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UNIVERSITY OF SOUTHAMPTON

DEPARTMENT OF SHIP SCIENCE

FACULTY OF ENGINEERING AND APPLIED SCIENCE

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R.A. Shenoi and G.L. Hawkins

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Precis of Contract No. GR/F86151, November 1990 - 1992.

Large Deflection Behaviour of GRP Panels with Attachments.

Principal Investigator: Professor G. J. Goodrich. Dr. R. A. Shenoi.
Reasearch Assistant: Mr. G. L. Hawkins.
Department of Ship Science, University of Southampton.

In large GRP structures the low modulus of the material means that all elements must be adequately supported to prevent excessive deflections. In a marine context this is usually achieved by the inclusion of bulkheads and top hat stiffeners. Since these are stiffening a naturally flexible structure the joints between these stiffening elements and the main structure are subjected to high loadings.

The work conducted in the course of this contract has aimed to develop an understanding of the mechanics of these joints, and to determine the factors that effect their performance. This has been achieved by several methods.

Firstly a thorough background study was made to assess previous work in this field. Current design practices were compared, and the history of their development was traced. This highlighted several "standard" test methods which have been used as a basis for the new work, allowing direct correlation with previous designs.

A series of different design were developed, taking account of previous work and present production considerations. These were tested experimentally and analysed theoretically, using the finite element method. The results of this work clearly showed the mechanics of the joint behaviour and this enabled a "new" design of joint to be developed.

A systematic parametric study was then conducted to assess the effect of varying the principal parameters of this new design. The results from this study provide the basis of a new design optimisation procedure.

The success of these analyses in optimising the joint design is shown when, in experimental tests, the new design failed at a load 1.5 times the ultimate load of the best current design without any of the premature delaminations associated with this. In full scale tests the new joint design remained intact at the limit of the test rig, and remained similarly undamaged when subjected to explosive loading. This increase in performance is achieved with a 60% weight saving, and a joint that is considerably simpler, and hence cheaper, to fabricate.

Whilst conducting the finite element analyses it became clear that the codes used, whilst adequate for comparative analysis, were limited in their ability to accurately define the behaviour of the composite materials in a quantitative sense. This is necessary when considering larger and more complex problems, and particularly when attempting to predict failure and follow its propagation. Solutions to overcome these shortcomings have been proposed, and a code is being developed to assess their practicality. This work is ongoing.

Large Deflection Behaviour of GRP Panels With Attachments Joint Research Contract between SERC, Vosper Thornycroft, and the Department of Ship Science, University of Southampton

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Introduction

Glass reinforced plastic (GRP) is used for the construction of many large structures which are subjected to high and complex loads. Designers try to ensure that the principal loads are carried in the plane of the fibre reinforcements but inevitably some out-of-plane loading occurs (typically pressure loading on panels). The relatively low modulus of rigidity of the material results in high out-of-plane displacements so designers incorporate additional structural elements to control these.

In the marine field this is typically done by adding top hat stiffenners and bulkheads. The design of the methods of attachment of these elements to the main structure is critical because high loads need to be tranferred and there is inevitably a large differential in stiffness between the two items being joined. Additionally it is rarely possible to incorporate fibre reinforcements across the joint interface so loads must be transfered via the matrix, or additional adhesive materials.

In a large and complex structure such as a ship these joints can make up a significant proportion of the structural weight and require many man-hours to produce. Hence there is a need to develop a suitable design procedure which will result in more efficient jointing methods. This was the purpose of this project, outlined in the Proposal, which is included here as Appendix 1.

Objectives

The main aims of the project were:

1) To model non-linear, large deflection behaviour of attachments to plating.

2) To validate the theoretical models through simple (static) laboratory experiments.

3) To apply the analytical scheme to the practical design of attachments in GRP structures.

Achievements

A. Background Study

An extensive literature search has been conducted to review any previous work in this area. Many references about the joining of composites have been found in the marine, aeronautic, and general engineering fields, all of whom make use of composite materials to a greater or lesser extent. However, the particular problems associated with out–of–plane joints have received limited attention. This work has, in most cases, been directed towards solving a particular design problem, and not to developing a design method. Nevertheless it was found that, in the sphere of naval GRP vessels, certain test methods had been adopted as "standard" for bulkhead–to–shell joints (tee–joints) and top–hat stiffener–to–shell joints.

B. Theoretical modelling

Classical analytical methods cannot accommodate such complex geometries so it is necessary to utilise an iterative method. The finite element method was chosen as the most productive method by which to analyse composites since it readily allows geometric and material variations within three–dimensional models.

Initial studies considered the method itself. Models were developed of simple structures to enable comparisons to be made between the results given by different element formulations, and classical analytical methods. Good correlation was achieved.

Since no clear design parameters were highlighted by the background study, several models were developed representing, in a simplified form, two adjacent compartments of a GRP ship with a top–hat stiffened bulkhead between them. Some models included integral tank structures. These were loaded to represent extremes of hogging, sagging, dry docking, and wave loading. This study showed that, despite the large range of loading conditions, the global loadings on the region of the joint followed a few regular patterns. This showed that the "standard" test methods adopted in previous work were reasonably representative, for comparative studies, of onboard conditions.

It was felt that, if the "standard" test conditions were applied as boundary conditions to the detailed models, this would also be reasonably representative. This also had the additional advantage that the test results from previous work could be used as validatory data for the theoretical modelling. Thus the test boundary conditions were applied to detailed models of a range of different tee–joint designs.

At this stage a study was made to ascertain the effect of modelling parameters (such as element mesh density, iterations to convergence, small or large deflection analysis, linear and non–linear material properties). The results of this study enabled a standard of modelling to be maintained throughout the parametric study so that fair comparisons could be made between different designs. It became clear that the properties of many of the constitutive materials that make up the composites were not available so a test programme was completed to determine the most crucial values. Others could not be found because the physiological properties of the materials made the production of void–free test coupons extremely difficult. After consultation with the manufacturers, assumptions were made based upon the properties of similar materials.

The series of different designs were analysed using extra displacement functions to model large displacements, and with non–linear material properties. Multiple load steps were used to allow the finite element program to follow the non–linear material properties and to show the effect of load upon displacements and stresses. The results from the analyses showed, quantitatively, the stress levels in the different elements of each joint, and hence, in a qualitive sense (since values of failure stress were not available for some materials), the root and path of failure. One design in particular proved to have a clear advantage (in terms of low internal stresses at high applied loads) over the others considered. The mechanisms of the joint behaviour could be followed, and hence understood, using this modelling technique. This enabled the more important design variables to be determined.

Once the influential variables had been identified a parametric study was completed. The output from this study created a series of design curves which can now be used to determine the optimum joint geometry given the scantlings (and hence properties) of the elements being joined. When compared to the traditional tee–joint designs, the joint design developed during this program can withstand 2.5 times the maximum applied load, is 60% lighter, and considerably cheaper to manufacture.

A similar program to that described above was completed for top–hat stiffenner–to–panel joints. The optimised joint design was found to match the strength of the original designs, whilst reducing weight and allowing greater production flexibility.

The analysis was completed by developing a series of large scale models of stiffeners attached to plating with the same element definition as in the detailed models. These were limited in number since they were very large and complex but they illustrated that the "standard" test load and boundary conditions applied to the detailed models produced the same internal stress distributions as the more general (and more representative) loading applied to the large scale models. This confirmed the assumption made at the beginning of the analysis.

C. Experimental Programme.

An experimental program was undertaken in parallel with the theoretical work. This comprised four elements.

1. Testing of composite coupons to determine their fundamental material properties. As mentioned in B. above material properties for some materials were not available. The combination of an elastoplastic matrix with different reinforcements meant that traditional methods of calculating composite material properties (eg. rule of mixtures) could not be used. Simple tensile testing was used to determine the stress/strain characteristics and UTS.

2. The series of different joint design referred to in B. above were fabricated by Vosper Thornycroft. These were tested on a special jig inserted into a tensile test machine. Plots of load vs. displacement were produced for each sample tested, as well as video film of each sample failing. This was most useful as frequently final failure occured too rapidly to be seen by the naked eye and a frame–by–frame review of the video gave an additional perspective. A limited number of samples were tested with photo–elastic coatings attached to them. This allowed surface strain values to be recorded. Some problems were encountered, however, in finding a coating that could cope with the 100%+ strain to failure capacity of the test samples.

The test data was digitized and used as validating data for the theoretical analysis. Good correlation was found for samples in which the dominating structural components were fibre reinforced, but correlation was poor for samples in which the joint was made wholly, or substantially, from unreinforced fillet material. This was one material whose properties had been assumed (for reasons given in B. above). A closer examination was made of the assumptions used. These were modified in light of the known effects of additional additives in the material, and better correlation was achieved.

A further set of designs were tested to confirm the trends shown in the theoretical parametric study. These were tested in a similar manner as before, with output in terms of load/deflection curves.

3. To confirm the performance of the optimised joint design a test was conducted on a full scale tank, fabricated using the new design on one face. This is also one of the "standard" tests used in the past to achieve UKMoD approval for a new design. The new design withstood the maximum pressure the test rig could apply with only minor delaminations whilst extensive damage occurred elsewhere in the tank despite these areas being reinforced with steel supports.

4. The final stage of testing was to subject the optimised design to underwater shock loading. This was done by fabricating a false bulkhead into the bottom structure of HMS Inverness, which was undergoing the first–of–class shock trails. No damage occurred to the optimised design joint.

Unfortunately financial and time constraints prevented a similar experimental program being conducted for the top hat stiffener–to–shell joint.

D. Further Theoretical work.

During the course of the theoretical work it became clear that the general finite element codes being used, whilst fine for a comparitive study, were cumbersome and limited when applied as an analytical tool to a given composite structure. This is in no way a criticism of the codes, it is simply a result of their general nature. Certain aspects were highlighted as being limited and several new features were listed as being desirable. These are being incorporated into a new code specifically for the analysis of composite materials. This work is on–going and, as yet, has not been the subject of publication but a precis is included as Appendix 2.

Dissemination of Results

Several papers have been published or are in preparation describing the above work. These are listed below and included as Appendix 3.

1. Shenoi, R.A., Hawkins, G.L.: "Large Deflection Behaviour of GRP Panels with Attachments – a Preliminary Study". Department of Ship Science Report

No. 47, ISSN 01403818, April 1990.

2. Hawkins,G.L., Holness, J.W., Dodkins, A.R., Shenoi, R.A.: "The Strength of Bonded Tee–Joints in FRP Ships". Proceedings of Conference on Fibre Reinforced Composites, FRC 92, Plastics and Rubber Institute, Newcastle–Upon–Tyne, March 1992.

3. Shenoi, R.A., Hawkins, G.L.: "Influence of Material and Geometry Variations on the Behaviour of Bonded Tee Connections in FRP Ships". Composites, Volume 23, Number 5, September 1992.

4. Dodkins, A.R., Shenoi, R.A., Hawkins, G.L.: "Design of Joints and Attachments in FRP Ships' Structures". Proceedings of the Charles Smith Memorial Conference "Recent Developments in Structural Research", Defence Research Agency, Dunfermline, Scotland, July 1992.

5. Hawkins, G.L., Shenoi, R.A.: "The Practical Design of Joints and Attachments in Marine Structures". in "Composite Materials in Marine Structures", Volume 2, edited by Shenoi, R.A., and Wellicome, J.F., Cambridge University Press, 1993 (to appear).

6. Hawkins, G.L., Shenoi, R.A.: "A Parametric Study to Determine the Influence of Geometric Variations on the Performance of a Top–Hat Stiffener to Shell Plating Joint". Composite Structures (to be accepted).

7. Shenoi, R.A., Hawkins, G.L.: "A Parametric Study to Determine the Influence of Geometric Variations on the Performance of a Bulkhead to Shell Plating Joint". Journal of Composite Materials (to be accepted).

