value drops again as the gap increases above $40mm$.
- For single-clamp loading, the stress in the fillet increases as the backfill angle reduces but then falls rapidly as the angle approaches 0^0. For two-clamp loading, the stress in the fillet remains constant, and then rises slightly as the backfill angle increases to 45^0. This is matched by an increase in stiffness at this angle.

It is very important to point out that the main thrust of the current practice seems to be one of aiming to make the joint, both laminated tee-joint and top-hat stiffener joint, as stiff as possible. Furthermore, there is a mistaken belief that increased stiffness also corresponds to increased strength of the joint. The results of the numerical and experimental studies (see section 5) conducted by Shenoi, Hawkins and Dodkins [13] [14] [36] corrected this false premise.

4.4 Single Skin Top-Hat Stiffened Panels

It is well known that the design and fabrication of large GRP structures are frequently limited by the low specific stiffness of the material. The specific stiffness of GRP is one-third to one-quarter that of metals, and so this necessitates the use of stiffened construction. That is why top-hat stiffeners are in general use, but are particularly widespread use in the GRP shipbuilding industry.

Design of the panels reinforced by top-hat stiffeners must clearly include careful consideration of elastic instability. Smith [30] comprehensively examined compressive buckling of longitudinally stiffened panels and compressive buckling of transversely stiffened panels using approximate analysis, folded plate, finite strip and finite element analysis.

a Compressive buckling of longitudinally stiffened panels

There are two distinct forms of compressive failure which can occur in GRP panels for the compressive buckling of longitudinally stiffened panels. In the first case, collapse was precipitated by debonding of stiffeners caused by large local buckling deformations of the shell laminate. In this case, designers should pay special attention to stiffener attachment, and inspections carried out during the life of the structure should include a search for incipient debonding. In the second case collapse was caused by compressive material failure in the tables of stiffeners resulting from overall column-like buckling of stiffeners,virtually unaffected by local buckling.

b Compressive buckling of transversely stiffened panels

Folded-plate calculations carried out for a wide range of transversely framed top-hat panels have indicated that the lowest buckling stress usually corresponds to a local, interframe mode having one of the three forms shown in Figure 18. Data curves suitable for intial design purpose have been developed for each of the forms of buckling indicated in Figure 18. For practical design purposes Smith [30] suggested that preliminary estimates of initial buckling stress should be made using the data curves; at a later stage in design, these should be supplemented by using folded-plate analysis. Where folded-plate analysis is not available, linear finite element analysis, although less accurate and much more expensive, may be used instead to estimate initial buckling stresses. Nonlinear finite element analysis performed by the general purpose code ASAS-NL has confirmed that for FRP panels with typical, small imperfections, initial buckling stresses obtained by linear analysis provide a close approximation to collapse strength.

4.5 Pin-Jointed hybrid GRP/Steel Panels

An effective way of avoiding unwanted hull-superstructure interaction is use of a low-modulus material in the superstructure on a steel ship. GRP is an obvious candidate, offering tensile and compressive strengths of the same order as the yield strength of mild steel, together with a Young's modulus which is less than 10% that of steel.

Smith and Chalmers [37] made a thorough study of the use of GRP in ship superstructure . They compared some alternative composite superstructure designs with standard steel and aluminium designs showm in Fig. 19 according to overall and local maximum stresses, deflections, and relative weighs and relative costs. Deflfctions w, stresses and strains were estimated using composite beam theory with appropriate reductions in assumed effective breadth for slender metallic and GRP panels. In the case of Designs C, D, E1, E2 and F (see Fig. 19) folded-plate analysis was employed to account for the unequal span 'continuous beam' behaviour of the plating and the effects of coupled bending and twisting of asymmetric stiffeners. Shear stresses in stiffener wrbs were estimated using Eqn.1. Table 5 gave a comparison of the alternative stiffener panel designs.

Design of joints, both between components of a superstructure and between the superstructure and the main hull, is a critical part of the overall design requirement. The joints fall into the two categories: stiffener to panel joints and panel to panel joints.

As will be discussed below, stiffener to panel joints in all GRP or hybrid GRP/steel panels may be achieved either by adhesive bonding, by mechanical fastening (bolts or rivets) or by some combination of both.

Panel to panel joints must be able to transmit bending moments and direct and shear forces as shown in Fig. 20. These joints can be divided two types. One is butt connection and another is corner joint, illustrated in Fig. 21. According to fabrication process, for corner joint there are two possibilities: 'pin-jointed' connection illustrated in Fig. 22, which was unable to transmit bending moments across their boundaries, and rigid jointing of stiffeners to panel boundaries, which aimed at achieving full transmission of bending moments.

Smith and Chalmers [37] and Murphy and Smith [38] examined collapse behaviour of the hybrid GRP/steel panel under lateral load using the general purpose FE code ASAS-H. The panel is framed by bonding and bolting steel stiffeners to flat GRP sheets. Attention was focussed on the complex behaviour of the 'pin-jointed' connection.

Fig. 23 shows the finite element model of the panel investigated. Linear deformation of the panel computed under uniform lateral pressure is shown in Fig. 24. Deformation is evidently concentrated mainly round the stiffener ends where high bending and interlaminar shear stresses are found to occur in the GRP laminate; bending of the steel stiffener is slight.

Yielding in the stiffeners and large-displacement, membrane effects in the GRP laminate, found experimentally to be important at higher loads, are not however accounted for by linear analysis. In order to represent these effects approximately, nonlinear analysis was carried out using the special-purpose computer code FABSTRAN [39]. A comparison of computed and experimental load-displacement curves is shown in Fig. 25.

The study of the use of GRP in ship superstructures [37] [38] has confirmed stress reductions associated with hull-superstructure interaction and led to the conclusion that this form of construction would save up to 50% of the weight of a conventional steel superstructure and would eliminate the problem of fatigue cracking caused by hull-superstructure interaction.

5 Modelling of Static Structural Response–Experimentation

While numerical analysis of stresses at out-of-plane joints will provide insights into joint performance and may be used as a means of improving joint geometry, purely theoretical estimates of joint strength are unacceptable as a basis for design because of uncertainty about imperfections, local stress concentrations and material failure under multi-axial stresses within a connection. Reference must therefore be made to test data, and development of new high-performance designs should include a thorough programme of tests on all important joints for evaluation of static, fatigue, creep and impact strengths.

It is worth mentioning that considerable experimental research work on how to improve attachment methods for stiffening frame on large GRP ships were carried out by many researchers [17] [40] [41] [42] [43] [44] [45] [46].

5.1 Single Skin Tee-Joints

Apart from performing finite element analysis, Shenoi and Hawkins[14] and Hawkins, Holness, Dodkins and Shenoi [47] also conducted experimental work to demonstrate the validity of their finite element analytical results. Eleven joint configurations were considered to study the effect of varying geometry and resins. The variables included in their researches are as listed below:

- fillet radius;
- no. of plies in boundary angle;
- material make-up of boundary angle plies;
- edge gap between web and flange of tee;
- shape of edge of wedge.

The details of the specimens are outlined in Table 1. All the configurations were formed around standard plate members comprising 15 layers of woven rovings laminated with polyester resin.

Four samples of each configuration were subjected to the same load and restraints as F.E. model (see Figure 6). The load-deflection curves for the samples are shown in Figure 26. Figure 27 gives failure configurations in various samples.

Sample B (current practice) failed initially by delamination within the boundary angle corner on the tension side (see Fig. 27 a). This delamination occurred within the third ply of the fillet. The load carried by the joint dropped away almost to zero, as can be seen from Figure 26. However, by reapplying further displacement, it was found that the joint continued to carry load, with little reduction in stiffness, up to and above the level at which the initial delamination occurred. Final failure occurred when the remaining boundary angle delaminated and the fillet failed.

Sample C failed catastrophically in the fillet. It was reported that the point of initiation of failure was impossible to judge due to the speed at which it occurred. Figure 27 b shows the position of the cracks in the fillet.

Sample F (a new design) failed catastrophically by complete delamination of the overlay on the face in tension and splitting of the fillet (see Fig.27 c). These two mechanisms occurred too rapidly to distinguish their order. Some acoustic emissions could be detected at lower loads but no delamination was visible. These samples showed an almost constant stiffness up to failure and withstood the highest loads of all samples tested.

The results have confirmed the F.E. analytical results:

- increasing the thickness of the overlamination has a detrimental effect on the properties of the joint;
- increasing the fillet radius enables the joint to withstand higher loads.

The following can be deduced from Shenoi and Hawkins' experimental results:

- increasing the gap between the base plate and the tee piece (comparing sample D with C) allowed much greater deflection for a given load (100%) and a greater load potential (50%). Hence the joint was able to absorb more energy prior to failure.

- Bevelling the edge of the tee piece (comparing sample E with C) reduced the joint stiffness by less than 10% but increased the failure load by over 50%.
- Overlaminating (comparing sample F with C) increased stiffness by 50% and failure load by 100%.
- Increasing the radius of the fillet (comparing sample G with C) made very little difference to the stiffness but increased the failure load by 150%.

Correlation between laboratory tests and 'real' structures can be achieved by water pressure testing of GRP tanks. Vosper Thornycroft Test House [48] [49] investigated the relative effect of material and geometry changes using sixty two boundary angle specimens in this context.

For all specimens, the base plate and the vertical plate were built up of 15 plies of woven rovings, resin/glass ratio 50 : 50 by weight. All the specimens were 100 *mm* wide. For the first series of test specimens, the length of the base plate was 800 *mm* and the vertical plate was 400 *mm*. All the other specimens had base plate length of 560 *mm* and vertical plate length of 260 *mm*.

The test results show that increasing the internal radius of the boundary angle had a marked effect in increasing the strength of the specimens. Using a resin mixture of 50% crestomer 1080 and 50% polyester resin also give an improvement in strength. On the basis of experimental results, specimen B had better performence than the others. Specimen B is internal radius of 35 *mm* and fillet radius of 55 *mm* which is made of crestomer. The large scale tests confirmed the results obtained on scale models in laboratory experiments [14] [47].

Cui, Liu and Liu [50] performed experimental research not only on tee-joint but also on angle joint and compared their experimental results with theoretical results obtained by SAP5. The conclusions from their work are similar to the observations made by Hawkins et al.[47].

An important aspect defining failure mechanism, apart from material and geometry considerations, is the load direction and the boundary conditions. The failure pattern shown in Figure 27, for example, pertains to the boundary angle being clamped on the flange piece of the tee as shown in Figure 6. However, it has been observed [51] that alternative forms of load direction and boundary conditions, such as a 3-point bend test, lead to a different failure mechanism, see Figure 28. It will be noticed that the initial failure mechanism is a cract in the fillet which causes delamination along the web-overlaminate boundary.

5.2 Sandwich Tee-Joints

Pattee and Reichard [52] conducted detailed studies on bulkhead-to-hull joint configurations in sandwich construction. Twelve different designs for bulkhead-to-hull joints were evaluated using static laboratory testing; six configurations for each of the two core types, balsa and foam. The joint geometries include: tapering the core to solid fibreglass in the hull panel under the bulkhead joint , continuous core in the hull under the joint , continuous core in the hull with radius and wedge fillets at the joint, and an underlying pad of extra core material at the joint with the inner hull reinforcement running under the pad and one with the inner hull reinforcement running over the pad. The configurations were subjected to both a compressive and tensile load in the plane of the bulkhead, and a bending load perpendicular to the bulkhead. The tensile configuration was similar to that shown in Figure 28, except that the load was applied on the web piece rather than on the base/glange plate. The compressive configuration was the inverse of this and the bending mode was achieved by a manner similar to that shown in Figure 6.

The tests carried out by Pattee and Reichard showed that continuing the core under the bulkhead yields higher strength and stiffness compared to removing the core. In general, the compression and tension tests are the primary measure of overall joint performance, and the bending test is secondary. For the primary tests, the filleted designs showed higher static strength and stiffness benefits compared with no fillets. The padded design with the inside hull skin over the underlying pad showed even higher strength and stiffness benefits for the tensile loading, but no advantage in the other tests. The failure modes for the various designs, core materials and tests were presented graphically in their paper [52]. For compression and bending tests the failure occurred mainly due to core failure, and for tensile test the predominant failure mode is the bottom hull skin buckling. Figure 29 indicates typical failure modes for PVC foam fillet and foam pad joints. In the compressive test, the filler fillet joint failed by the boundary angle disbonding from the fillet material which itself showed cracks while the triangular foam pad joint failed through shearing of the core under the web piece. In the tensile test, both types failed through the outer skin on the flange piece disbonding from the core. In the bending test, the filler fillet joint failed through core shear in both the web and flange pieces, whereas in the foam pad joint only the core in the web piece sheared.

Shenoi and Violette [35] investigated the behaviour on the laminated tee-joint in the sandwich ship structure using three typical joint geometries (five different joint configurations illustrated in Figure 12) which are found in boatbuilding. All experiments have been performed on a JJ instruments M30K testing machine, with the compressive load being applied on the web of the tee (i.e. the hulkhead) and with the two extremities of the flange (i.e. the hull panel) being simply supported at the ends. Failure loads (including theoretical failure loads [34]), maximum deflections and slopes of the graphs obtained by Shenoi and Violette's test are summarised in Table 6. Table 6 shows relatively close agreement of experimental results with analytical predictions (the basis of which was discussed earlier in Section 4).

5.3 Top-Hat Stiffener Joints

When using top-hat stiffeners, it is vitally important that a good bond is achieved in the top-hat stiffener joint. The conventional fabrication method involves the lamination of the hull shell and flat or gently curved deck and bulkhead panels. Rigid foam cores are bonded to these unstiffened structures where stiffness is required and GRP laminations are built up around the cores. When constructing a large hull, it is not uncommon for a substantial delay to occur between shell lamination and the addition of the stiffeners. Early exploratory work [17] established that a delay of greater than 7 days prior to stiffener lamination led to an excessively weak secondary bond between hull shell and the flanges of the stiffener, if no special precautions were taken.