Progress and Expenditure

Item	Cost	Budget	Balance
Employees – Academic Related	35,139.99	33,445.00	–1,694.99
Employees – Technical	5,066.72	4,249.00	–817.72
Travel and Subsistence	2,224.35	1,300.00	–924.35
Equipment and Services	8,535.94	7,078.00	–1,457.94
Total Expenditure	50,967.00	46,072.00	–4,895.00
Total Income	49,224.93	46,072.00	3,152.93
Balance			–1,742.07

Appendix 1

Contract Proposal: Large Deflection Behaviour of GRP Panels with Attachments.

UNIVERSITY OF SOUTHAMPTON
DEPARTMENT OF SHIP SCIENCE
RESEARCH PROPOSAL

LARGE DEFLECTION BEHAVIOUR OF GRP PANELS WITH ATTACHMENTS

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 effects 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 problem to model and needs to be tackled in several stages. The proposed first stage in this context will be to study and model the behaviour of attachments to plating in a non-linear, static mode; this forms the basis of the research proposal.

Whilst it is appreciated that failure in a dynamic mode may involve mechanisms which are not included in the static tests and are not fully understood, it is considered that the proposed work programme will provide an essential first stage before extending the method in further studies covering dynamic and shock aspects.

2. Key Aims

The main aims of the project are:

1) to develop analytical procedures to model non-linear, large deflection behaviour of attachments to plating;

2) to validate the theoretical model through simple (static) laboratory experimentation;

3) to apply the analytical scheme in the practical design of attachments to GRP structure.

3. Tasks Involved

A. Background study

The first task is to identify sources of information and study the problem background with particular reference to :

- kind of failure mechanisms in the structure arising out of shockwave loading.
- developments in materials and production technologies relevant in this context.
- potential similarities in aerospace, civil and nuclear engineering industries.

B. Failure analysis modelling

This is to be developed in light of the above background study. The theoretical model is to account for two types of attachments - namely bulkhead boundary angles and top-hat stiffeners.

Input to the model is to include variations in:
- structure geometry,
- material compositions in plating and attachment connections,
- load directions and related particulars.

Output is to include:
- location of initial crack,
- magnitude of load for crack initiation,
- strain or energy levels associated with this,

The objective in this task would be to gain understanding of delamination within stiffener flanges or shell plating as well as in the bond line between two components.

C. Experimental and numerical verification

The theoretical model is to be validated in comparison with simple peel tests and static pull-off tests involving bulkhead boundary angle attachments as well as top-hat stiffeners. The tests are to involve variations in structure geometry, scantlings, materials and load patterns.

Further and more detailed study is to be done using existing finite element software. Modelling is to be done using anisotropic, multi-layer composite elements and involve large deflection beahaviour analysis.

D. Application to practical design

The techniques developed in (B) are to be applied in the design of typical bulkhead boundary angle connections to the shell and top-hat stiffened panels. The effects of changing geometries, materials, and other scantlings are to be investigated in a systematic (parametric) manner.

E. Report write-up

4. Project Administration and Industrial Co-operation

The work is to be undertaken on a contractual basis by the (Department of Ship Science) University of Southampton.

The project is to be of two years duration, to be carried out from April 1990 till March 1992.

The co-operating organisation for this research is Vosper Thornycroft (U.K.) Ltd.

5. Research Experience

Professor G.J.Goodrich (Supervisor) has over 40 years experience in naval architecture and ship design. He has been Professor of Ship Science since 1969.

Dr. R.A.Shenoi (Co-supervisor) has been a Lecturer of Ship Science since 1981. His current research interests include design/testing of ship's appendages made from composite materials, expert systems in structural design/production, structural design and materials engineering aspects of SWATH ships.

Additional expertise in the areas of composites and structural mechanics is available in the Institute of

Sound and Vibration Research (I.S.V.R.) at this University; it is anticipated that the project tasks would be carried out in consultation with Professor R.G.White and his team in I.S.V.R.

6. Resource Requirements

A. Staff

The work involved in this project is mainly theoretical in nature; it requires an understanding of structural mechanics, materials and numerical analysis. There is also an element of simple testing involved. Consequently, the required Research Assistant will have to be a person with a good first degree in engineering or mathematics and, preferably, have some industrial experience as well.

The testing programme would involve the fabrication of an experimental rig and setting-up of related equipment. This requires the services of a Technician for a period of about four months.

B. Travel

The work is to be carried out in close collaboration with the Ministry of Defence Procurement Executive (MOD) and the Admiralty Research Establishment (ARE). MOD is likely to be highly interested in the direction of this research and its results. Four, one-day trips at six month intervals are therefore planned. ARE has extensive experience in the design and testing of marine structures. Specific to this project is the experience of its staff in the area of GRP structures. Six visits to Dunfermline of one day duration each (at four month intervals) are planned.

It is also advantageous to keep abreast of the latest research thinking in other countries. Apart from forums provided by learned bodies (such as RINA), experts gather at established conferences such as BOSS and PRADS. (The venues and dates of these vis-a-vis the start of the work are not known as yet.) Some money has therefore been budgeted for this purpose.

C. Equipment and other costs

The project work involves extensive theoretical modelling and the use of some data and design techniques that are likely to be sensitive in nature. The work requires a dedicated computer work station with no public access. Furthermore, for experimentation purposes there is a need to design and fabricate the test rig and acquire load cells, strain gauges, dial gauges, etc.

D. Proposed breakdown

Research Assistant	(24 Months)	:	£ 30,326
Technician	(4 Months)	:	4,249
Travel		:	2,300
Equipment and consumables		:	7,100

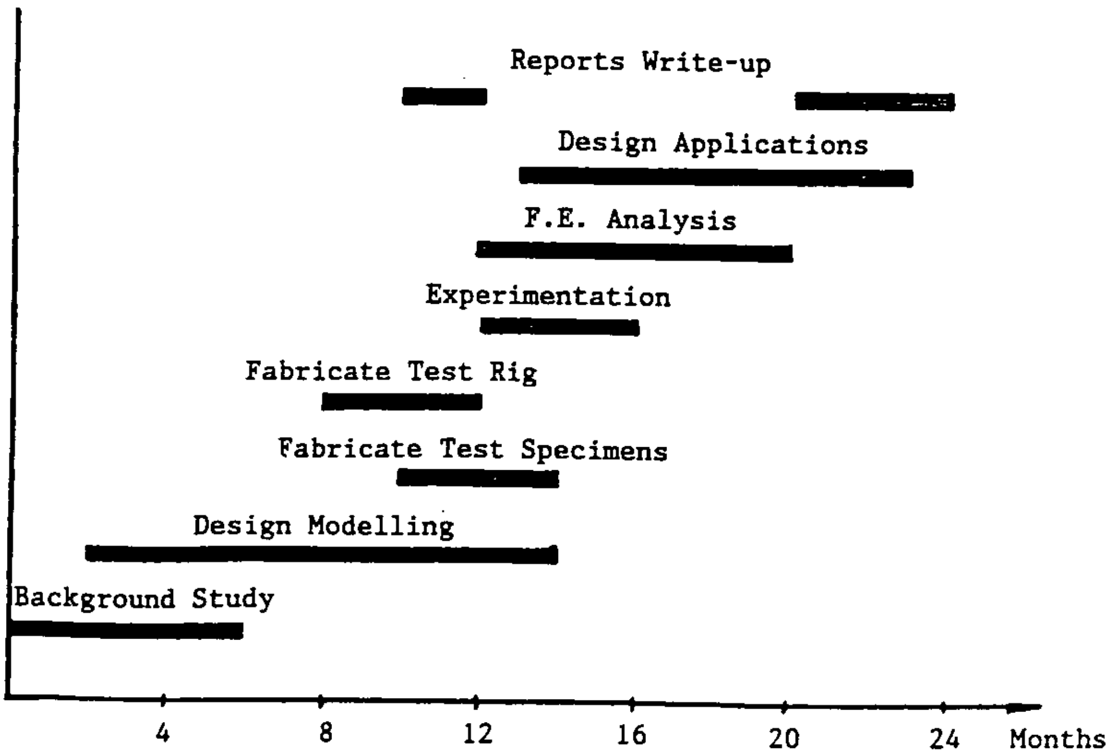
7. Likely Applications and Exploitation of Work

The results of the work will have an immediate application in the design of GRP structures and connections. The results will be used to gain an understanding of the failure mechanisms under non-linear behavioural conditions such as those involved in the shock load situation.

Exploitation of the results of the work will be carried out by the co-operating organisation, Vosper Thornycroft (U.K.) Ltd.

The philosophy of the investigation, techniques used and broad conclusions will be the subject of publication.

8. Time-Table for Tasks



Appendix 2

Precis of Further Theoretical Work.

Precis of Further Theoretical Work.

Limitations of General Finite Element Codes.	**Methods used to overcome these in Further Theoretical Work.**
1. Displacement functions within elements are linear or have extra displacement terms for large deflection analyses. This gives reasonable results for the overall element in the plane of the composite but is inadequate to model the through thickness behaviour if the layers have greatly differing properties.	**1.** Element displacement function remains standard in the plane of the composite but in the through thickness direction the displacement function becomes the sum of the displacement functions of each layer that lies between the point in question and a given node, summed for all nodes. This gives the correct displacement across the element since the scaling factor applied to each layer displacement function is derived by summing all the layer displacement functions through the thickness of each element and equating this to a standard result. In terms of the overall result this is the same as having one element per layer through the thickness but without the problems of node and element definition that this entails.
2. Three–dimensional solid elements, the type most commonly used to model complex geometries such as these since some stacking is required, lose some accuracy when the element is tapered. To model any curved surface requires some tapering and hence some approximation in the results.	**2.** By incorporating scaling factors into the element formulation the effects of tapering in all directions are accomodated, since the effective area and length can be calculated for any point within the element. Thus stress and strain values are more realistic. These scaling factors incorporate the conversion factors from element to global coordinates so require little additional computation.

3. Material properties are calculated using a simple rule of mixtures type of formulation. This results in each element being assigned "nominal" material averaged from the layer properties. This gives reasonable results for the overall element behaviour as long as the layer properties are reasonably similar. If these are different then the element material properties used will bear little resemblance to the "real" values. Similarly, when calculating the results, stresses and strains within the element are simply averages of the nodal values and thus are not representative of "real" values. This becomes very important when trying to determine the failure load, the root of failure and failure paths.

3. Element "material properties" are found by applying a unit strain to an element and calculating, based upon the layer material properties, the three dimensional stress state within the element. This is repeated for the other two orthogonal directions and these stress states are used as the element material properties. Although this sounds computationally complex it is not too bad since even in the most complex model there is rarely more than one element type per 500 elements. Obviously, when calculating the results the element strain values are converted back to the detailed stress states, and hence a more representative picture of the layer, and inter–layer, stresses is achieved.

4. When materials of different types are next to each other output values are averaged across the boundary. This effectively reduces stress values in the stiffer material and increases them in the more flexible one. Since material boundaries are very common sites of failure they need to be modelled accurately.

4. The concept of applying stress states in place of material properties overcomes this problem since nodal strain values are equated from adjacent elements and thus stresses can be discontinuous across element boundaries if the properties of the two adjacent materials are different.

5. In addition to the above mathematical limitations the general F.E. codes do not take into consideration such geometric factors as layers finishing within laminates, and their structures do not readily allow parametric analyses of material and geometric changes. The incorporation of "dummy" elements to model delamination cannot be

5. The problem of analysing the stresses around a layer that finishes within a laminate is overcome by a combination of the three methods described in **1.,2.,** and **3.** above, with an additional preprocessing algorithm to define where these exist so that the correct layers are linked and continued into the adjacent element. The models are solved by done in the middle of an analysis. Thus if the stress state within a model reaches a level at which delamination will occur the analysis needs to be stopped, the dummy elements activated, and the process restarted. However when using non–linear material properties it is necessary that the restart occurs with the model in its pre–delaminated stress state, then for relaxation to be allowed to take place, before the analysis proceeds.

effectively treating each element as a substructure which is reanalysed each time the solver passes over that element. Each element has "dummy nodes" between each layer within it by default, these connecting with the adjacent "dummy node" in the next element. The preprocessing algorithm mentioned above simply allows this default connecting to be overidden to define the end of a particular layer. Additionally, each node or "dummy node" can divide if a delamination criteria is reached. The properties assigned to this division are material dependant and can be based upon "spring stiffness" or energy release rates. Stress values are assessed at each pass (or as often as defined) and if delamination has occurred the area is resolved to allow relaxation (or propagation if the crack is unstable). Since failure by delamination is incorporated within the analysis the conditions applied around the delamination are more representative of real life.

One of the main benefits of this somewhat complicated element formulation is that model definition is greatly simplified since it is not necessary to use more than one element through the thickness of any laminate or sandwich panel. With simpler model definition parametric studies can easily become "hands off" by having a control file to run the process.

Appendix 3

Publications Arising from the Contract.

LARGE DEFLECTION BEHAVIOUR OF GRP PANELS WITH ATTACHMENTS - A PRELIMINARY STUDY

by

R.A. Shenoi

G.L. Hawkins

April 1991

Department of Ship Science

University of Southampton

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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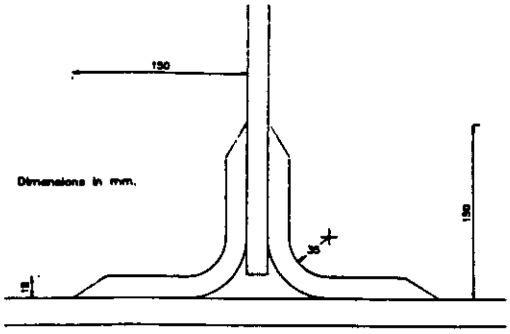

Figure 1: A Typical Shell/Bulkhead Connection

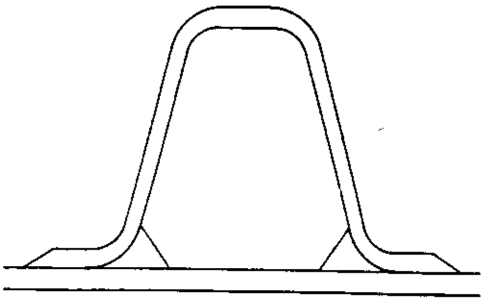

Figure 2: A Typical Top-Hat Stiffener

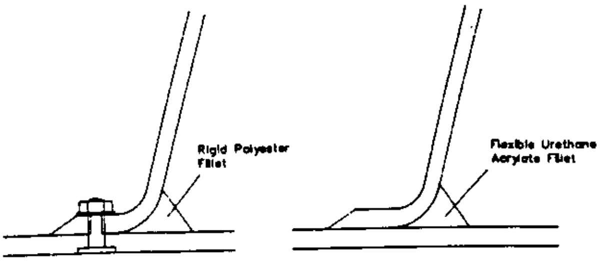

Figure 3: Alternative Boundary Angle Attachments

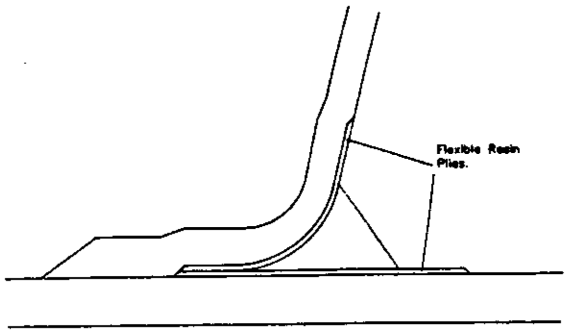

Figure 4: Incorporation of Flexible Resin Plies in Boundary Angle

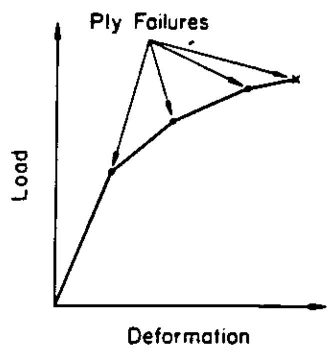

Figure 5: Variation in Stiffness Characteristics after Successive Ply Failures

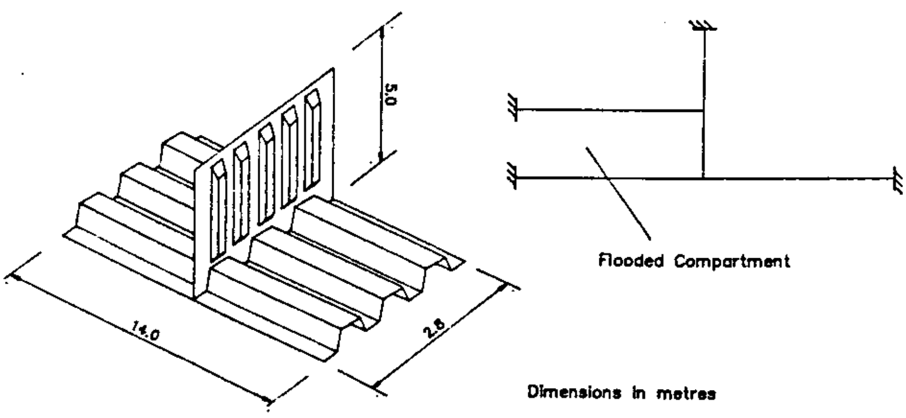

Figure 6: A Simple 2D Framework Model

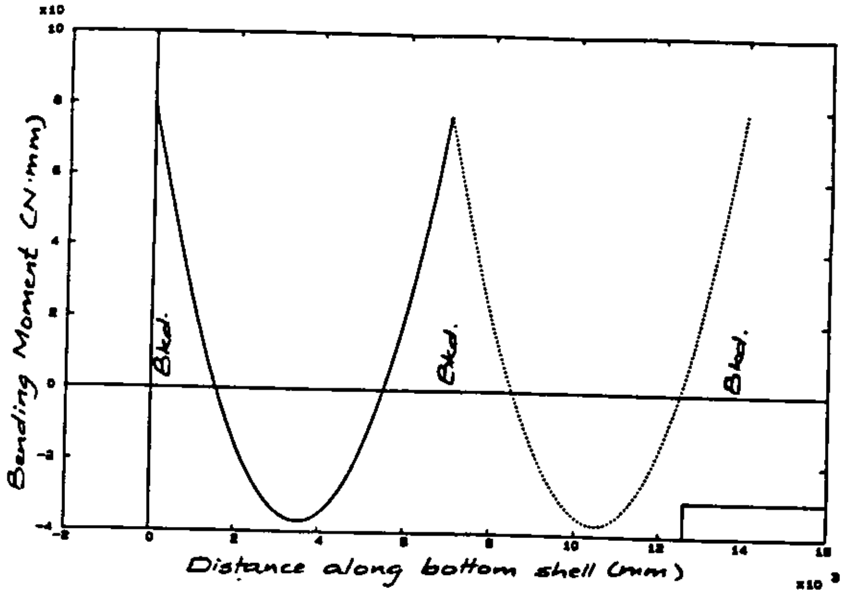

Figure 7a: B.M. Diagram - Shock Load

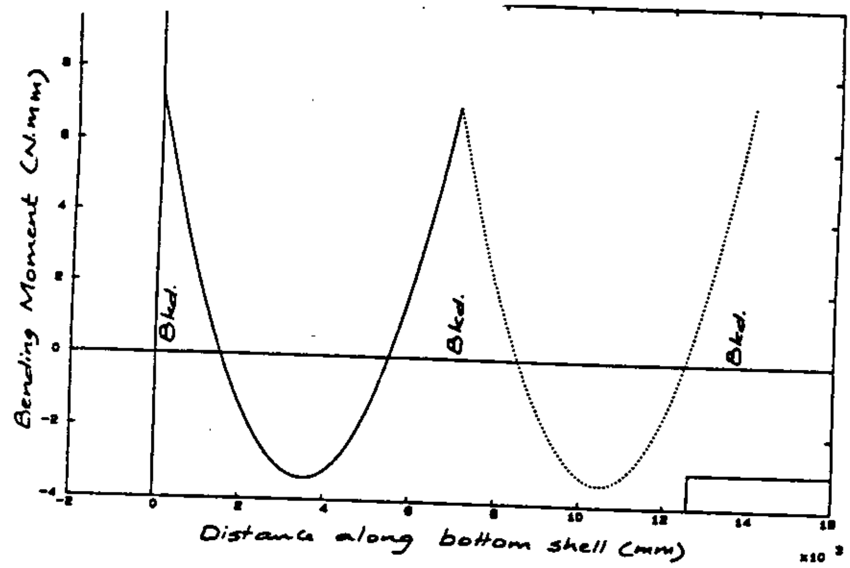

Figure 7b: B.M. Diagram - L/10 Wave

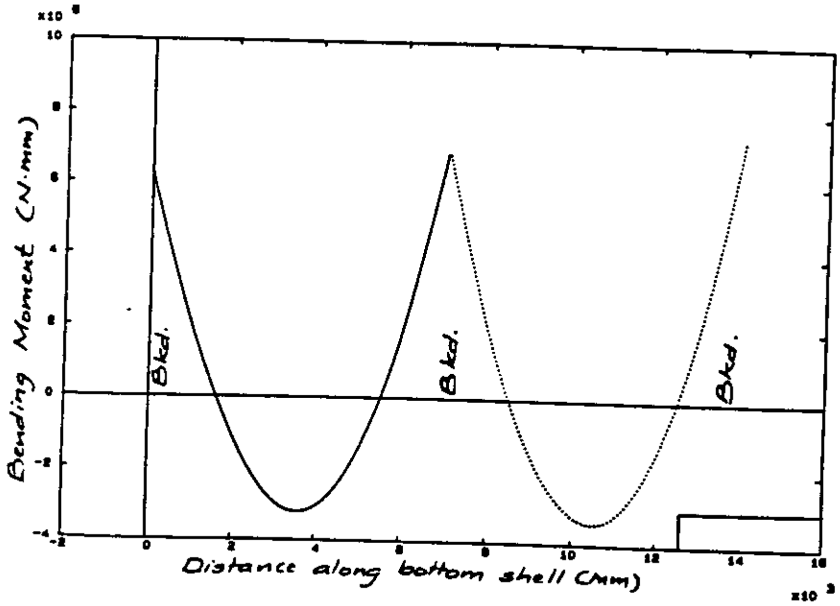

Figure 7c: B.M. Diagram - L/10 Wave and Compartments Flooded

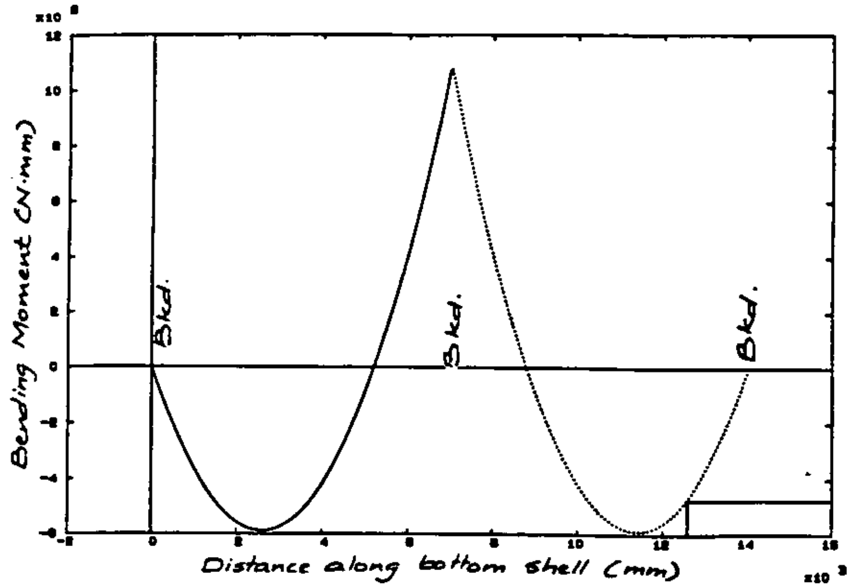

Figure 7d: B.M. Diagram - L/10 Wave with Attachments at Remote Bulkheads Assumed Pin-Jointed