Surface treatments for the hull immediately prior to stiffener lamination were evolved, these involving abrasion, wiping with solvents and the use of peel plies. These precautions enable stiffeners to be fabricated that perform satisfactorily in most circumstances, with a secondary bond whose transverse tensile strength equals the interlaminar tensile strength of the main hull laminate [40].

In the case of naval minesweepers and minehunters, the hull will experience not only static loading but also dynamic loads from hull motion, slamming and possible explosive loads. Under explosive loading, a top-hat stiffener joint may be exposed to substantial through-thickness tensile stresses. These are caused both by reflection of a transmitted shock wave through the laminate and by differential inertia forces associated with overall dynamic response of stiffened panels to impact loads. Even though the surface preparation of the hull prior to stiffener lamination yields a secondary bond whose transverse tensile strength equals the interlaminar tensile strength of the woven glass roving/polyester resin laminate of the hull, there could be a tendency for the stiffeners to debond from the hull during explosive loading at or just below the secondary bond line.

Dixon and Ramsey [17], Smith and Pattison [41] and Harris [42] overcame the tendency for stiffener and hull to part under critical shock loading conditions by bolting stiffener flange to the hull shell with non magnetic titanium bolts. Figure 30 gives the load versus displacement curves for slow pull-off test on traditional constructions with brittle A2785CV polyester resin throughout, one design was bolted whilst the other was not. The curves show that at modest load around 14 kN a crack was initiated in one of the two fillets. For the unbolted construction this crack rapidly propagated causing the flange to separate from the hull (base panel). For the bolted version, both fillets cracked, but crack propagation was restrained by the bolts and separation did not occur. This illustrates clearly the reason for including bolts which is not to inhibit crack formation but solely to restrain crack propagation and prevent catastrophic failure. The titanium bolts simply acted as crack arresters. In addition to this, the non magnetic titanium bolts were expensive and labour intensive to fit.

Considerable work has been carried out to try and overcome this problem by the use of mechanical fasteners [40] [43]. The stiffness of the stiffener web and flange corner played an important part in determing the stress distribution across the secondary bondline and defining the failure sequence. The initiation of failure occurs at an acute stress concentration, where the traditional material present as the fillet, ie. polyester resin and milled glass fibres, is acutely brittle in nature. It is an inappropriate material for providing any form of stress relief. The configuration requires a fillet of high compliance which will confer on the structure the ability to deform elastically by a substantial amount. Such deformation allows significant amount of elastic energy to be stored by the structure.

Bird and Bashford [44] carried out experimental research on how to use flexible resins, which have elongation values of up to 120%, in the top-hat stiffener joint to achieve acceptable properties under static and dynamic loads. Table 7 gives the basic mechanical properties of their selected flexible resin and A2785CV polyester resin. Crestomer 1080 is a thixotropic resin suitable for wet lamination in minesweeper construction [44]. For the top-hat stiffener joint it was necessary to produce a fillet of high strain to failure if the goal of delaying crack formation was to be reached. A solution was found by adding colloidal silica to the flexible resin. This had the effect of thickening the resin to a trowellable paste without over-reinforcing the cured resin. As a consequence, a significant proportion of the high strain to failure was therefore retained.

In order to increase the flexibility of the joint, compliant resins were used in the woven roving plies either side of the secondary bondline, between stiffener and hull, as a means of increasing the volume of compliant material. Two main designs containing flexible resins studied here [44] are as follows:

- Design A – standard construction [16], as shown in Figure 1 b, but with high strain to failure resin fillets;
- Design B – As for Design A, but seven additional woven roving plies containing flexible resin were added. Three were placed on the base panel beneath the stiffener whilst four others were laid over the fillet and around the flange heel corner as shown in Figure 31.

The results of slow pull-off tests for Design A and B are given in Figure 32, in comparison with a through-bolted construction. It is clear from Figure 32 that Design A achieved good performance in reaching a failure load of around 40 kN without any crack formation up to that load. The load was equivalent to the maximum achieved by the bolted version. For Design B much higher loads were achieved at around 70 kN coupled with a very large amount of elastic deformation (displacement). Again cracks did not form until maximum load. From this evidence it was concluded that configurations of the Design B type were highly promising. Also, the sole use of flexible resin fillets (Design A) had merits based on ease and simplicity of construction.

The high strain rate performance of GRP out-of-plane joint is very difficult to test. Bird and Bashford [44] laid up two types of panels to investigate high strain rate performance. The panels were subjected to explosive loading of varying severity. The high strain rate tests indicated that increasing the compliance of the joint area greatly reduced the risk of delamination. It is claimed that the use of flexibilised laminates gives the greatest protection against delamination whatever the severity of test.

Once again it is interesting to point out the differences in failure patterns for alternative load directions and boundary conditions. Elliott [51] investigated three options as shown in Figure 33. The corresponding failure modes are illustrated in Figure 34. For the 3-point bend test, there is an initial disbond of the fillet to overlaminate interface followed by through skin thickness delamination of the overlaminate. This latter is coupled with a failure of the base plate inside (or tensile) surface. For the reverse 3-point bend test, failure consists of through fillet cracking and a fracture of the fibre in the outer plies of the base plate on the tension surface. In case of the pull-off load test, the failure is very simple, with the overlaminates being peeled-off base plate.

6 Creep Characteristics

Creep can be defined either as the increase of strain which a material undergoes over a period of time under constant stress or as relaxation of stress under conditions of constant strain. Differing from the conventional ship structural metals, polyester based GRP laminates like many other 'plastics' exhibit pronounced creep characteristics at ambient temperature. Crabbe and Kerr [53] carried out the short-term creep tests of out-of-plane joints.

The specimens used for this investigation were the MCMV type out-of-plane joints made $\frac{2}{3}$-scale. There were five defferent tests performed by Crabbe and Kerr:

- frame-shell test;
- frame-keel test;
- deck beam-frame loaded parallel to deck test;
- deck beam-frame loaded normal to deck test;
- deck beam-frame test (knee test).

The specimens were subjected to a shear strain combined with varying degrees of bending strain. The loads were applied incrementally and held at the maximum values for 15 minutes after which they were reduced incrementally to zero and 30 minutes allowed for recovery of apparent permanent strain before reloading. From load-displacement measurement during loading and unloading creep-time and recovery-time diagrams have been constructed. Representative force-deflection and creep diagrams are given in Figure 35 [53].

It was observed from the research that:

- the data obtained indicated that in most cases after 15 minutes under load creep is still significant;
- the deformation suffered by the load for up to 15 minutes duration can almost be recovered within 30 minutes of unloading if the maximum applied load did not exceed 50% of failure load;
- the load which exceeded 50% of failure load generally resulted in some permanent deflection;
- recovery of apparent permanent strain on removal of the load was always rapid over the first 10 minutes but, although continuing for several hours, this was at an ever reducing rate;

Perhaps, more important are creep characteristics of out-of-plane joints at elevated temperatures or immersion in water. These aspects are notably not covered in the existing works.

7 Fatigue Characteristics

Design against through-thickness fatigue cracking of the laminate in FRP hulls may be based on reference to S-N curves for representative material specimens. Service experience on GRP ships indicates that fatigue damage, comprising resin cracking and fibre debonding at stress-raisers such as holes and hatch corners, usually remains very localized with negligible effect on overall structural behaviour. Fatigue is most likely to pose serious problems at structural connections, especially at out-of-plane joint. However, detailed theoretical and experimental studies in context of out-of-plane joints are very rare at present.

The earliest work in this regard was due to Dixon, Ramsey and Usher [17], who conducted static and fatigue testing of out-of plane joints as shown in Figure 36. Fatigue loads are $5-25\%, 5-33\%, 5-50\%$ and $5-70\%$ of static failure load.

The main conclusions from the test were:

- Visible damage in the form of resin crazing and delamination occurred in some of the tests at loads and lives much less than those to cause final failure;
- Fatigue loading caused significant and progressive loss of stiffness in some but not all of the tests;
- Fatigue testing had no significant effect on subsequent static strength even in those cases where resin crazing and/or delamination were apparent after fatiguing;

It is noteworthy that Dixon, Ramsey and Usher investigated the effects of water immersion and elevated temperature on the fatigue behaviour of top-hat stiffener joint (see Fig.36 a). The first sign of damage on the immersed specimen at 30^0C was at 3748 cycles ($40,153$ cycles for unimmersed specimen) and the test was stopped at $49,820$ cycles ($78,600$ cycles for unimmersed specimen) because of extensive delamination at least as severe as in the unimmersed specimen. The results support the need for further study of this area.

This was followed by the work of Scholte and van Leeuwen [54], who carried out the fatigue test of the shell-bulkhead connection in the Ship Structure Laboratory of the Delft University of Technology. The stress ratio (R) was set at -1 and loading frequency during fatigue testing was variant from $0.6Hz$ at high loads with large deflection to $1.15Hz$ at the lower loads with small deflection. All together 12 specimens were tested. The following conclusions were derived from this work:

- Fatigue strength of the shell-bulkhead connection is strongly dependent on the strength properties of the resin;
- Delamination occurs and crack initiation starts at very low tensile loads and in a very early stage of fatigue life. This observation was in confirmation with the conclusions of the early study[17];
- Under fatigue loading many cracks develop in planes between the layers of reinforcement. The cracks show a large preference to the planes between the composing parts of the shell-bulkhead connection;

- Fatiguing does not cause a reduction in static tensile strength;
- The glass pins did not show a measurable effect upon the results [54].

A more recent study [55] has focussed on laminated tee-joints with configuration types B and F mentioned above. The load configuration and boundary condition are illustrated in Figure 6. The cyclic load pattern used was a sinusoidal wave set to vary from zero to a constant maximum pull-off load (stress ratio $R = 0$). A series of maximum load values were $90\%, 70\%, 50\%$ *and* 30% of the ultimate joint strength. Each of these tests was performed at a constant frequency of 1 Hz, which was chosen to reduce the risk of heating within the specimens.

The P-N curves derived from their fatigue test were shown in Figure 37. The curves exhibit the familiar inverse S-shape commonly seen for GRP laminates. Preliminary conclusions from this work are:

- the damage path and failure mode under cyclic loading are the same as in the static context;
- damage accumulation during fatigue of tee joints involves a degradation of the stiffness characteristics (see Figure 38 and 39);
- there could be a load value above which the fatigue process is geometry dependent and below which the fatigue process is material dependent. This feature, however, requires further study.

Although several very useful results were given, no attempt was made to derive theoretical methods on how to predict the fatigue life and on how to estimate the residual strength after fatigue, which are of great concerns of the researchers and designers in the marine structure in FRP.

8 Recommendations for Future Research Works

In conclusion we have shown that there is a significant body of evidence relating to the behaviour of such out-of-plane joints. While this seems adequate for many joint configurations, there is still a need for further work. Five principal areas where such effort is necessary are briefly outlined below.

(1) Accurate measurement about the basic material data

There is need for some basic material data, especially those pertaining to resin strength and modulus and laminate interlaminar tensile strength values, when finite element modelling is performed. Now, no such reliable data is available for the interlaminar tensile strength of laminates containing urethane-acrylate resin. So it is necessary to do accurate measurements about the basic material data.

(2) Theoretical methods used for design

Although the behaviour of out-of-plane joints can be examined using F.E. modelling it may be preferable, for intial design purpose, to make an approximate estimate of the ultimate strength without using a computer. So, there is an urgent need for the development of a design procedure which addresses concerned variables explicitly and which can readily be applied without resort to complex F.E. studies. Additionally, there is still a need for more F.E. studies to examine stress/strain patterns in the various sandwich joint configurations.

(3) Static testing and characterisation

This is concerned with sandwich tee-joints and top-hat stiffener configurations. With regard to the former, it is important to examine the influence of different combinations of geometric variables on strength and failure modes. With regard to the latter, there is a need to study the effect of varying both geometry and material parameters (as shown in Figure 16, for example).

(4) Fatigue performance

It is clear from the review above that the fatigue performance study in context of out-of-plane joints is limited to preliminary experimental research work. Therefore, apart from continuing to do experimental research on 'perfect' out-of-plane joints we should examine the fatigue performance of the out-of-plate joint with realistic damage which occurs to vessels in service to determine whether such damage is significantly worsened during a typical ship lifespan. It is also important to investigate the effects of water immersion and elevated temperature on the fatigue behaviour of out-of-plane joints. Attention should be paid to theoretical methods on how to predict the fatigue life and how to estimate the residual strength after fatigue.

(5) Impact performance

This is a most important topic with potentially serious consequences from a structural viewpoint. There are a number of publications relating to the impact performance of laminates, laminated plates and sandwich plate/beam panels. However, there is almost no reported evidence of work done with regard to out-of-plane joints and is thus a source for future attention.

9 References

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Table 1: **Details of laminated tee-joint test specimens**

Specimen	Boundary angle thickness (mm)	Fillet radius (mm)	Fillet overlay	Resin for $B.A.^a$ or overlay	Edge gap (mm)	Edge detail
A	10	30	–	Polyester	20	Plain
B	14.5	50	–	CR1200	20	Plain
C	–	75	–	–	15	Plain
D	–	75	–	–	25	Plain
E	–	75	–	–	15	$6mm$ Bevel
F	–	75	2WR	CR1200	15	Plain
G	–	100	–	–	15	Plain
J	–	75	2WR+CSM	CR1200	15	Plain
K	–	50	2WR	CR1200	15	Plain
L	–	75	2WR+CSM	Polyester	15	Plain
M	–	50	2WR	Polyester	15	Plain

a B.A.=Boundary angle

Table 2: **Impact of variations on laminated tee-joint performance**

Response feature	In plane stress in laminate	Through-thickness stress in laminate	Principal stress in fillet
Increasing thickness of overlaminate	Decreases	Increases	Decreases
Increasing radius: Overlaminate	Decreases	Decreases	Decreases
Increasing radius: Pure fillet	—	—	Approx. same
Impact of overlaminating on a fillet	—	—	Decreases
Impact of using CR1200 over polyester resin	Decreases	Decreases	Increases

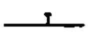
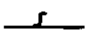
Table 3: Top hats: Impact of geometrical variations-Centre clamp

Increase in property	Effect on stiffness	Effect on stress in fillet	Effect on stress in overlaminate
Radius	Marginal increase	Minimum at $75mm$	Decrease
Overlaminate thickness	Marginal increase	Increase	Increase
Gap	Marginal increase	Decrease	—
Backfill angle	Marginal increase	Decrease	—

Table 4: Top hats: Impact of geometrical variations-Two clamp

Increase in property	Effect on stiffness	Effect on stress in fillet	Effect on stress in overlaminate
Radius	Increase	Minimum at $75mm$	Decrease
Overlaminate thickness	Increase	Decrease	Increase
Gap	Maximum at $30mm$	Minimum at $30mm$	—
Backfill angle	Marginal increase	Marginal increase	—

Table 5: Comparison of alternative superstructure panel designs

Design particulars		Overall bending		Local panel bending		Relative weight	Relative cost
		σ_{max}/σ_p	w_{max}/w_p	σ_{max}/σ_p	w_{max}/w_p	W/W_A	$£/£_A$
Design A (Reference design: 5083 alloy)		0.91	0.39	0.75	0.21	1.0	1.0
Design B (Grade 50D steel)		0.63	0.20	0.76	0.24	1.98	0.35
Design C (all-GRP(hand-laid))		0.68^a	0.72^a	0.64^a	0.89^a	0.87	0.85
Design D (6082 alloy bulb angle on hot-pressed GRP)		0.65^a	0.56^a	0.91^a	0.42^a	1.00	0.74
Design E1 (Cold-formed HY80 Z-section on hot-pressed GRP)		0.78^a	0.40^a	0.68^a	0.41^a	1.11	0.54
Design E2 (Cold-formed HY80 hat-section on hot-pressed GRP)		0.51^a	0.30^a	0.68^a	0.54^a	1.23	0.59
Design F (hand-laid corrugated GRP)		0.67^a	0.74^a	0.49^a	0.26^a	0.74	0.71
Design G (hat-section GRP stiffener on balsa-core GRP sandwich)		0.84	0.99	0.87	0.64	0.71	0.75
Design H1 (unstiffened PVC foam core sandwich: $100mm$ core)		2.12	3.84	-	-	0.56	0.76
Design H2 (as H1 but with $200mm$ core)		1.05	1.04	-	-	0.88	1.00

w_p (overall bending) = $25mm$; w_p (local panel bending) = $10mm$

[a] Obtained by folded-plate analysis.

Table 6: Comparation of experimental results with theoretical prediction values

Specimen	Experimental failure load (N)	Theoretical failure load (N)	Maximum deflection (mm)	Slope (N/mm)
Triangular foam insert	852	1000	85.10	10.012
Foam pad	1080	1050	91.00	11.868
Filler fillet (radius 10 mm)	1070	995	95.41	11.215
Filler fillet (radius 25 mm)	920	1005	93.00	9.832
Filler fillet (radius 40 mm)	1000	1055	101.00	9.901

Table 7: The basic mechanical properties of the flexible resin and A2785CV polyester resin

Resin Matrix	Initial Tensile Modulus (GPa)	Tensile Strenhth (MPa)	Strain to Failure (%)
A2785CV Isophthalic Polyester resin	3.6	26	1.8
Scott Bader Modified Acrylic Oligomer resin- Crestomer 1080	0.5	25	120

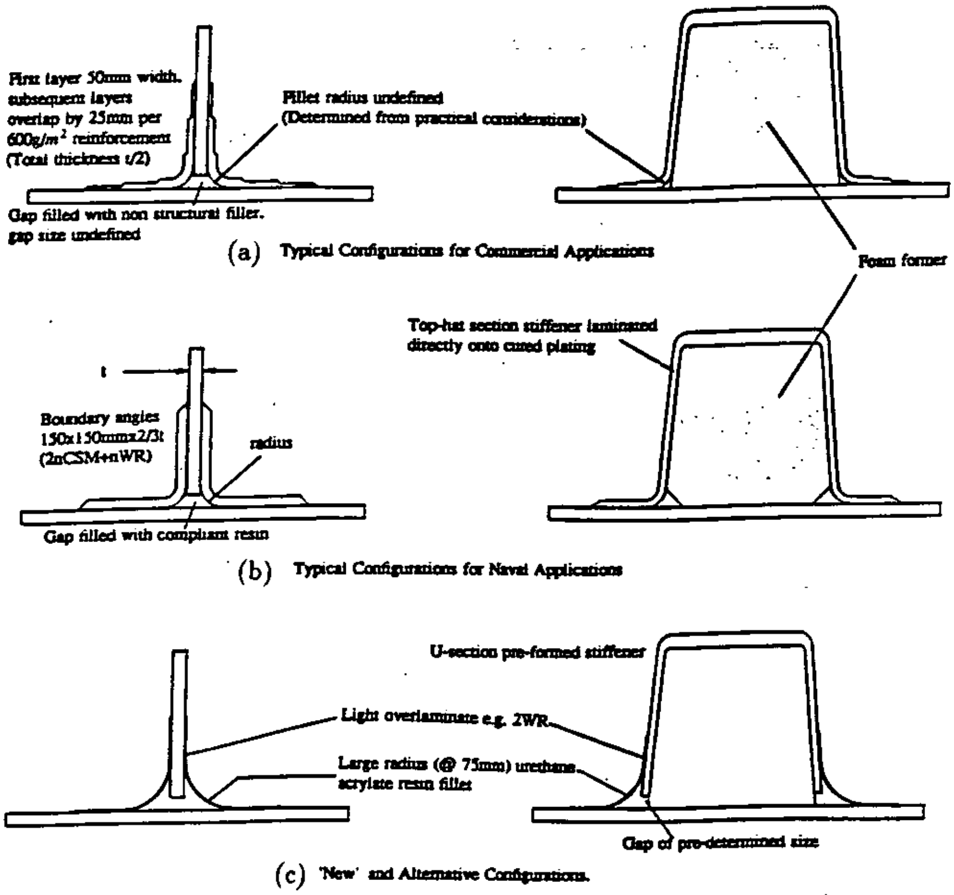
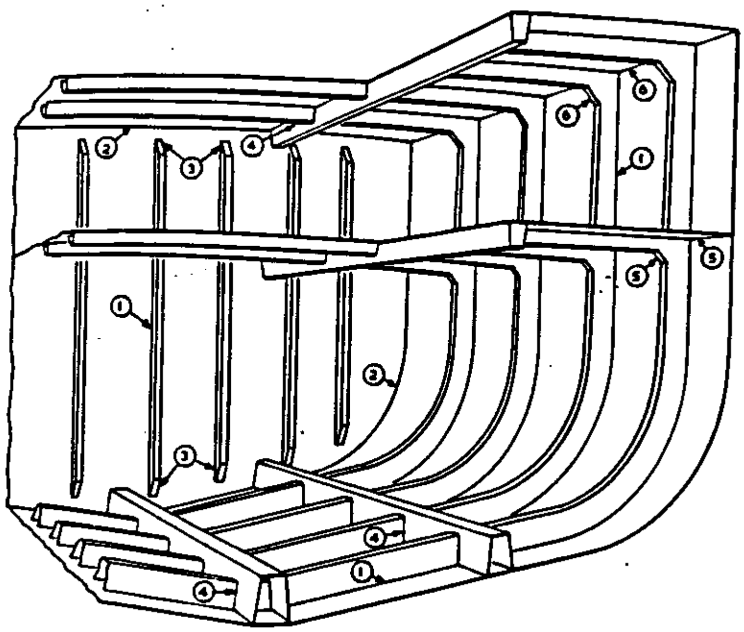


Figure 1 Tee-joint and top-hat stiffener configurations.



(1) Frame/shell
(2) Bulkhead/shell
(3) Stiffener ending
(4) Stiffener intersection
(5) Deck-edge (tee)
(6) Deck-edge (knee)

Figure 2 Ship hull compartment showing typical structural connections.

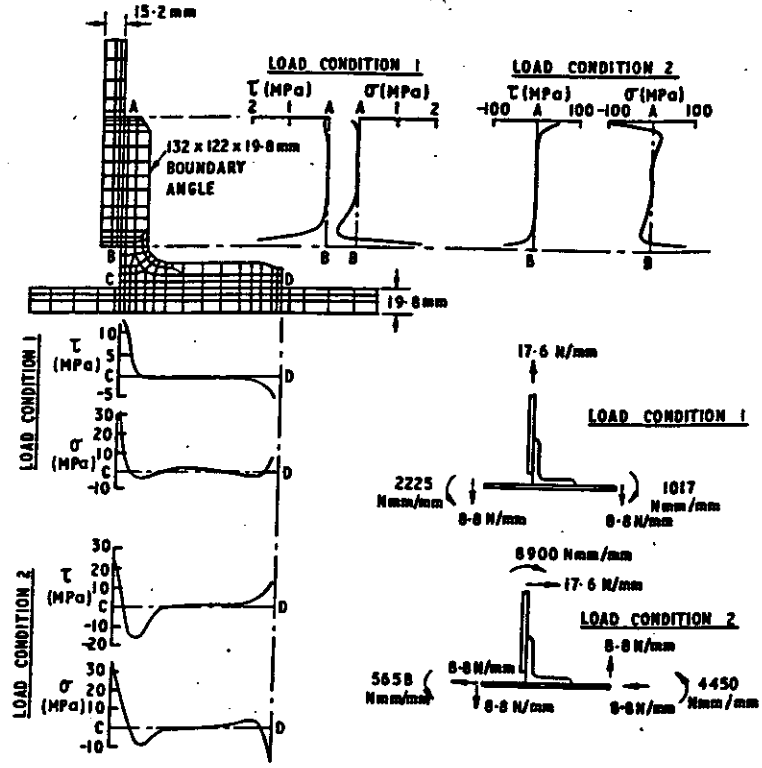


Figure 3 Stress analysis of bulkhead/shell connection.

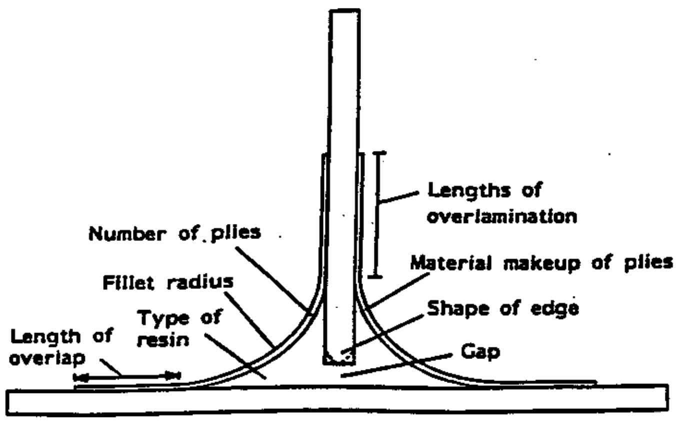


Figure 4 Tee-joint design variables.

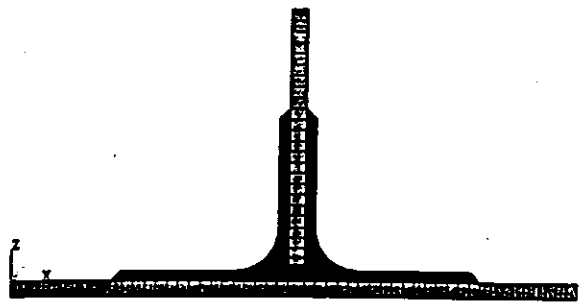


Figure 5 Typical F.E. model of tee-joint.

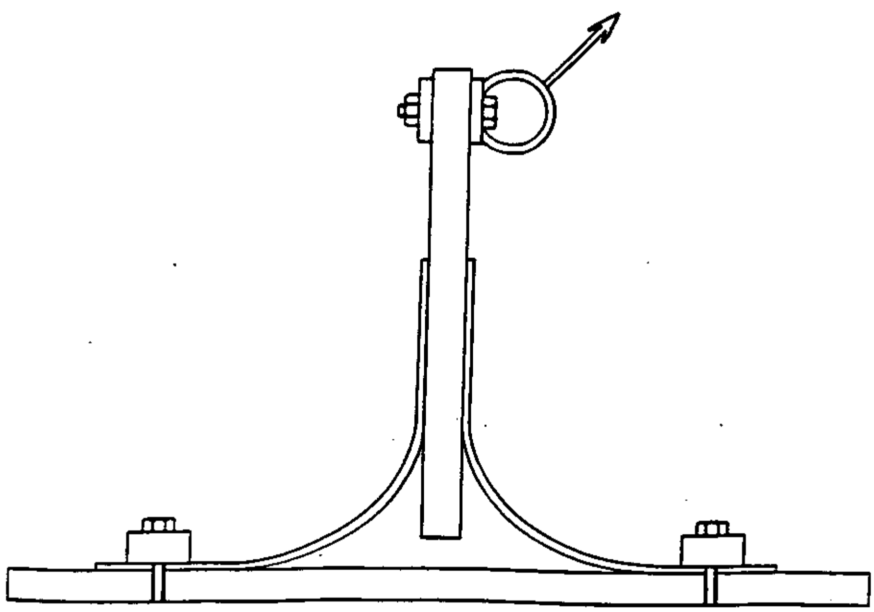


Figure 6 Edge restraints and loading method for tee-joint test.