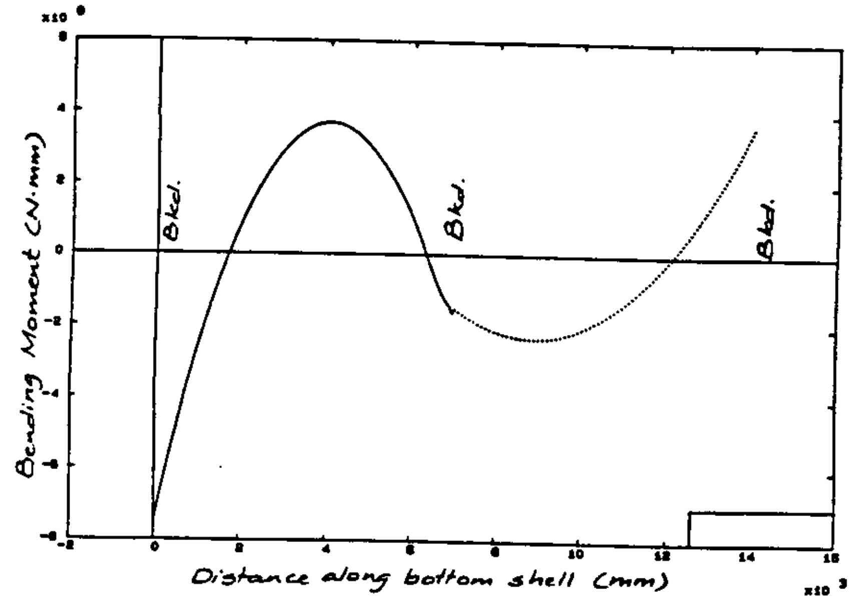

Figure 7e: B.M. Diagram - Design Draft and Tank Flooded

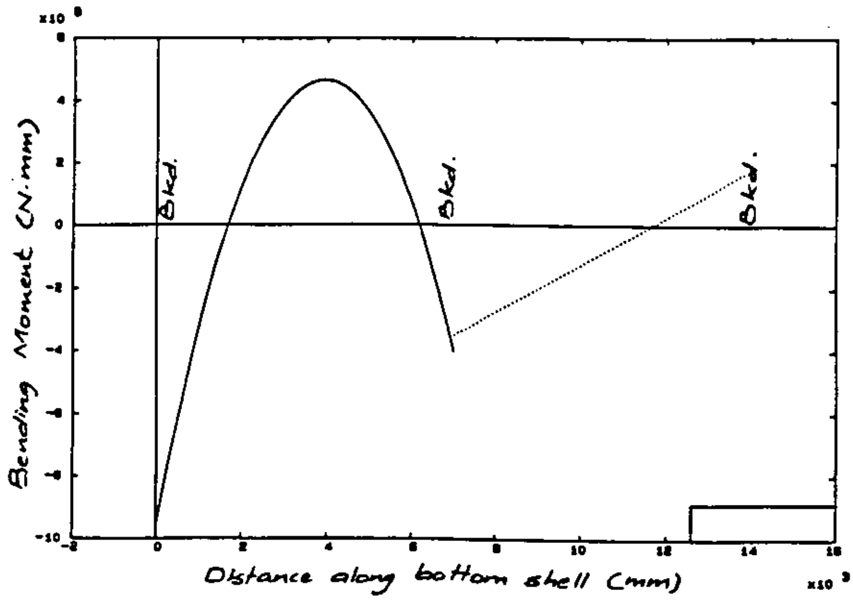

Figure 7f: B.M. Diagram - Drydock Condition

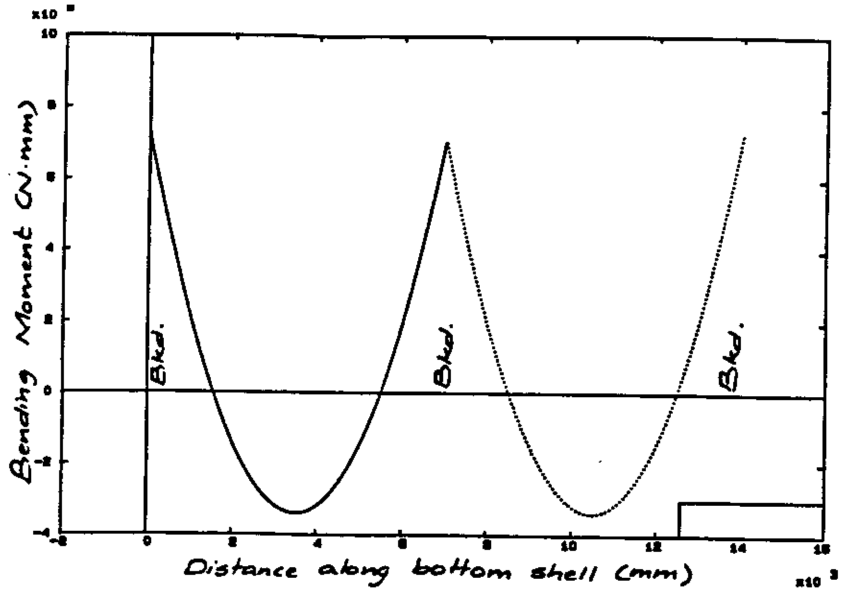

Figure 7g: B.M. Diagram - L/10 Wave with Low Modulus at Attachments

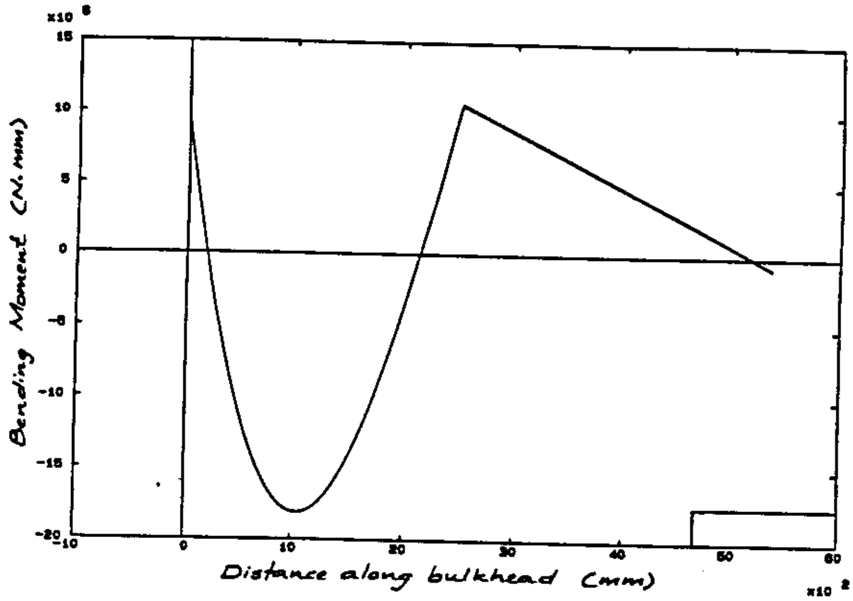

Figure 8a: B.M. Along Bulkhead - 1/10 Wave and Compartment Flooded

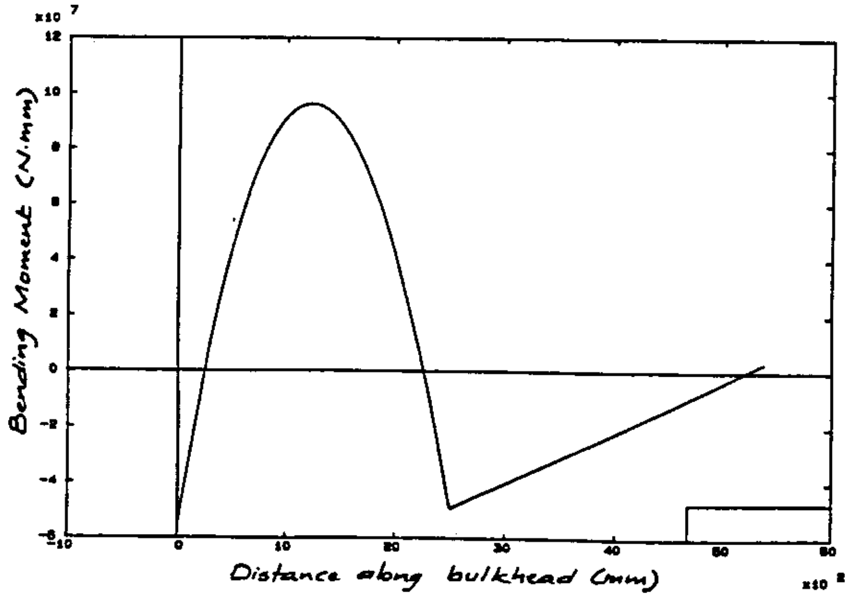

Figure 8b: B.M. Along Bulkhead - Drydock Condition

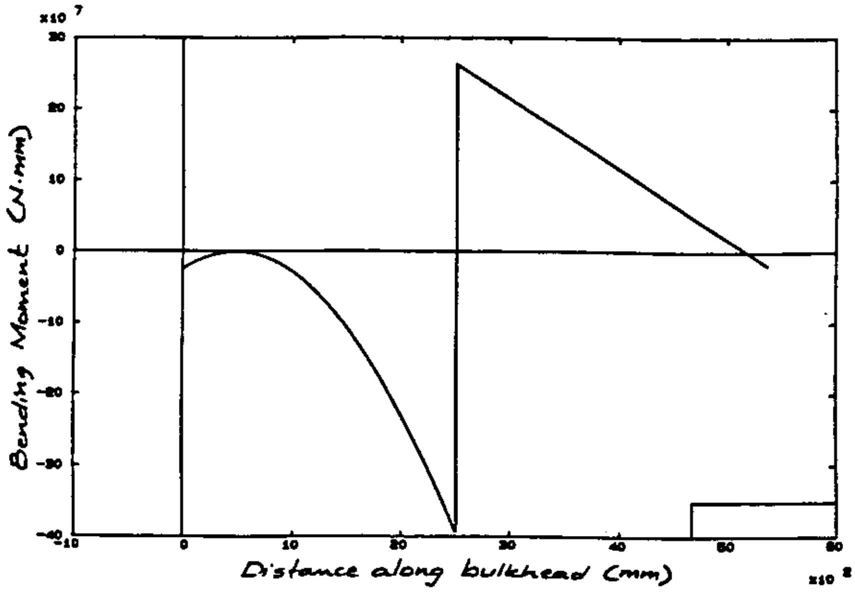

Figure 8c: B.M. Along Bulkhead - Design Draft and Tank Flooded

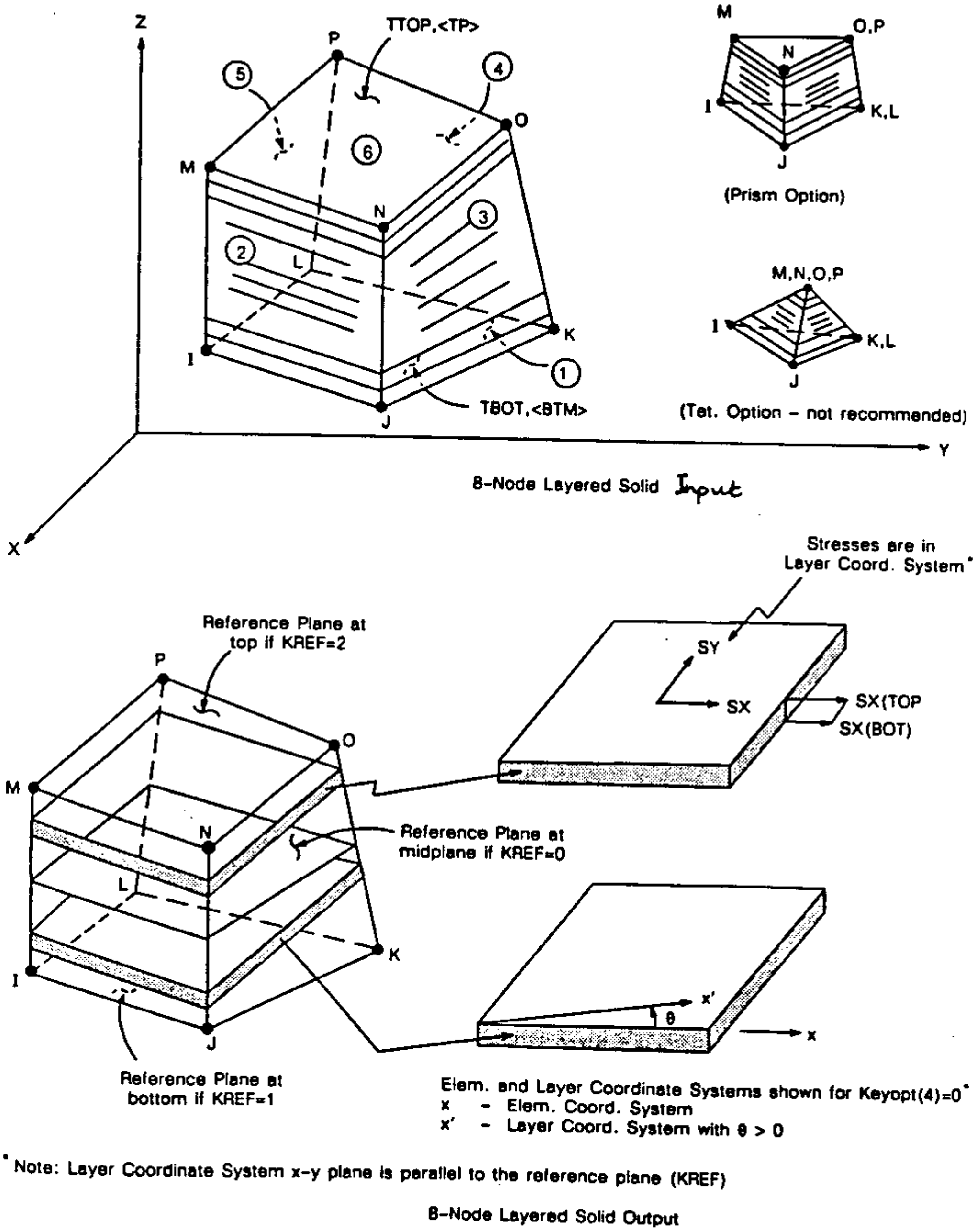

Figure 9: STIF46 Characteristics

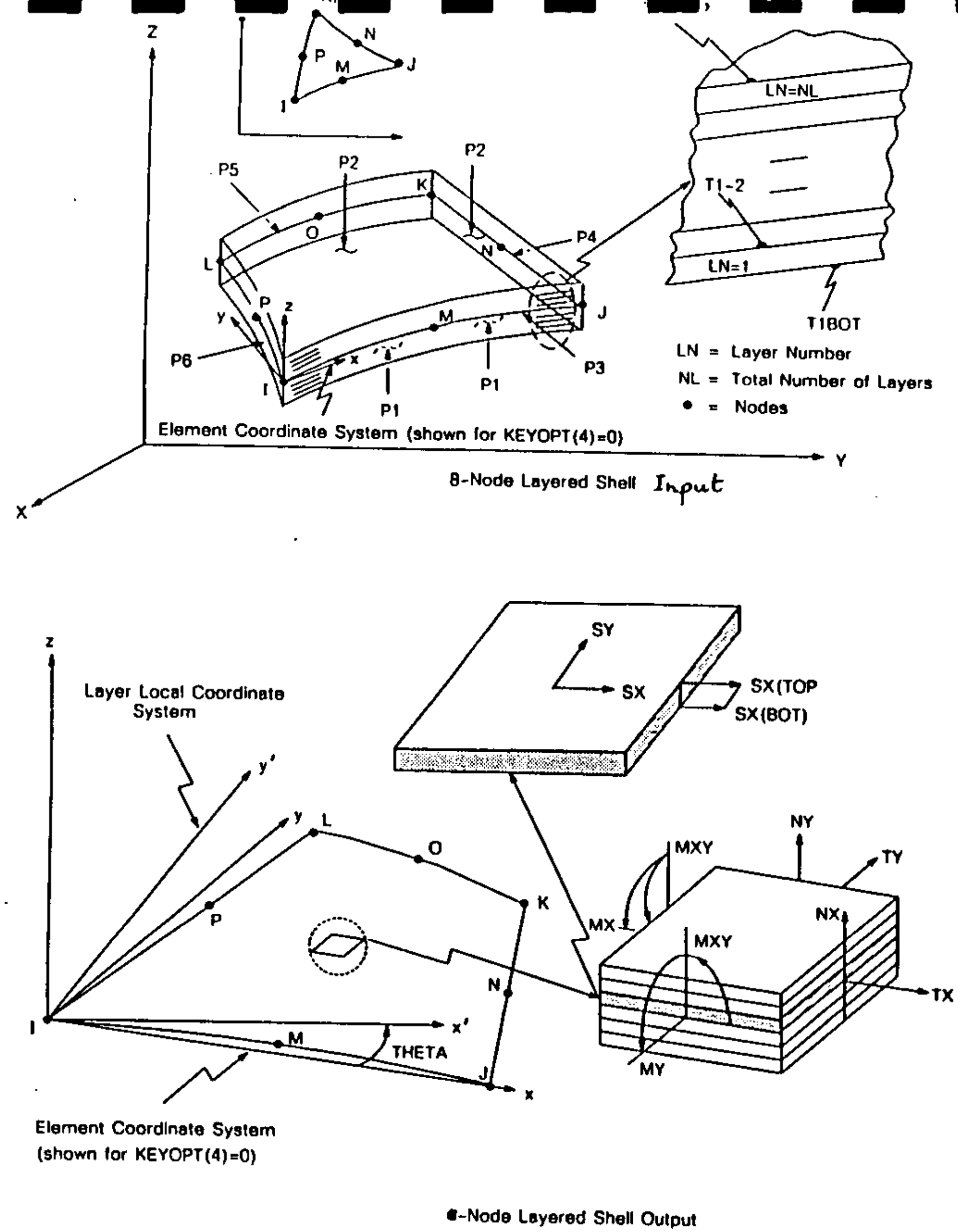

Figure 10: STIF91 Characteristics

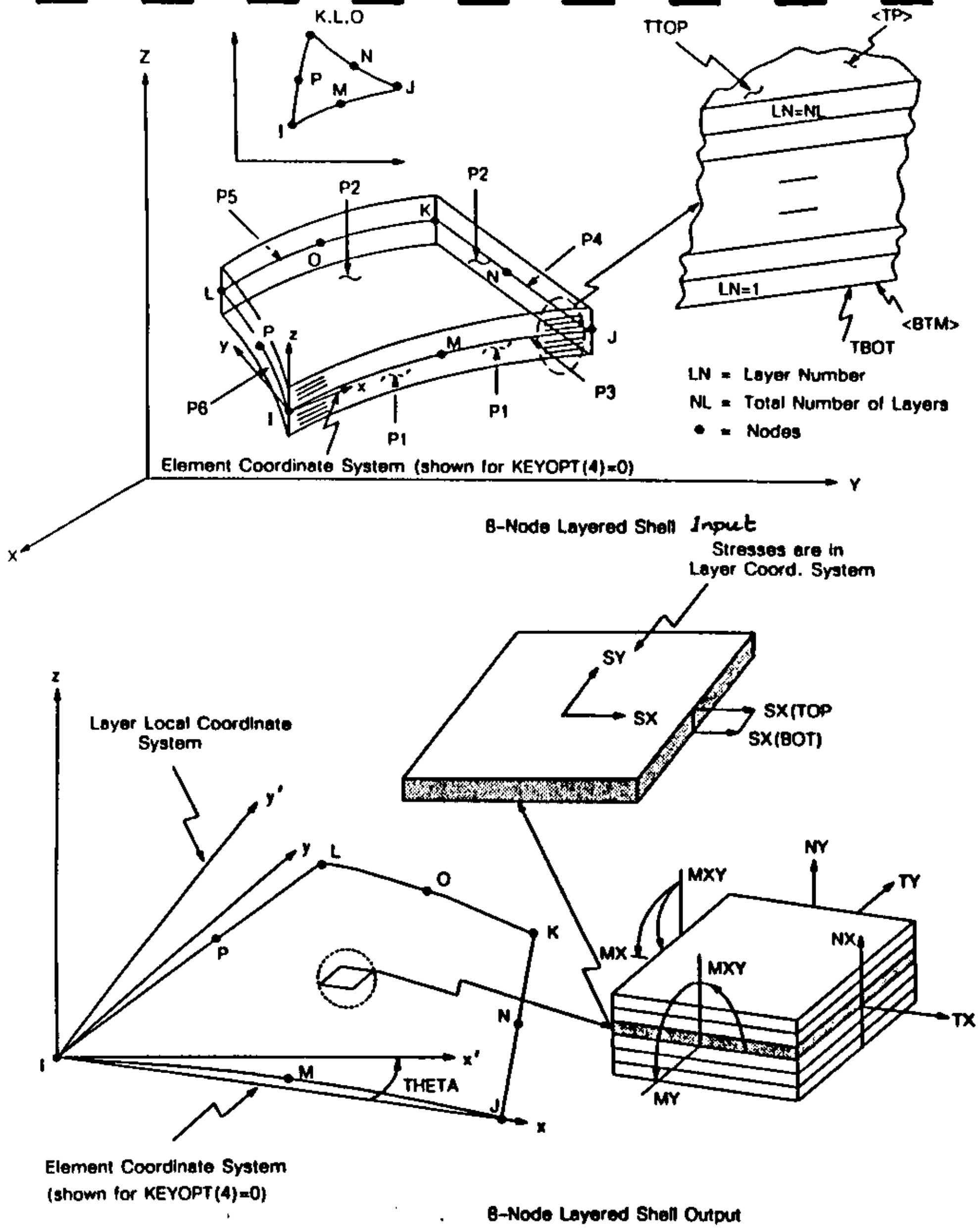

Figure 11: STIF99 Characteristics

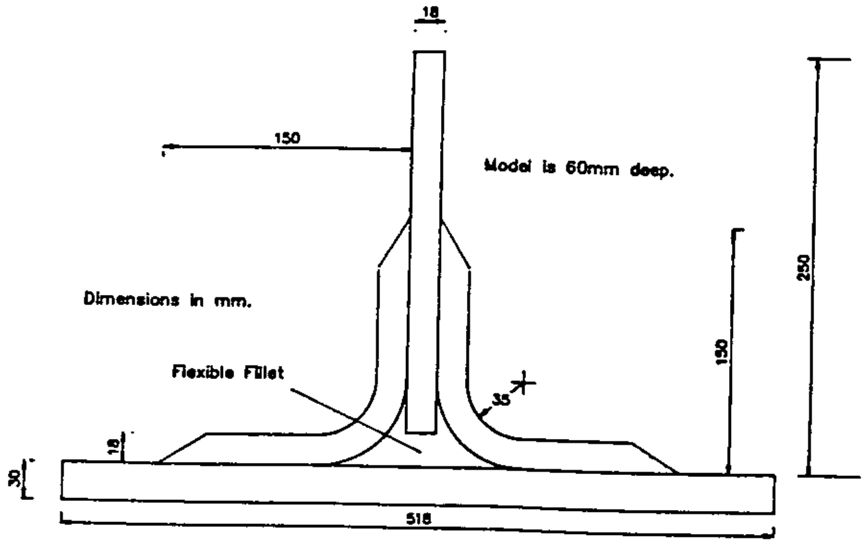

Figure 12: Geometry of the Preliminary Hull-Bulkhead Model

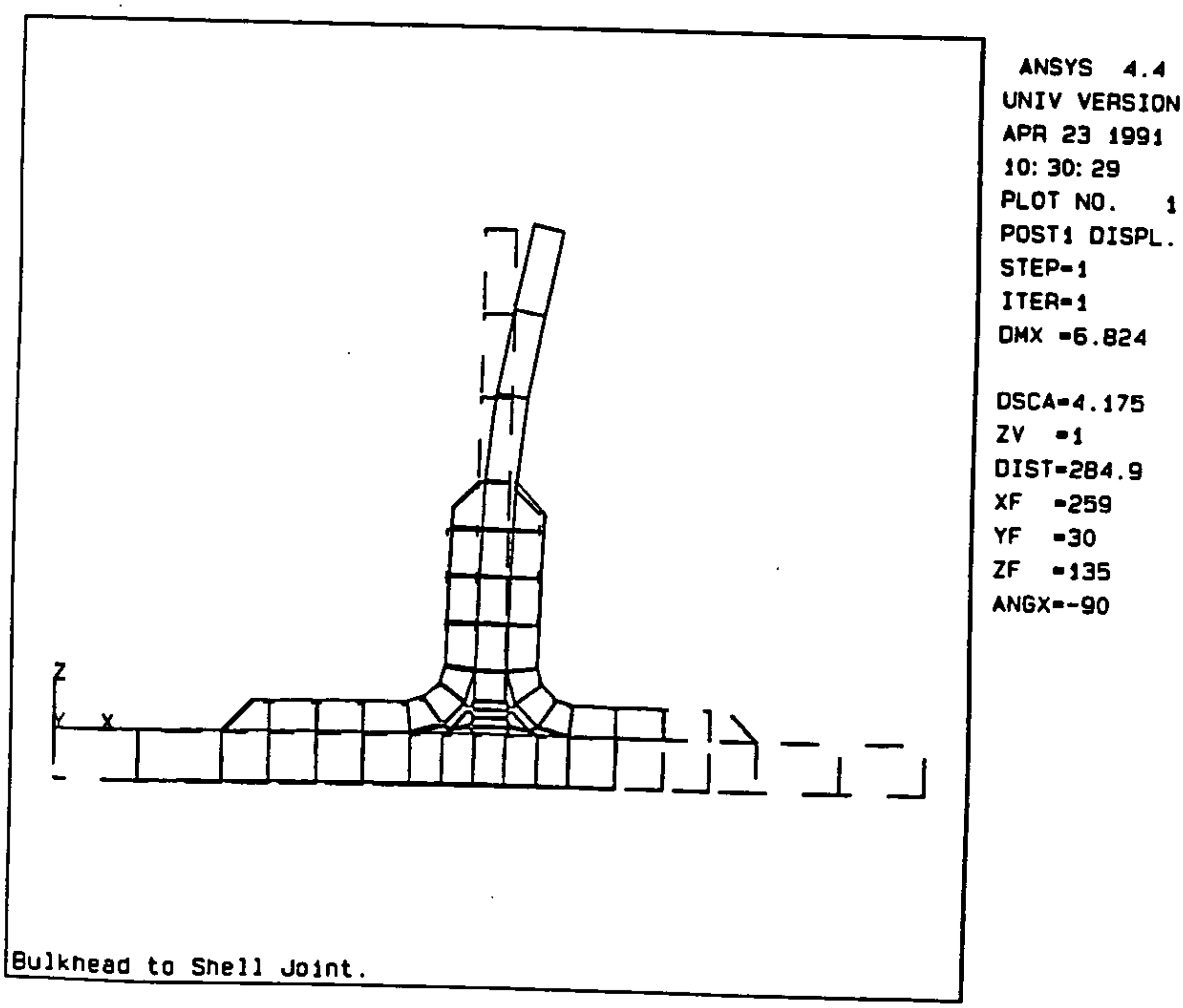

Figure 13: A Typical Displaced Contour of Joint

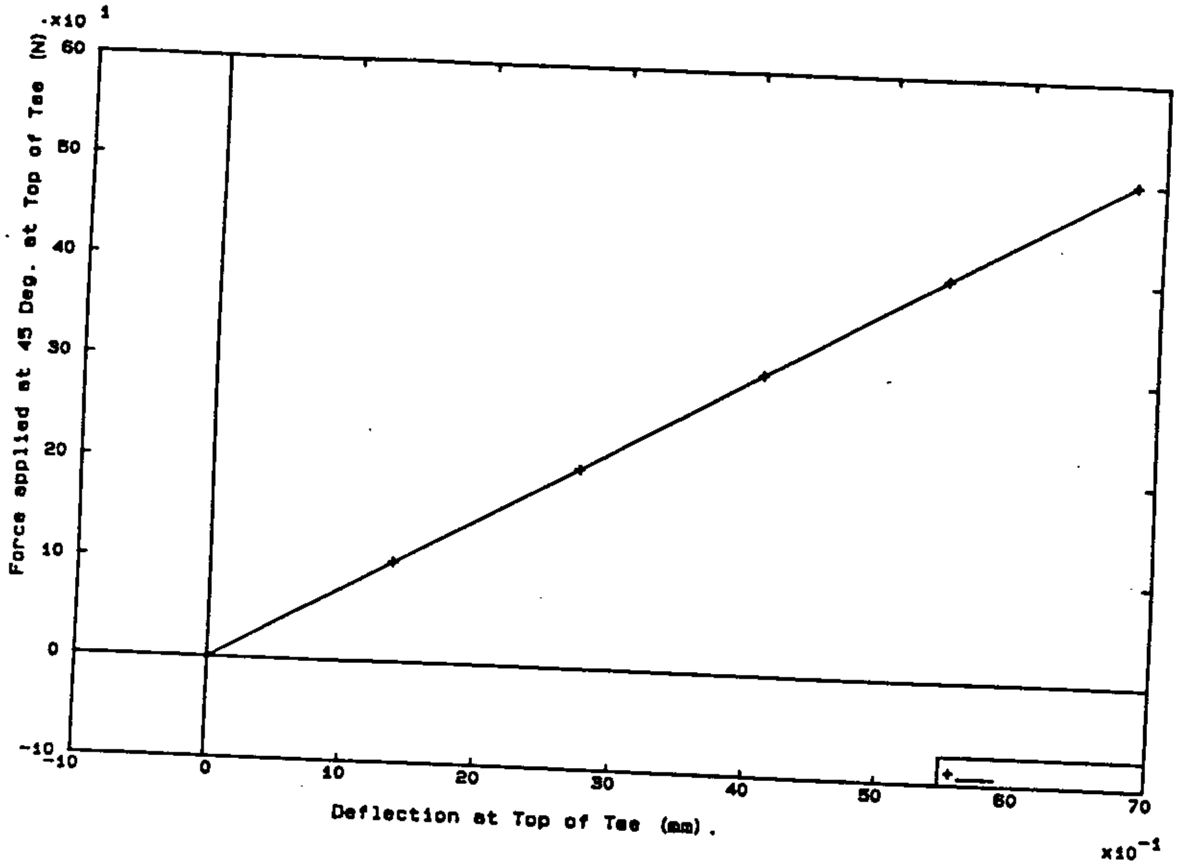

Figure 14: Load-Displacement Relationship

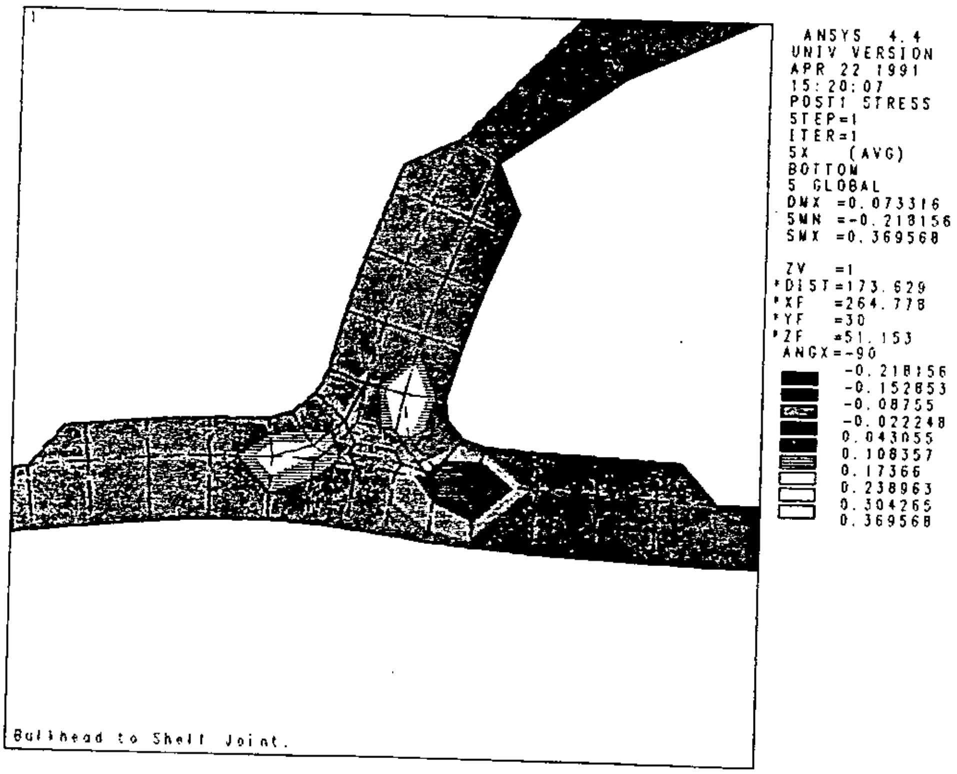

Figure 15: A Typical Stress Contour Plot

THE STRENGTH OF BONDED TEE-JOINTS IN FRP SHIPS

G.L. Hawkins*, J.W. Holness#, A.R. Dodkins#, R.A. Shenoi*