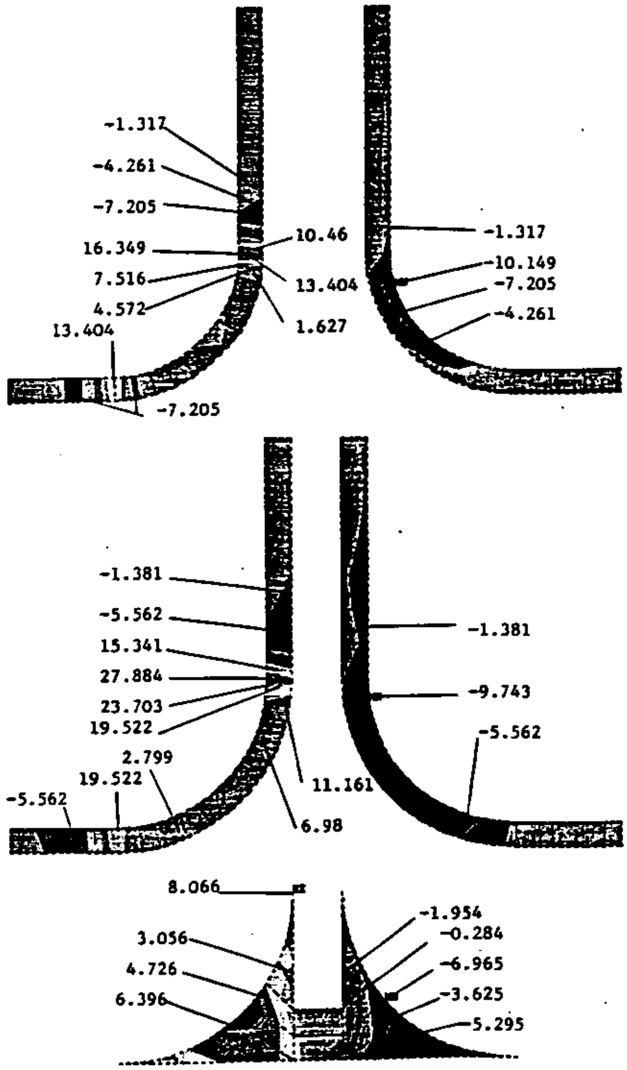


Figure 7 Stress contours within tee-joint Sample B–with gap elements. (All stresses in MPa.)

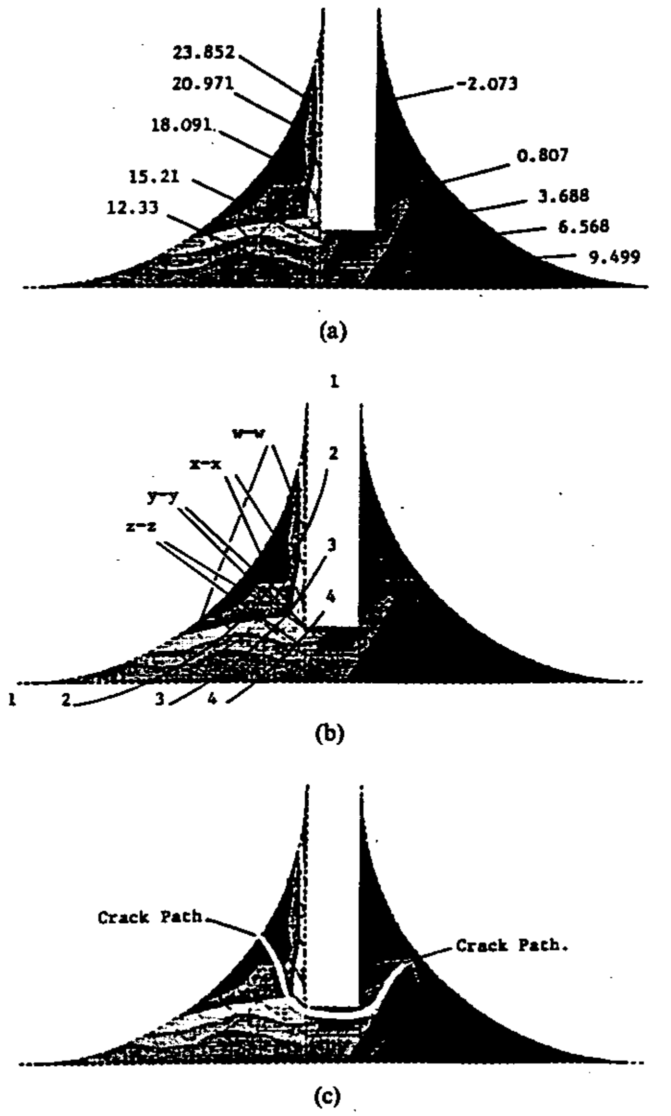


Figure 8 Stress contours within tee-joint– Sample C. (a) Stress contours within sample C; (b) region of high stress concentrations in sample C; (c) indicated route of crack propagation in sample C. (All stresses in MPa.)

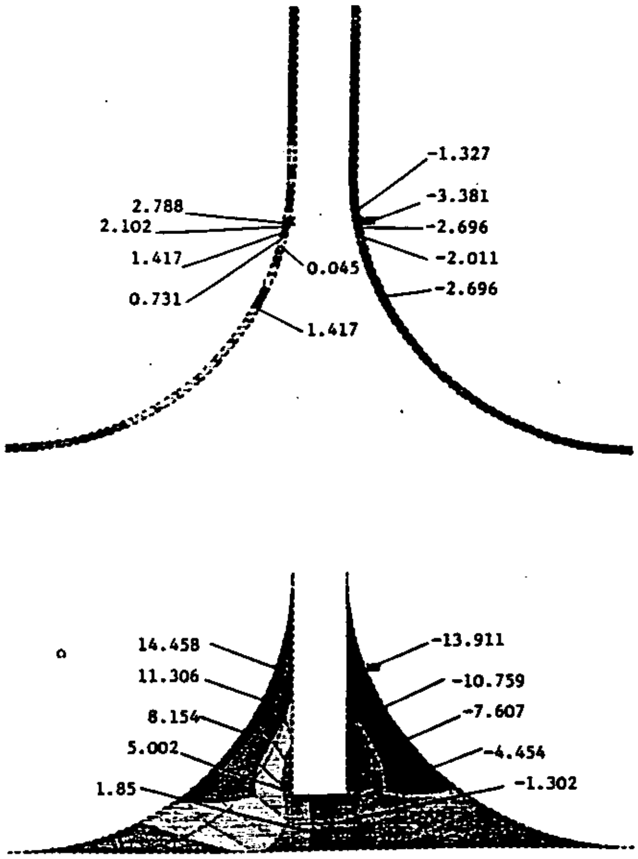


Figure 9 Stress contours within tee-joint– Sample F. (All stresses in MPa.)

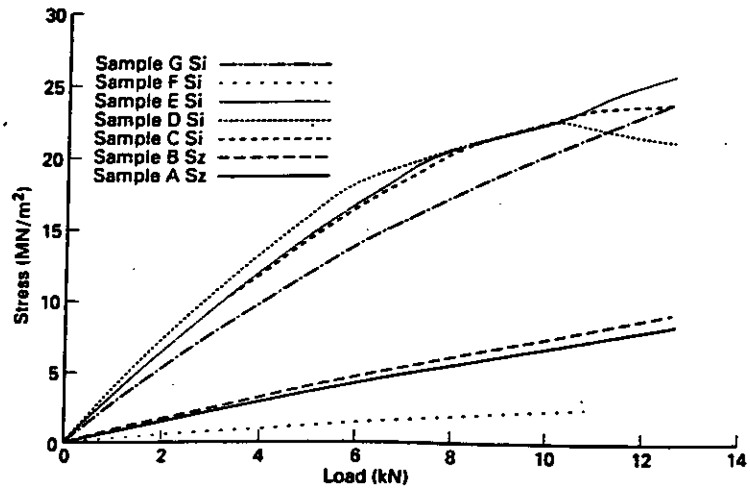


Figure 10 Peak stresses in tee-joint versus load.

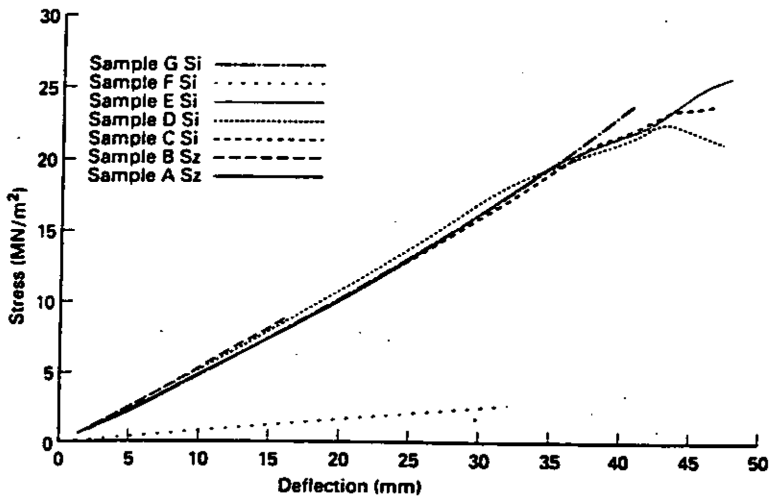


Figure 11 Peak stresses in tee-joint versus deflection.

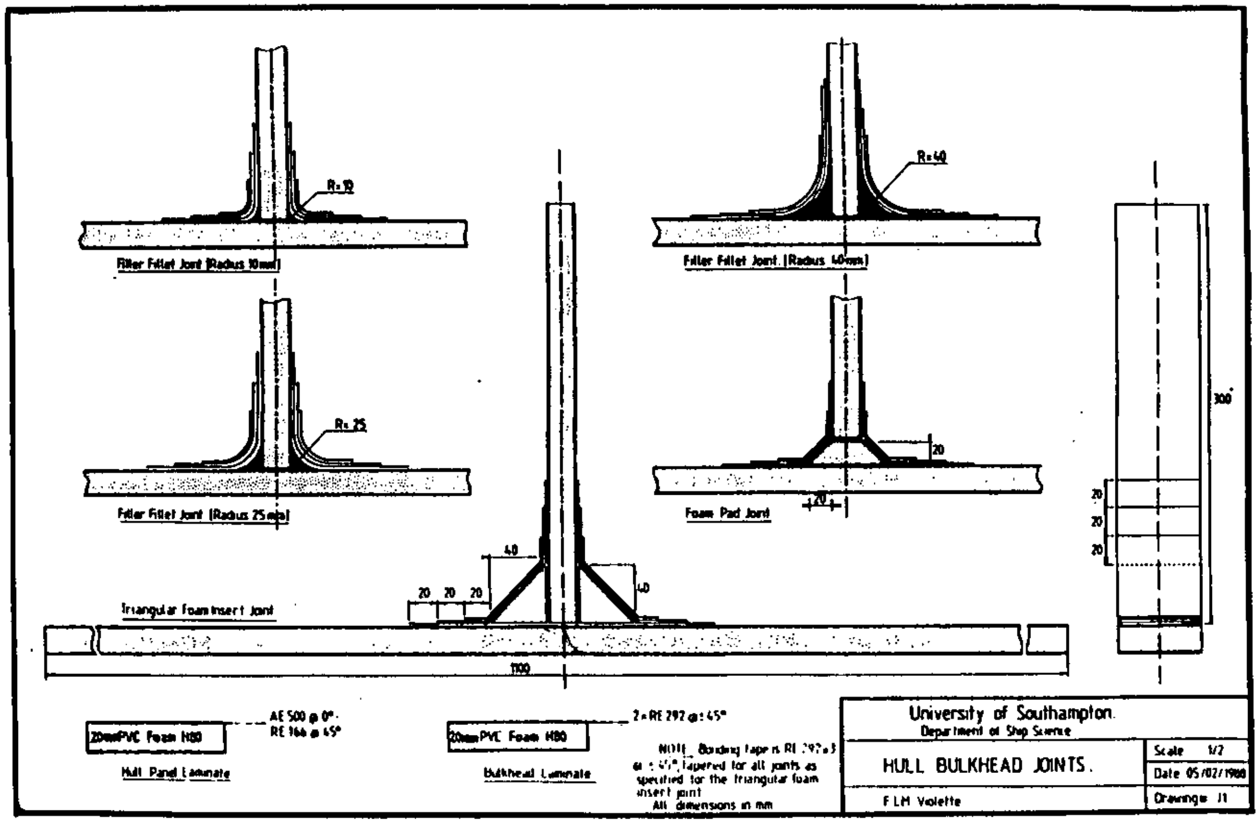


Figure 12 Design particulars of sandwich tee-joints.

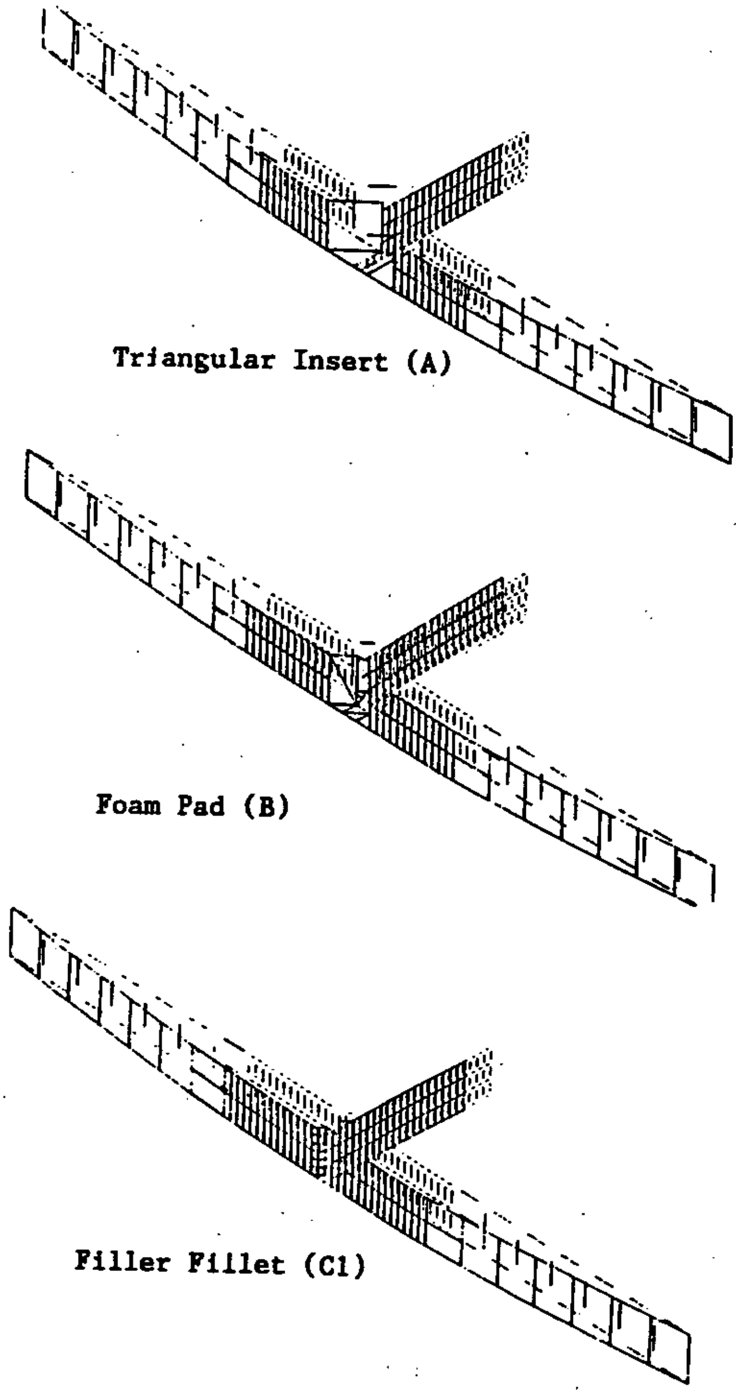


Figure 13 Finite element deflection plots.

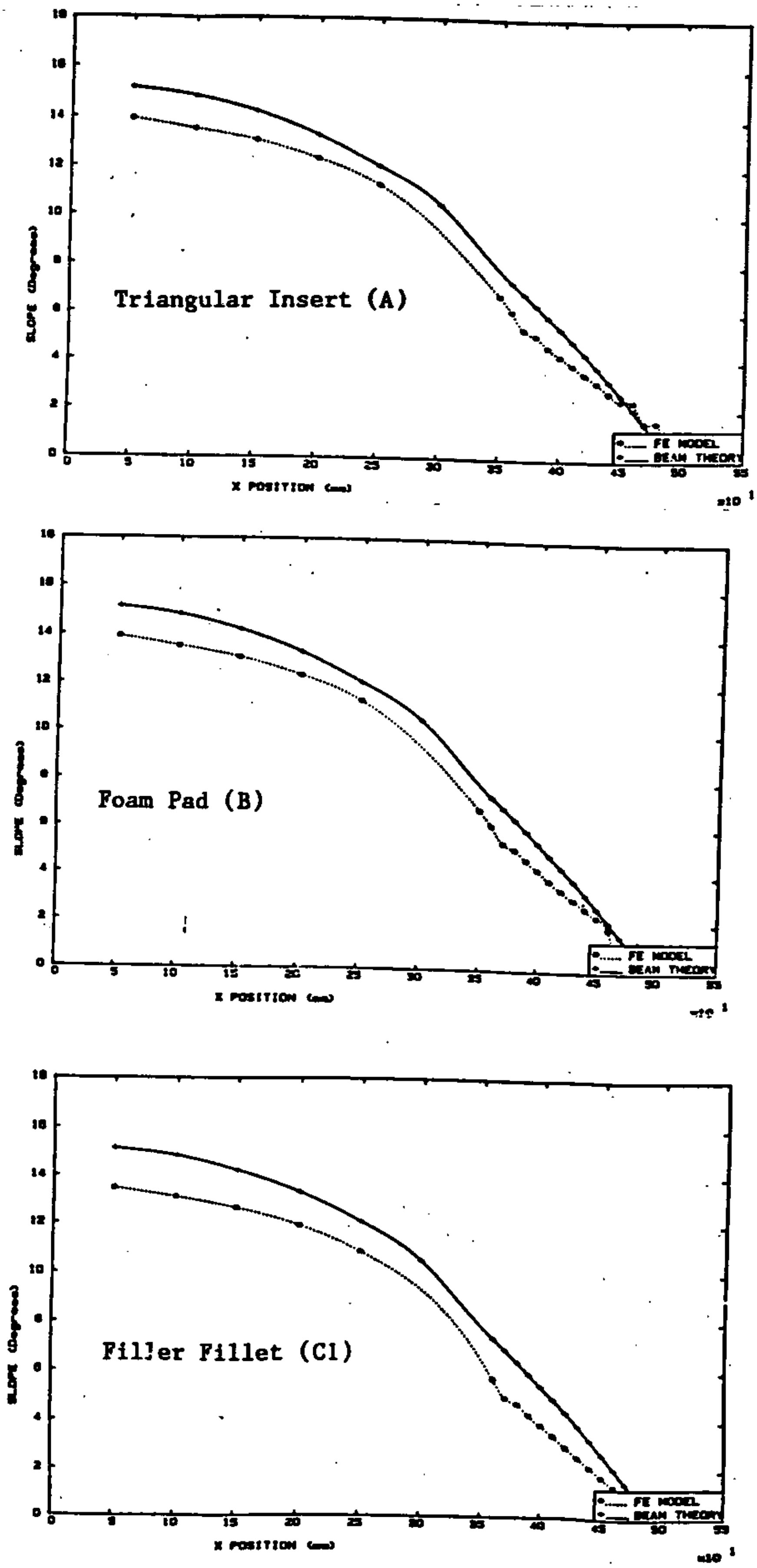


Figure 14 Slope along span: analytical model validation.

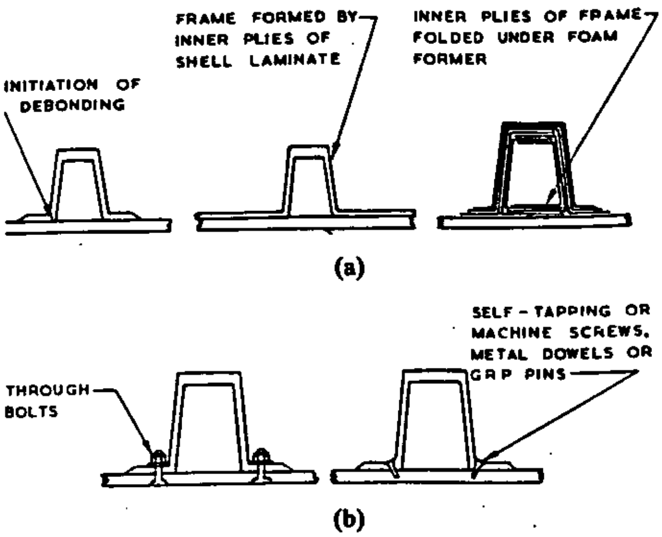


Figure 15 Frame/shell joints: (a) types of frame/shell attachment; (b) reinforcement of frame/shell joint.

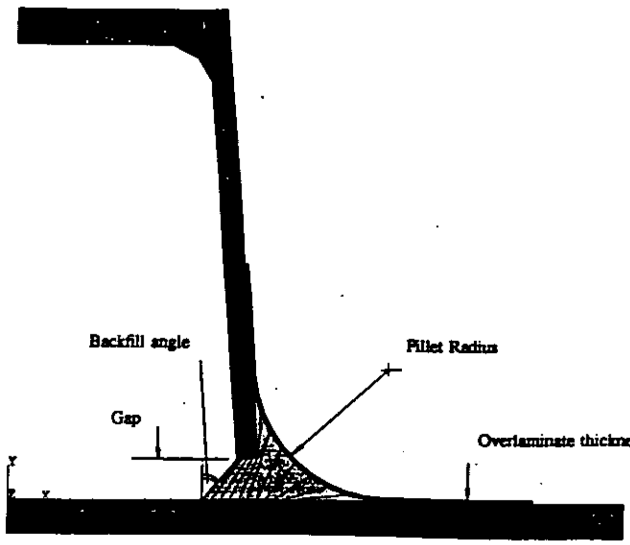


Figure 16 Design variations considered for top-hat stiffeners.

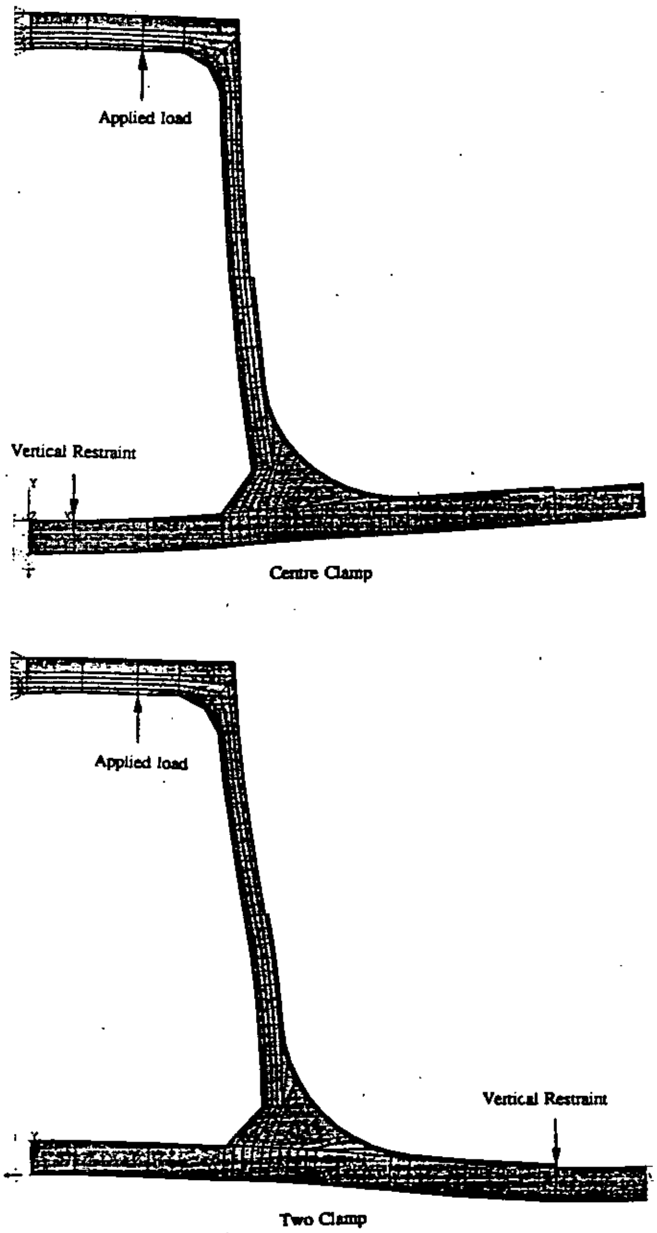


Figure 17 Boundary conditions for top-hat stiffeners.

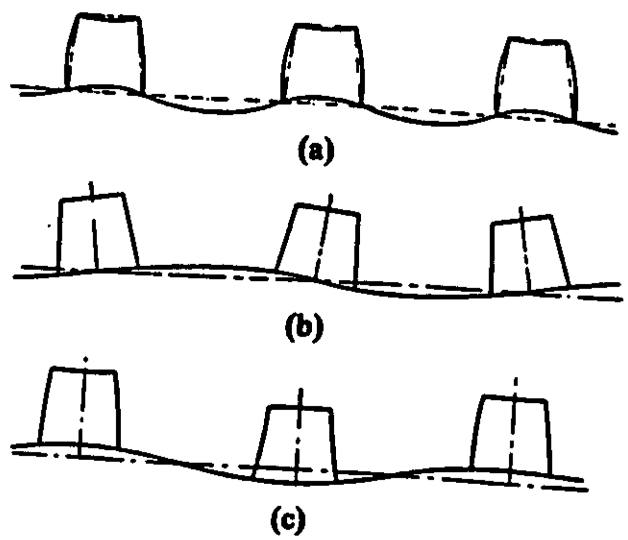


Figure 18 Interframe buckling modes for top-hat stiffened panels

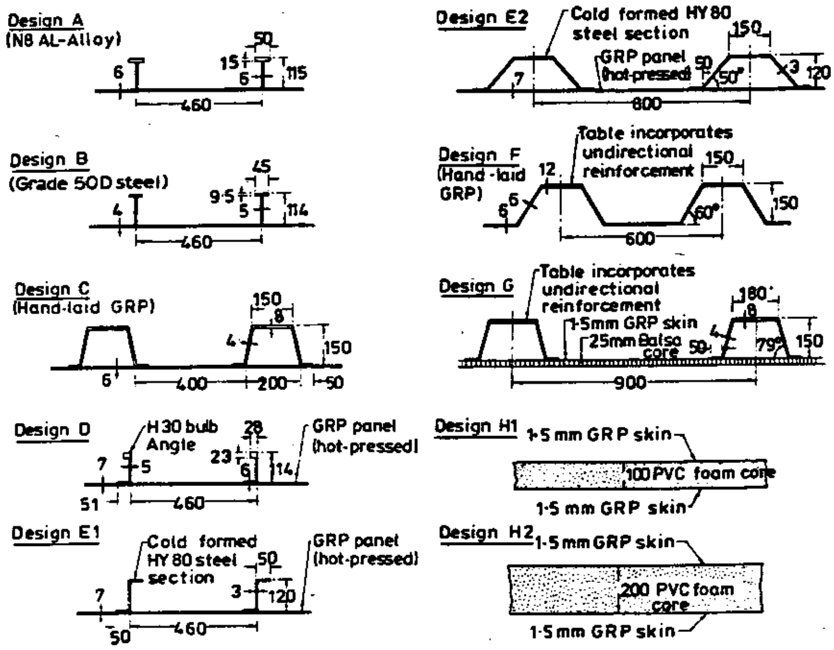


Figure 19 Alternative metal, GRP and hybrid stiffened panel designs.