This paper is concerned with the design of boundary angles (tee-joints) in single skin FRP ships and boats. The paper addresses two principal areas. Firstly, there is a brief review of the problem background and of methods used to synthesise the layup in typical marine joints. Designs are then generated on the basis of these methods with considerations being given to different types of materials and configurations. The second aspect of the paper covers an experimental programme completed on a range of joint configurations. The results of these tests are presented in terms of loadings and deflections. Finally, some conclusions are drawn regarding the efficiency of the different configurations and how these reflect back to the current design methods.

1. INTRODUCTION

Through the design of three successive classes of mine countermeasures vessels (MCMVs) in the UK, much progress has been made with regard to overall FRP hull structural design. However, apart from some testing and analysis in the early days, relatively little attention has been paid to the design of joint details. Whilst the resulting joints have fully met the stringent design requirements of static and underwater shock loadings, there is much scope for gaining a better understanding of joint behaviour and for optimisation of joint configurations.

A major proportion of joints in FRP ships are of the bonded variety and are accounted for in the connections between two orthogonally placed plate panels such as deck-to-bulkhead, shell-to-bulkhead and floor-to-tanktop. The current solution, universally used from large MCMVs down to small craft, is to form such joints by laminating strips of reinforcing cloth either side of the joint to form a double angle connection - a "boundary angle" - as illustrated in Figure 1.

When compared to welded joints on steel ships boundary angles or Tee-joints 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 MCMVs, boundary angles account for about 10% of total structural weight whereas 2.5% is typically allowed for all weld metal in steel ship construction. Hence they provide a focus for potential weight and cost savings.

The purpose of this paper is to discuss key aspects of a currently ongoing research programme (1). Specifically, attention will be focused on the nature of boundary angles and their design features, the materials used, an experimentation programme and discussion of the results from a series of tests.

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# Vosper Thornycroft

2. DESIGN FEATURES

The primary function of a Tee-joint is to transmit flexural, tensile and shear loads between the two sets of panels meeting at the joint. This load transmission is done entirely through the cloth/resin plies and any filleting resin in the boundary angle. The design of the joint therefore needs to be such as to ensure that it is strong enough to resist these loads.