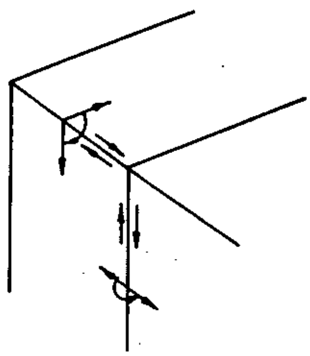


Figure 20 Forces and moments trasmitted at panel/panel joints.

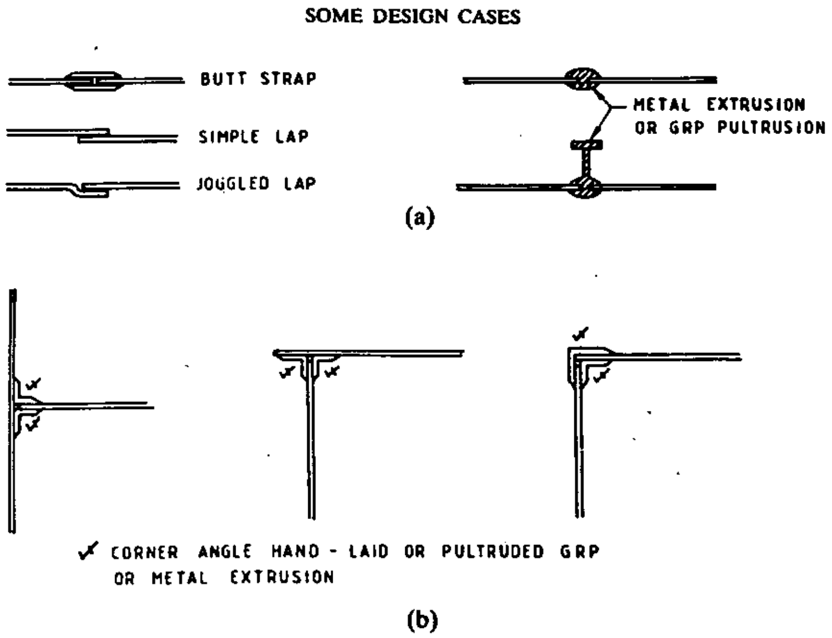


Figure 21 (a) Butt joints between GRP panels: (b) corner joints between GRP panels.

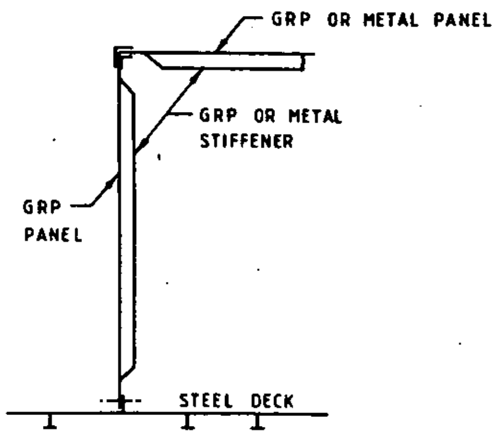


Figure 22 Pin-jointed panels.

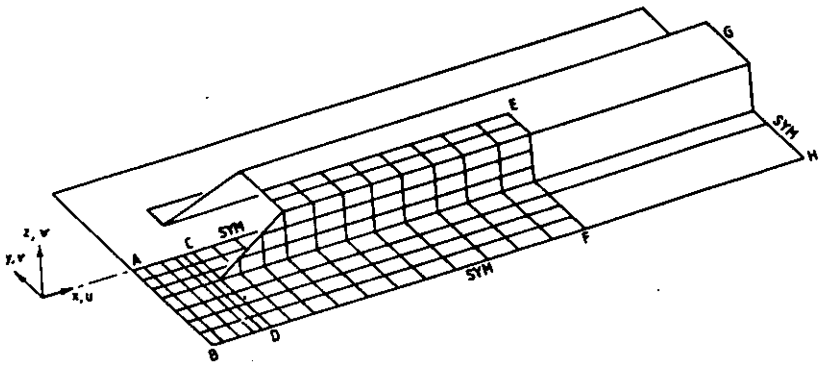


Figure 23 Finite element model of hybrid stiffened panel.

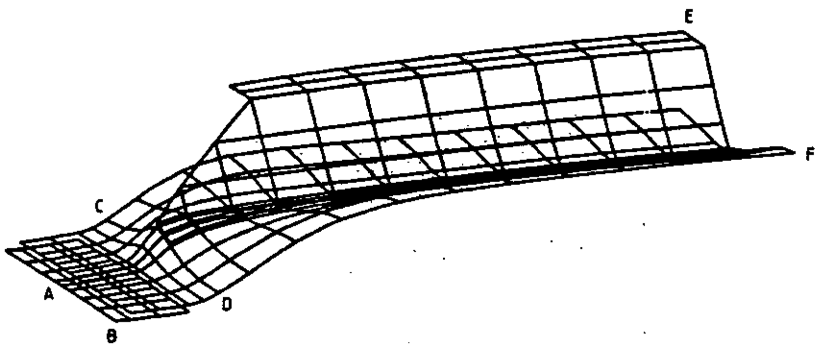


Figure 24 Computed deformation of hybrid stiffened panel.

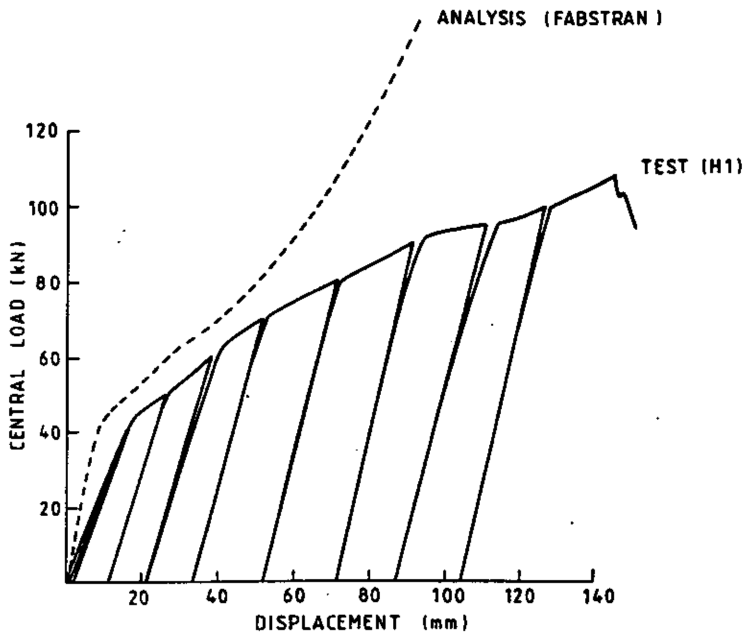


Figure 25 Autographic mid-span load-displacement record for specimen H1–compared with computed curve.

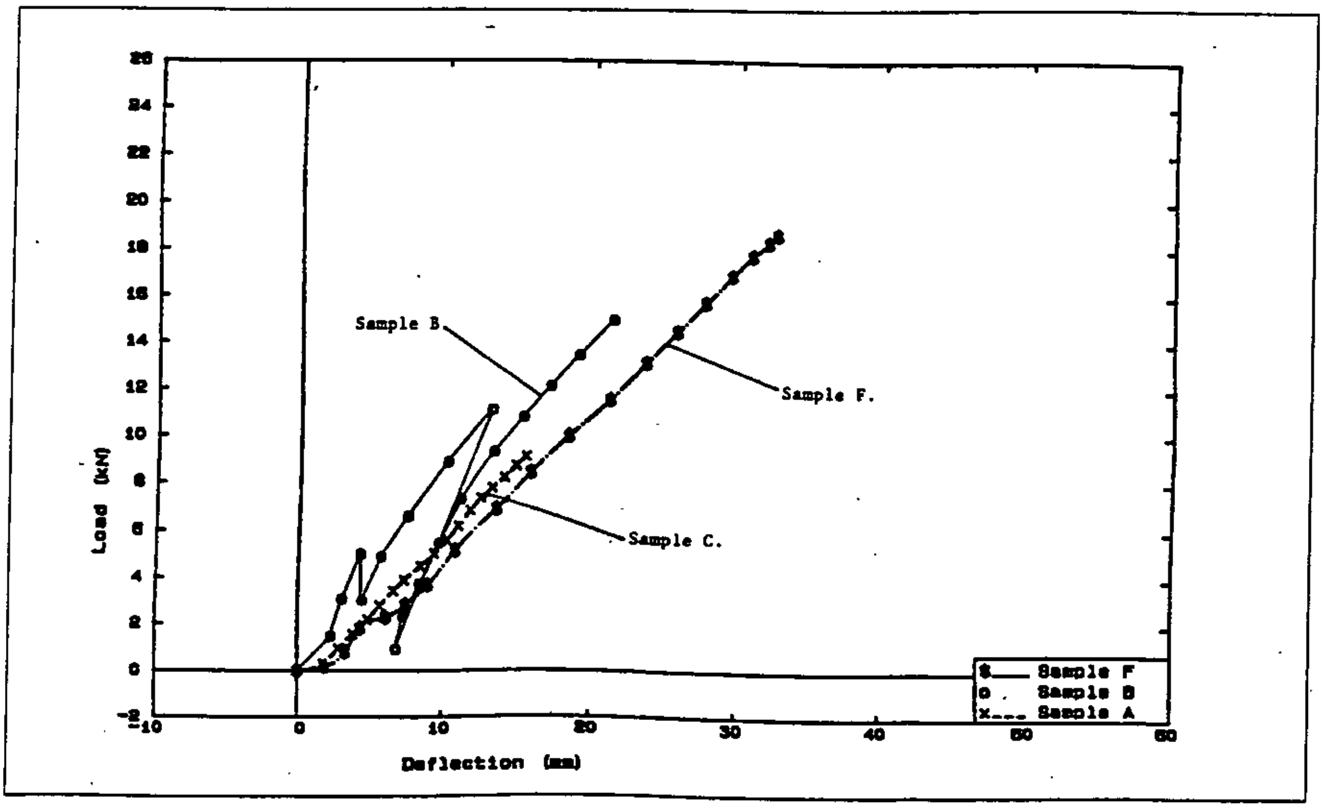


Figure 26 Load-deflection plots for static test samples.

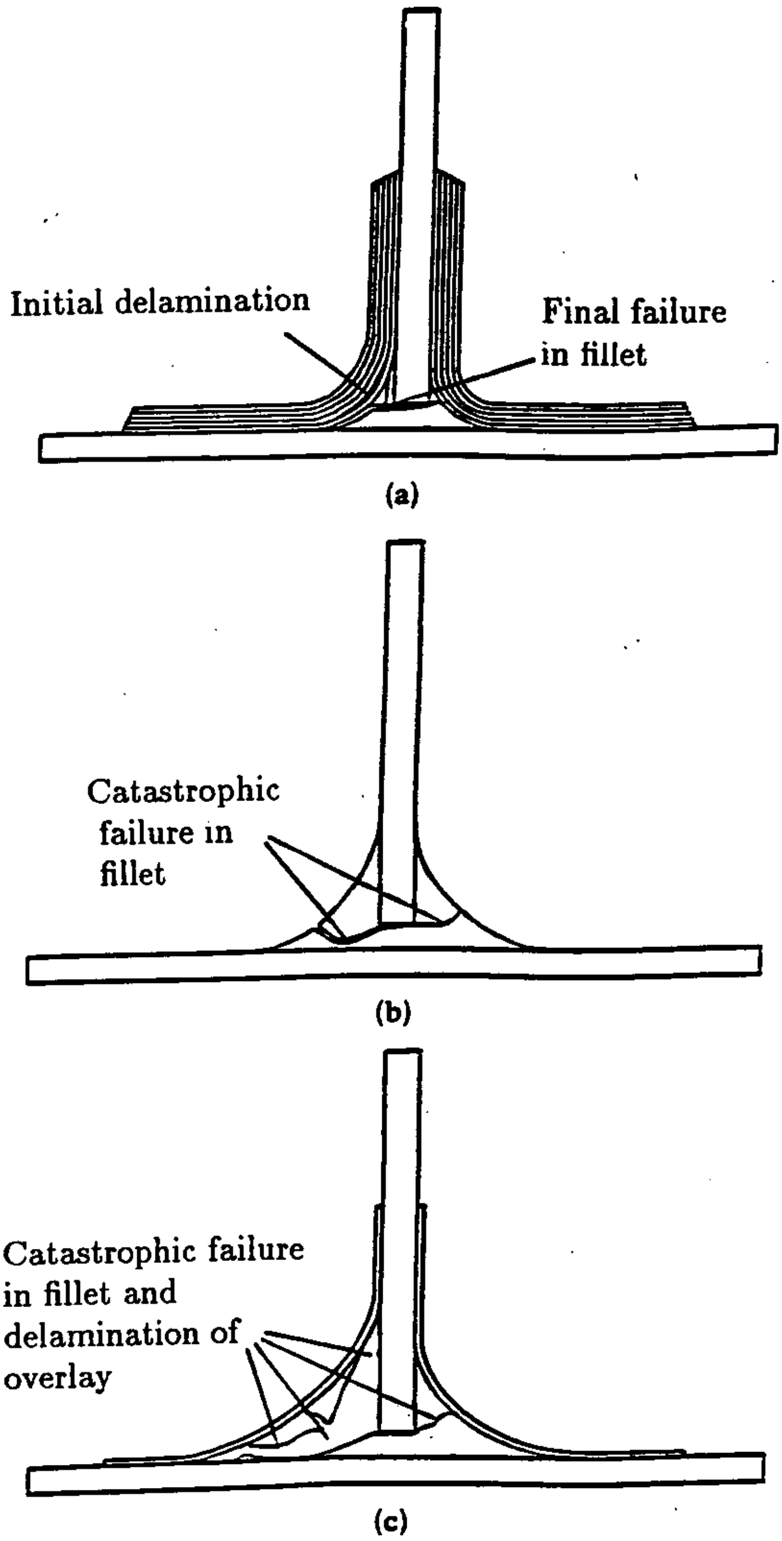
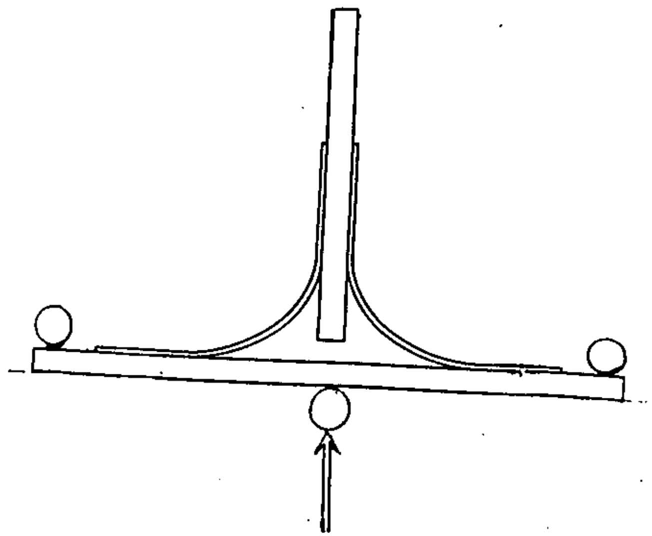
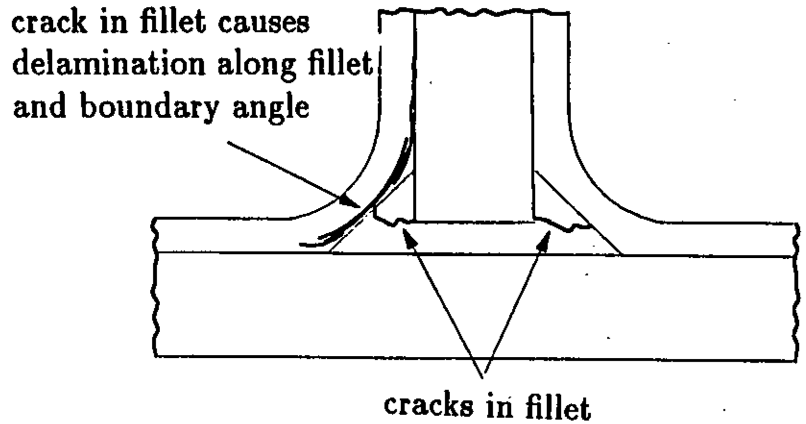


Figure 27 Failure configurations in various samples.

(a) Configuration B; (b) Configuration C (c) Configuration F

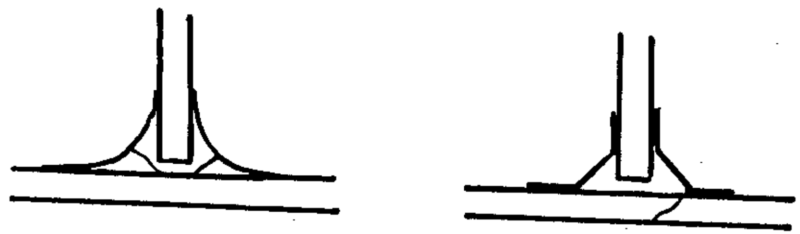


(a) Loading and boundary condition

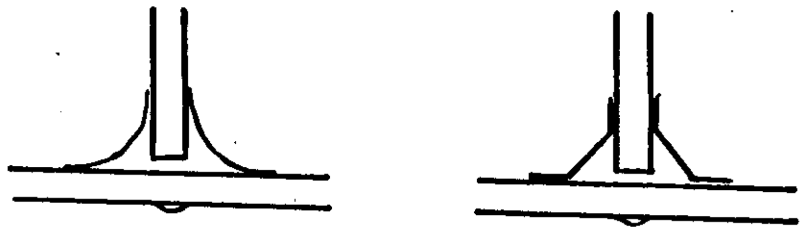


(b) Failure mode

Figure 28 Single skin tee-joint failure mode in 3-point bending



(a) Compressive test



(b) Tensile test

(c) Bending test

Figure 29 Failure modes in sandwich tee-joints

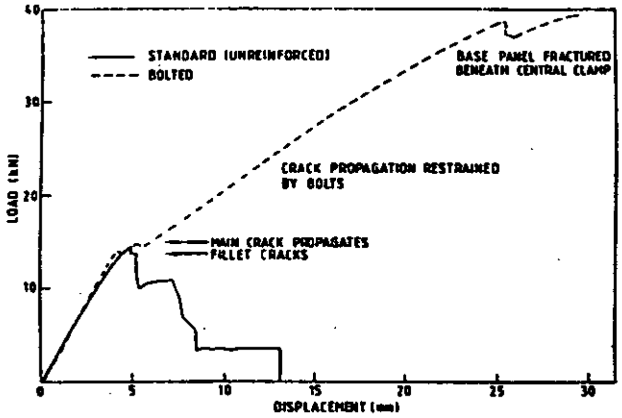


Figure 30 Representative load versus displacement curves for standard (unreinforced) and titanium bolted specimens.

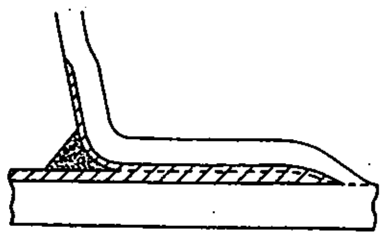


Figure 31 Design B with high strain to failure flexible resin fillet with additional woven roving plies.

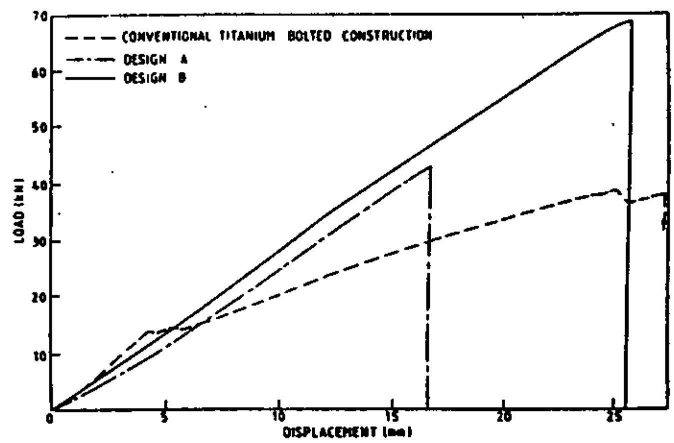
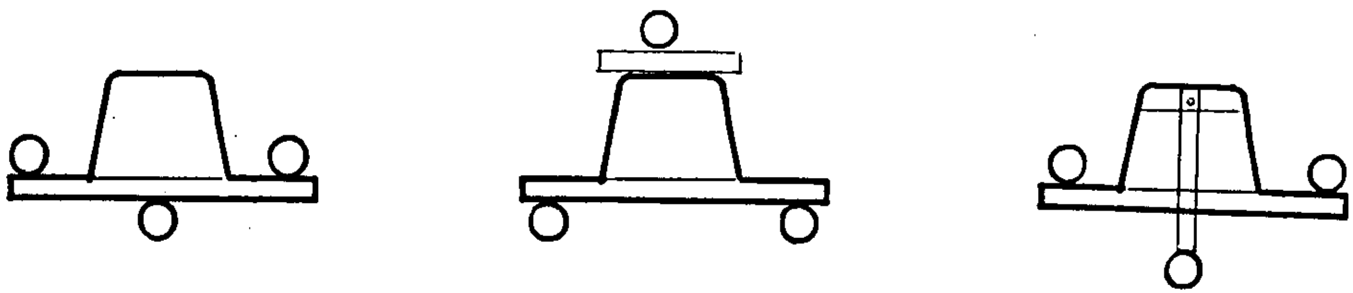
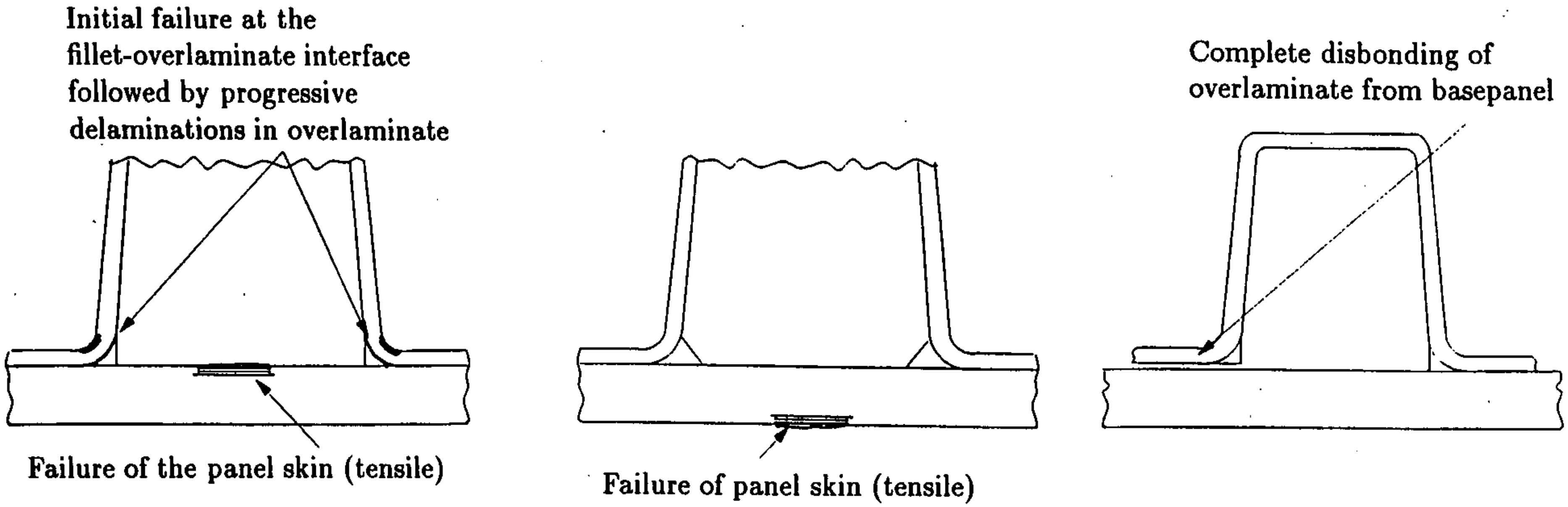


Figure 32 Comparative slow pull-off performance between a conventionally bolted and two toughened top-hat constructions.



(a) 3-point bend (b) Reverse 3-point bend (c) Pull-off

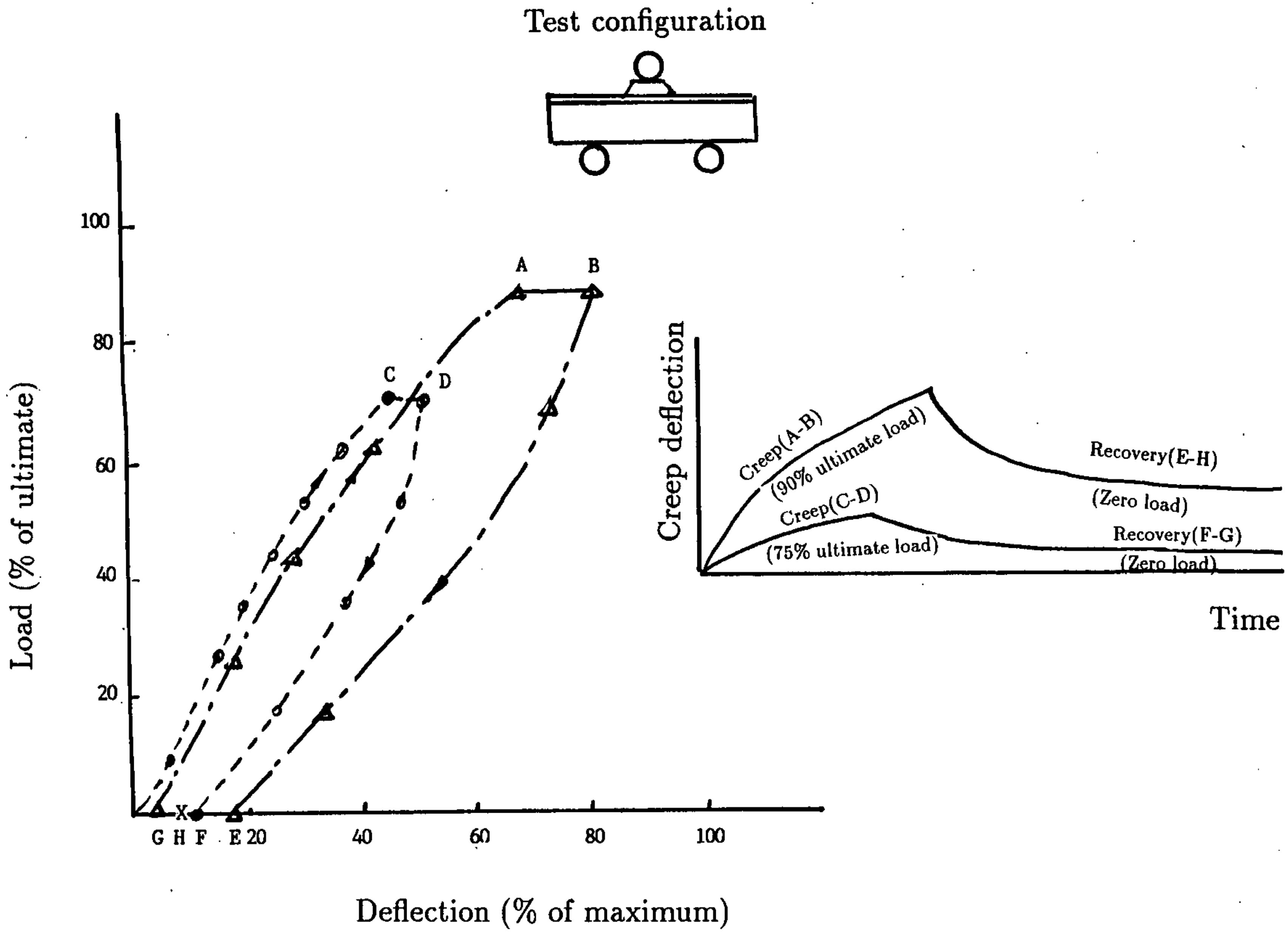
Figure 33 Load configurations for top-hat stiffener tests