Strength (or weakness) is mainly dependent upon these failure mechanisms. Firstly, the cloth/resin plies attached to the base laminates (i.e. the "web/flange" of the Tee) could peel off as a result of flexural or tensile loads. Secondly, the boundary angle plies on the "web" of the Tee could separate from the base laminate through shearing. Thirdly, under flexural loading, the boundary angle plies could separate from one another through interlaminar tearing.

The design procedure therefore needs to account for the load and failure mechanisms. Currently, design synthesis is performed primarily through reference to regulatory agency rules (2-5). These procedures are dependent mainly on empirical evidence and do not lend themselves to rigorous analysis. Four other reasons why an analytical approach has not been attempted are explained below.

■ There is a lack of continuity (of reinforcing fibres) across the joint since, from a practical viewpoint, the boundary angles can only be fabricated following cure of the parent laminates of the panels to be joined.

■ The strength of the connection between the boundary angles and the base laminate is dependent upon the relatively low tensile and shear strength of the laminating resin rather than the relatively good in-plane strength properties of the laminates - see also Section 3.

■ FRP has a far higher resistance to brittle fracture than unreinforced resin; each fibre acts as a crack arrestor at a microscopic level. However, as there are no continuous fibres extending through successive plies, cracks between plies (or delaminations) can propagate rapidly at a fraction of the load which caused initial failure.

■ The tendency of the boundary angles to peel and delaminate under load emphasises the irregularities and imperfections of the laminate, compared with ductile characteristic of homogenous metallic materials.

The approach adopted in this study has been to explicitly identify variables which define the Tee-joint design in order to account for some of the above points. Figure 2 identifies the main variables. The overall scope of the research includes the development of a theoretical tool allowing investigation of variations in the parameters and load configurations. This work is currently in progress and will be reported in a separate forum.

This paper concentrates on the material aspects and the outcome of a test programme to study variations in the design parameters.

3. MATERIALS ASPECTS

Material based variations could be relevant for the base laminates and for the boundary angle configurations. The lay-up and stacking details of the base laminate are mostly dependent upon gross structural design criteria and hence have not been the subject of variance. In this study, all base plates have been cut from GRP laminate consisting of woven rovings in a matrix of polyester resin. Table 1 lists the main engineering properties of this layup, and those of other laminates used for these test specimens.

With reference to the boundary angle configuration, the primary feature of influence is the compatibility (of properties) between the cloth/resin plies and the resin itself. The cloth/resin plies, i.e. the GRP laminates, have a low modulus and are relatively flexible. However, polyester laminating resin is an essentially brittle material and may be subjected to a stress raising effect at the interfaces with reinforcing fibres. A consequence of this is that the through thickness tensile strength of a laminate is low compared to the resin strength. The respective values are 10 MPa and 50-70 MPa, with the latter being associated with a strain to failure of about 3%. (6)

It can be seen that this may cause difficulties when designing a joint that may be subjected unavoidably to a through thickness tensile stress, especially as a GRP boundary angle is not normally flexible enough to alleviate peel loads. The situation may be greatly improved by use of compliant resins. These are urethane acrylate resins in a styrene monomer which are chemically compatible with polyester resins and may be blended with them. The ultimate strengths are lower than those of polyester resin but the elongation to failure is much greater - see Table 2.

4. EXPERIMENTATION PROGRAMME

Eleven joint configurations have been considered to study the effect of varying geometry and resins. These variables considered are:

- Fillet radius,
- No. of plies in boundary angle,
- Material make-up of boundary angle plies,
- Edge gap between the web and flange of the Tee,
- Shape of the edge of the web,

The variations have been compared with "standard", NES 140 prescribed, specimens. The details of the specimens are outlined in Figure 3 and Table 3.

The test specimens were subjected to a displacement controlled pull-off of the web at 45° to the base (flange) plate and restrained as shown in Figure 4. 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. Loads and deflections were systematically recorded.

5. TEST RESULTS

The load deflection curves for the samples are shown in Figure 5.

Sample A failed initially by delamination within the boundary angle corner on the tension side - see Figure 6a. This delamination occurred within the third ply of the fillet. The load carried by the joint dropped away almost to zero. 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 B failed in a similar manner to Sample A. The delamination occurred in a similar region and, as before, the stiffness of the joint and its ability to withstand load seemed little altered by this initial failure.

Sample C failed catastrophically in the fillet. The point of initiation of failure was impossible to judge due to the speed at which it occurred. Figure 6b shows the position of the cracks in the fillet.

Sample D also failed catastrophically in the fillet. However, this was preceded by large cracks appearing between the fillet and the Tee. As these cracks developed, the stiffness of the joint was reduced and large deflections occurred. At a deflection of about 45mm, the cracks ceased to propagate and the stiffness increased back to the original value. At this stage one of four samples failed completely while three others failed at significantly higher (75%) loads.

Sample E failed in a manner similar to Sample C i.e. by catastrophic failure in the fillet. This was not preceded by any significant cracks appearing in the fillet although the load-deflection curve does show a gradual reduction of stiffness towards failure.

Sample F failed catastrophically by complete delamiantion of the overlay on the face in tension and splitting of the fillet - see Figure 6c. These two mechanisms occurred too rapidly to distinguish their order. These samples showed an almost constant stiffness up to failure and withstood the highest loads of all samples tested.

Sample G also failed catastrophically with pieces flying away from the destroyed fillet. Some creaking could be heard at relatively low loads (@ 50% of failure) but there was no visible damage at this stage.

Samples J, K. L and M failed in a similar manner to F. Samples showed approximately linear load-deflection curves.

6. DISCUSSION

The effects of the geometrical and material variations on the properties of the joint can be summarised as follows.

- Increasing the gap between the tip of the web and the flange piece allowed much greater deflections for a given load (@ 100%) and a greater load potential (@ 50%). hence the joint can absorb more energy prior to failure.
- Bevelling the tips of the web piece reduced the joint stiffness (by less than 10%) but increased the failure load by over 50%.
- Overlaminating increased the stiffness by 50% and failure load by 100%.
- Increasing the fillet radius made very little difference to the stiffness but increased the failure load by 150%. This is consistent with previous studies (7,8).
- Increasing the overlamination led to a slight increase in the stiffness and strength (@ 4%) but reduced the deflection to failure so that the energy absorbed by the joint at failure was reduced by about 5%.
- Changing the overlaminating resin from Crestomer 1200 to polyester did not alter the stiffness but reduced the deflection and load to failure by about 15% due to the reduced resilience of the resin.
- Overlaminations greater than 3 layers had no effect on joint strength, and was detrimental in that stiffness was increased, and thus energy absorption was reduced.

7. CONCLUDING REMARKS

The "correct" design of the Tee-joints in FRP ships is very important with a view to improving structural performance and efficiency. Current design methods do not have explicit ways of modelling variations in parameters and examining their effect on performance. This is one of the major objectives of an ongoing research programme being conducted by the authors. The first phase of this programme has been directed towards experimentally identifying the features exercising most influence over joint performance.

The results have shown that increasing the thickness of the overlamination, traditionally the criterion used for the design of these joints, has a detrimental effect on the properties of the joint. It has been found that the radius of the fillet, traditionally gives little or no consideration by current design methods, is critical to the performance of the joint. In general increasing the fillet radius enables the joint to withstand higher loads.

This experimental programme has been successful. The joints developed in the course of this phase, particularly configurations, incorporating a large fillet radius (urethane acrylate resin) and thin laminate overlay, have shown an increase in load to failure and deflection at failure of 50% in comparison with "traditional" boundary angles. Further theoretical work to corroborate these findings is underway and will be reported separately.

8. ACKNOWLEDGEMENTS

The work outlined in this paper has been funded jointly by Vosper Thornycroft (UK) Ltd and SERC, administered through MTD Ltd.

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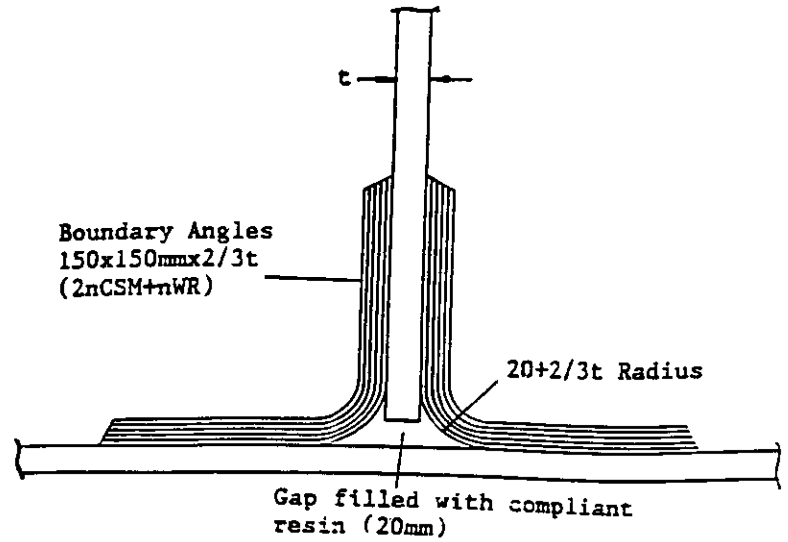
6. J. Bird and R.C. Allan. "The Determination of the Interlaminar Tensile Strength of Ship Type Laminates". Proceedings of the 7th International Conference on Experimental Stress Analysis, Haifa, Israel, 1982.

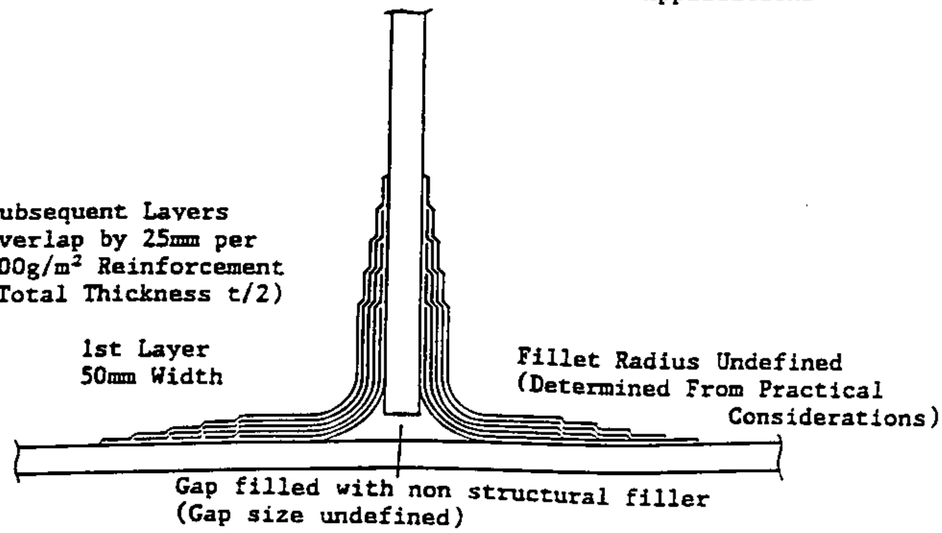
7. "GRP Boundary Angles Strength Tests", Report No. D/89-452, Vosper Thornycroft, 1989. (Restricted)

8. "Full Scale Tests on GRP Boundary Angle", Report No. D/89-455, Vosper Thornycroft, 1989. (Restricted).