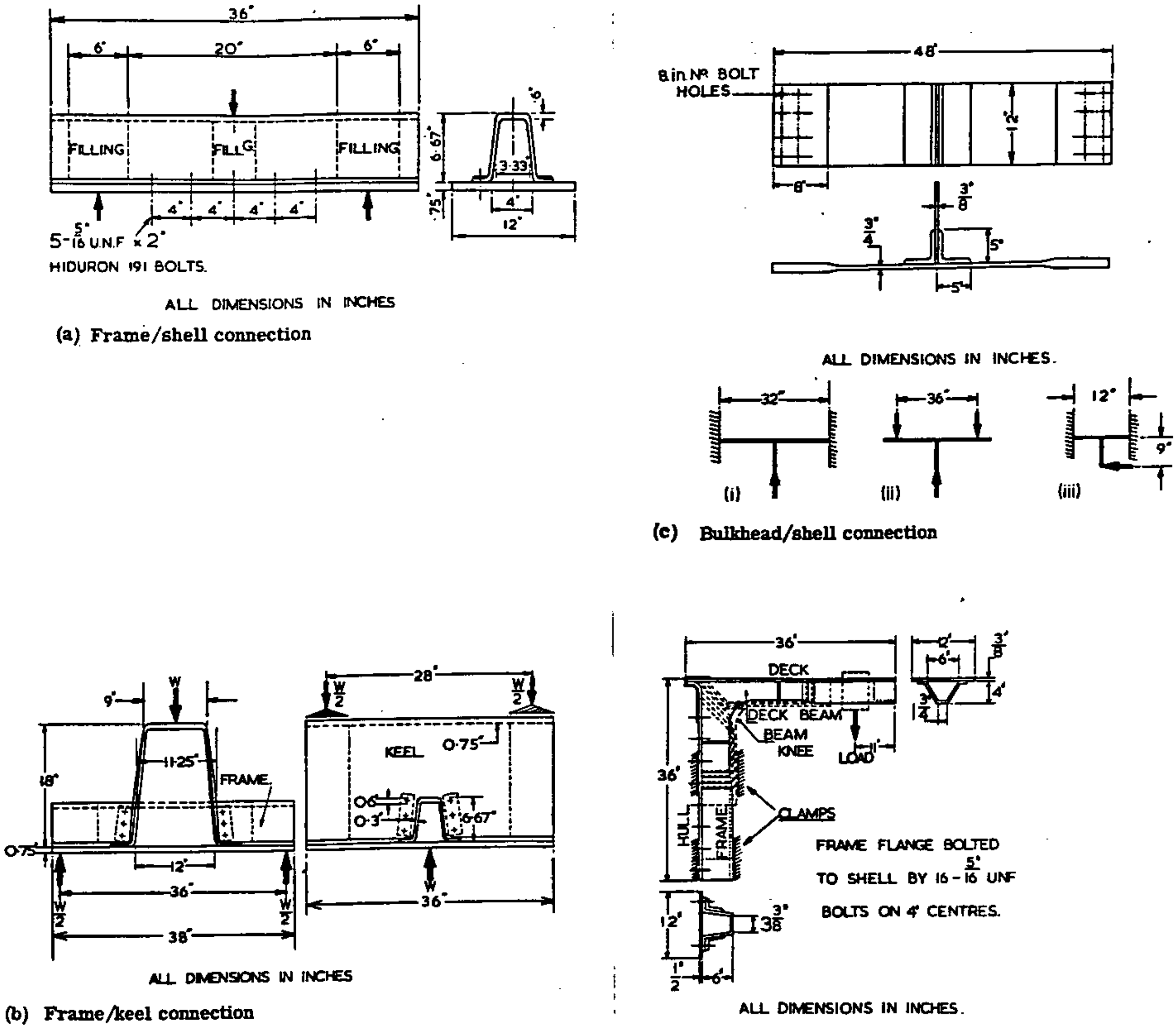


(a) 3-point bend (b) Reverse 3-point bend (c) Pull-off

Figure 34 Failure modes in top-hat stiffeners

Figure 35 Representative load-deflection and creep curves





(a) Frame/shell connection

(b) Frame/keel connection

(c) Bulkhead/shell connection

(d) Weather deck/shell connection.

Figure 36 Static and fatigue tests of typical structural connections.

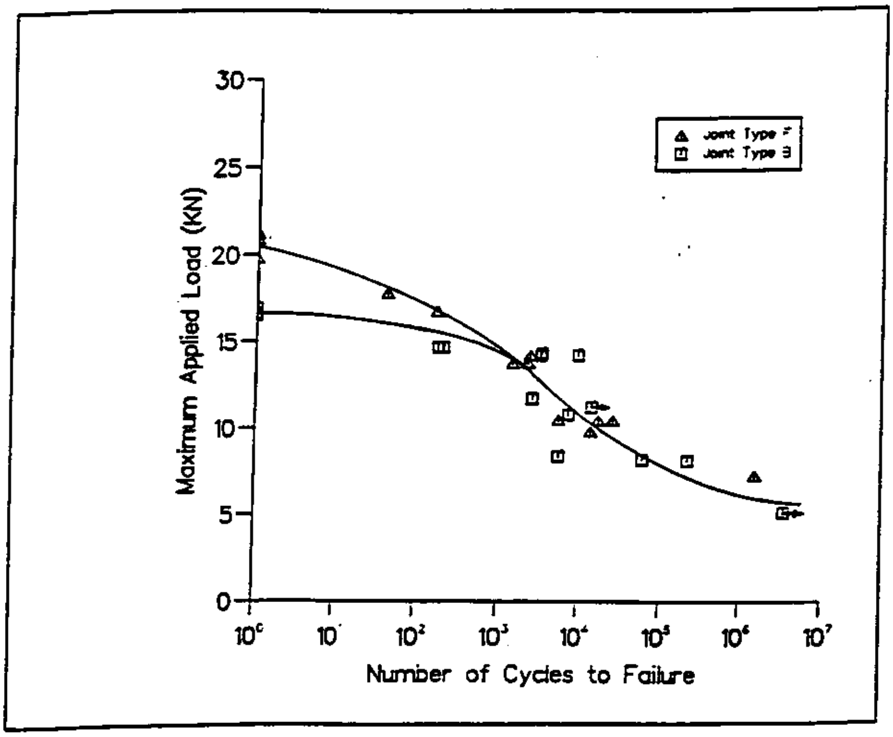


Figure 37 P-N curves for joint types B and F

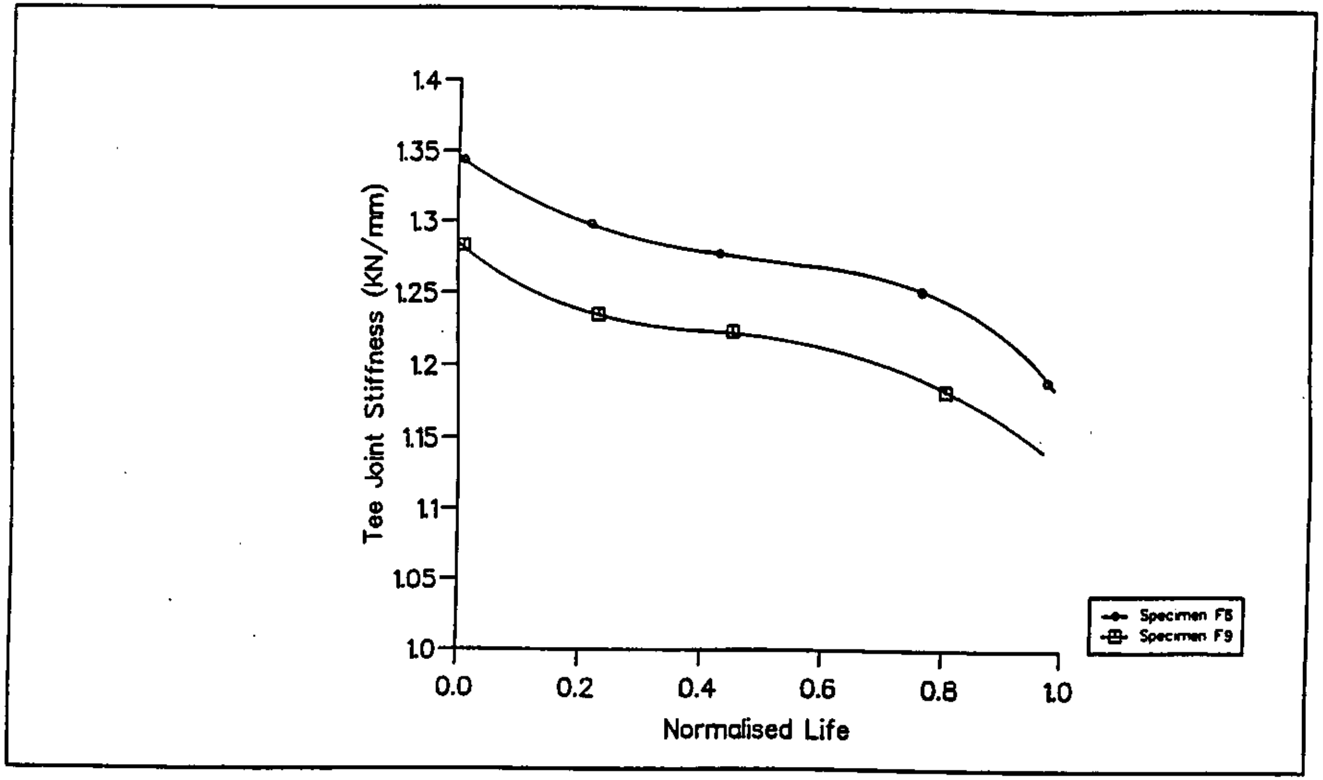


Figure 38 Stiffness degradation in two F specimens

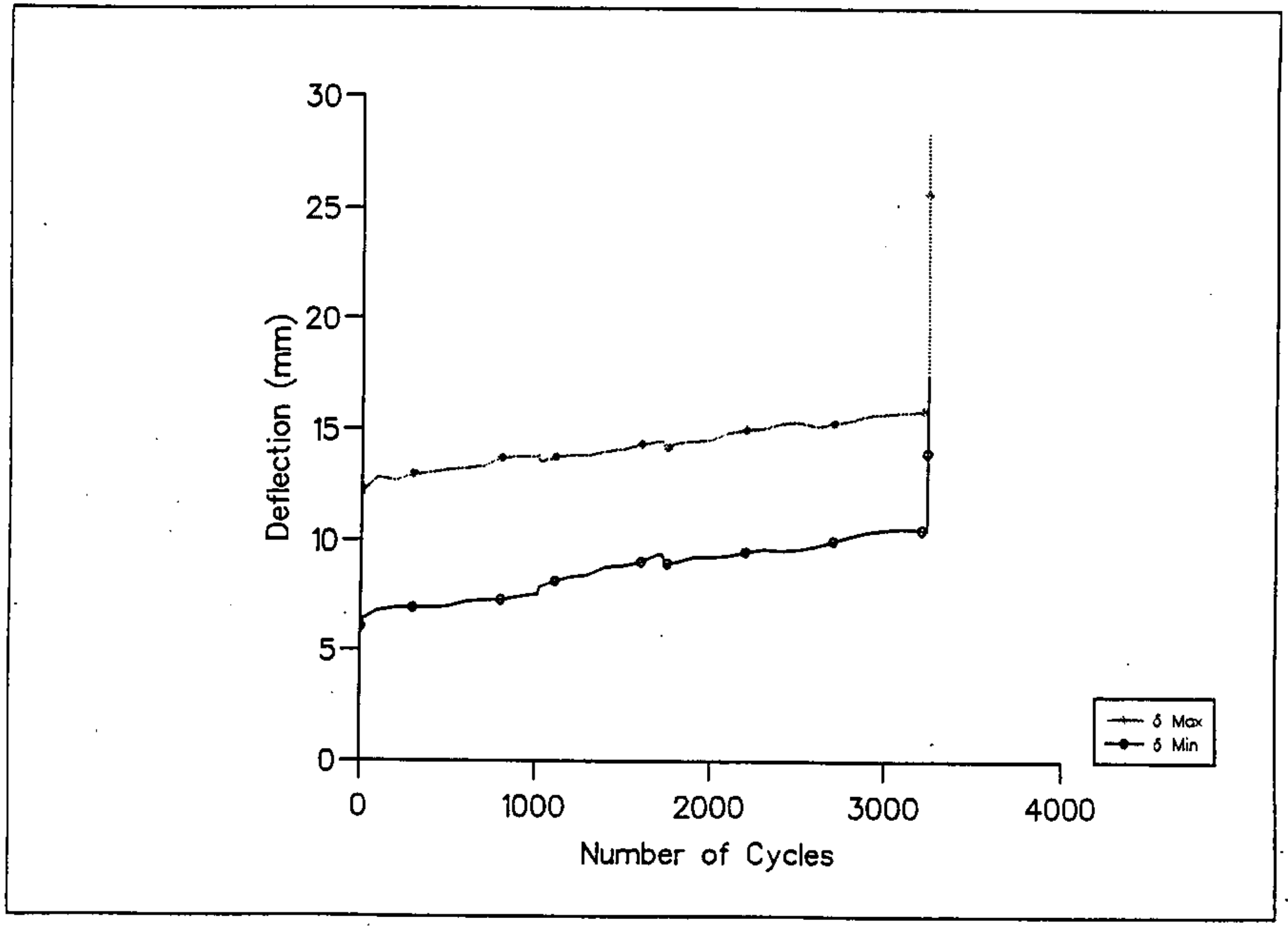


Figure 39 Deflection versus number of cycles for a B type specimen.