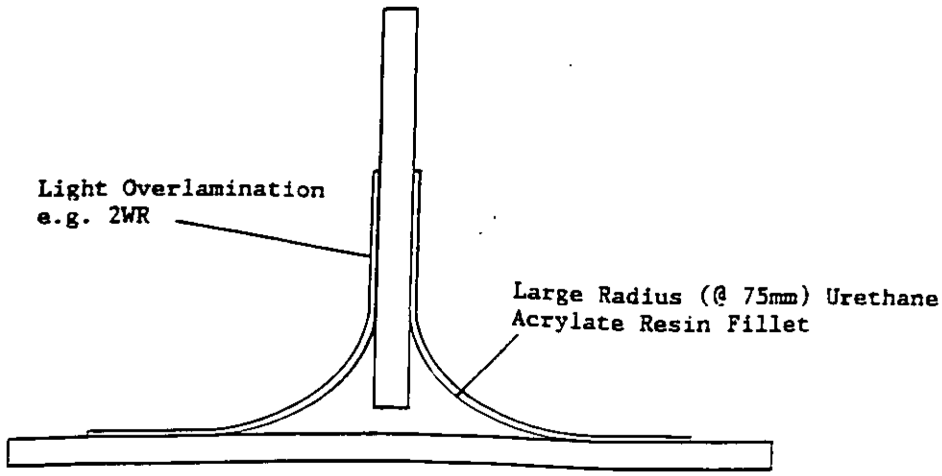
Figure 1: Typical Tee-Joint Connections



a) Typical Tee Joint Connection Developed for MCMV Applications



b) Typical Tee Joint Connection to Lloyds Standards for Commercial Applications



c) Proposed Tee Joint Connection

Figure 2: Tee Joint Design Variables

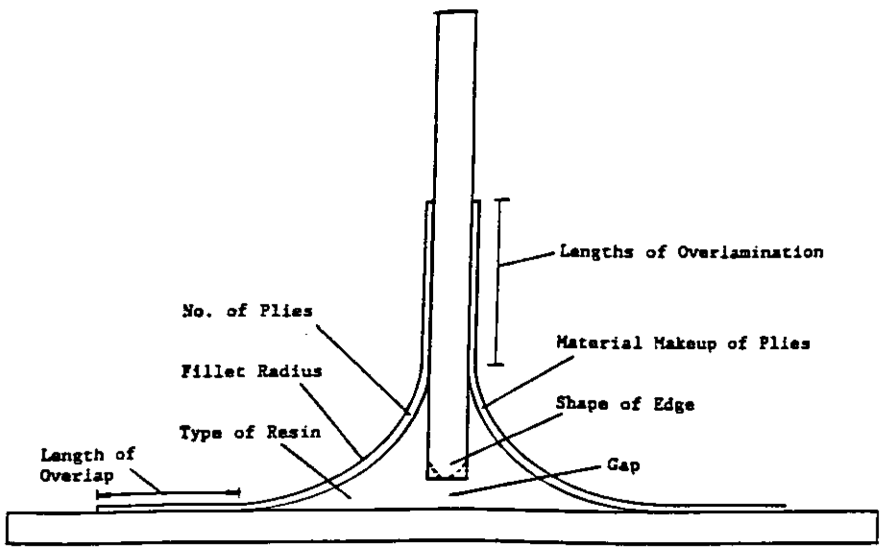


Figure 3: Experimental Programme

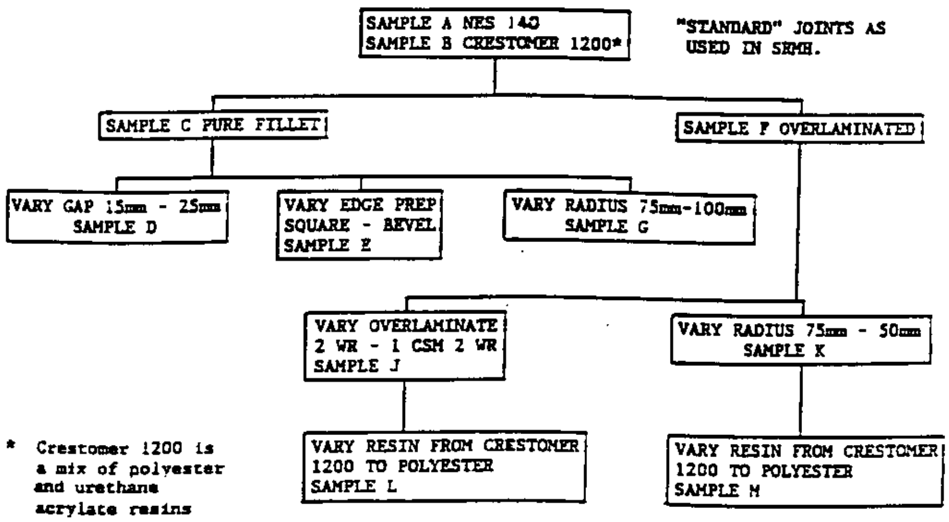


Figure 4: Boundary Conditions and Method of Loading

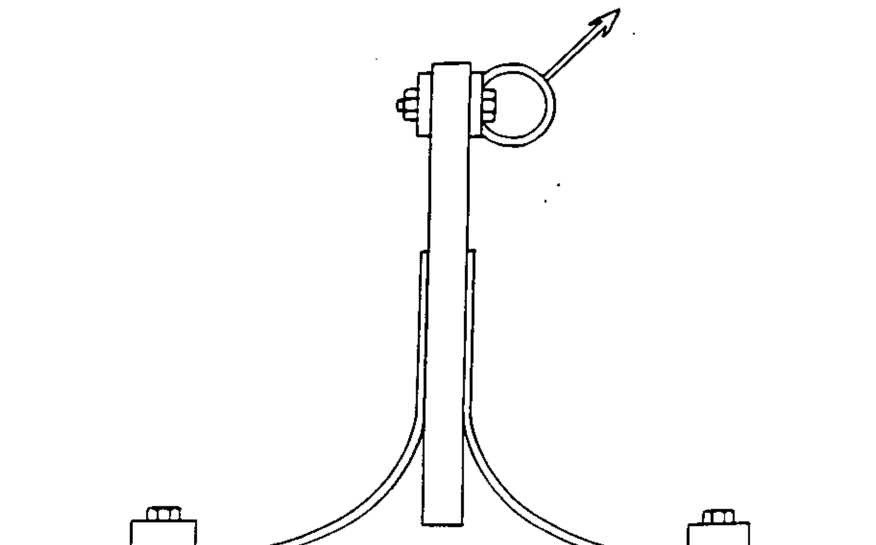


Table 1: Typical Laminate Material Properties.

Property	Tensile Strength (MPa)	Tensile Modulus (GPa)	Resin Ratio (Weight)	Poissons Ratio
W.R/Polyester				
Warp	238	13.22	45.5	0.178
Weft	175	12.33	45.5	0.152
CSM/Polyester	96.5	6.89	70.0	0.13
WR/CR1200				
Warp	183.3	6.37	45.5	-
Weft	188.9	3.93	45.5	-
CSM/CR1200	110.4	3.02	70.0	-

Table 2: Typical Cast Resin Properties.

Property	Tensile Stregth (MPa)	Tensile Modulus (GPa)	Elongation at Break %	Hardness
Urethane Acrylate	26.0	0.5 (initial)	100	65 (Shore D)
Polyester	75.0	3.5	3.5	42 (Barcol)

Table 3: Details of Test Specimens.

Specimen	Boundary Angle Thickness (mm)	Fillet Radius (mm)	Fillet Overlay	Resin for B.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

Figure 5: Comparison of Test Samples

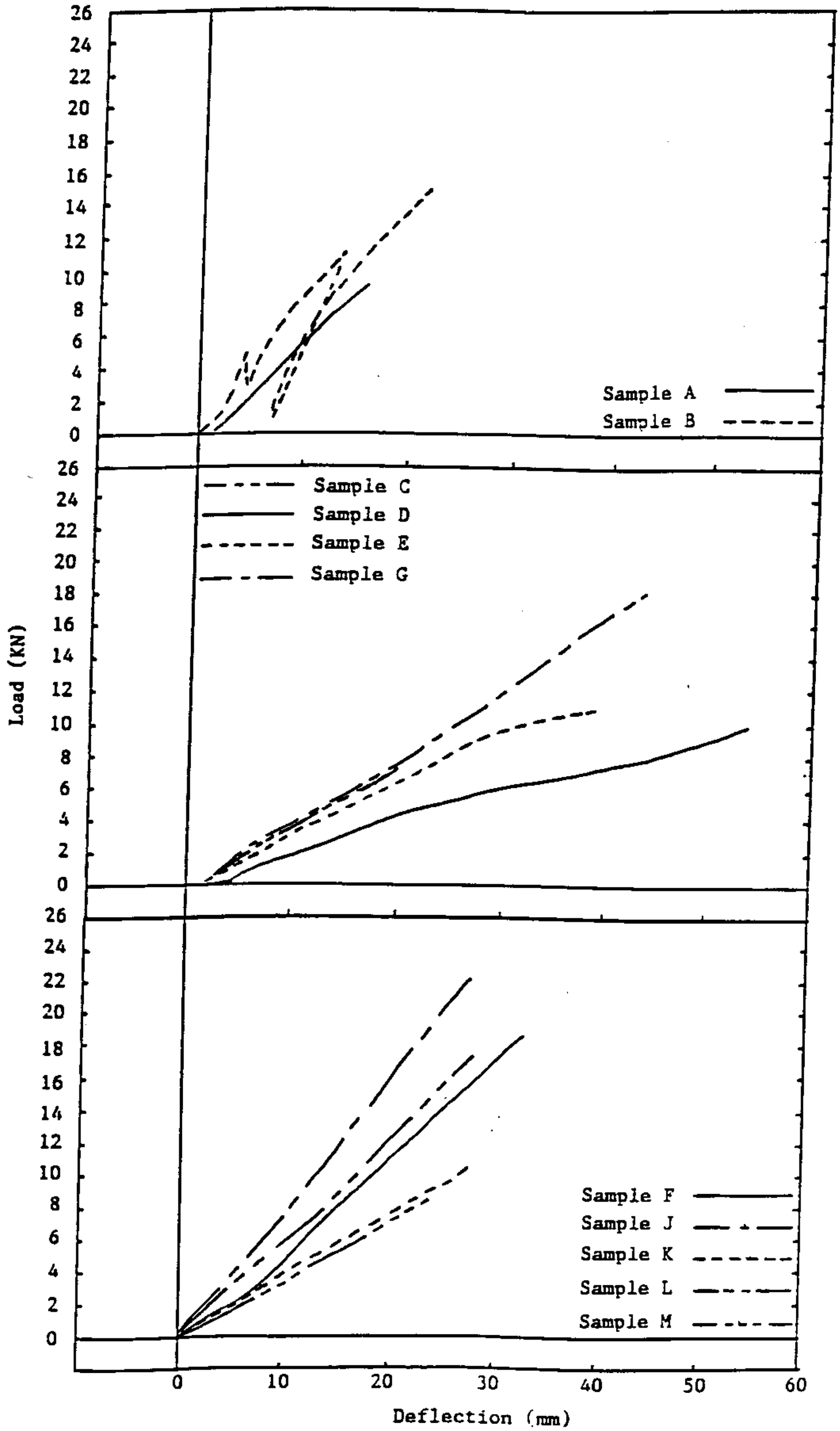
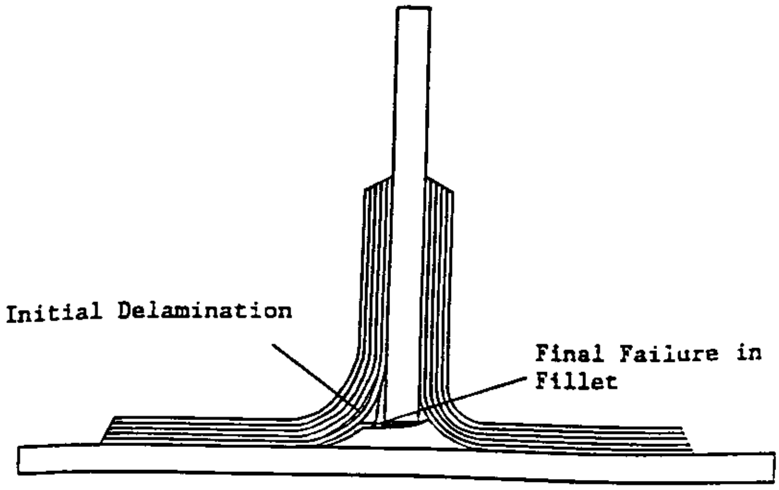
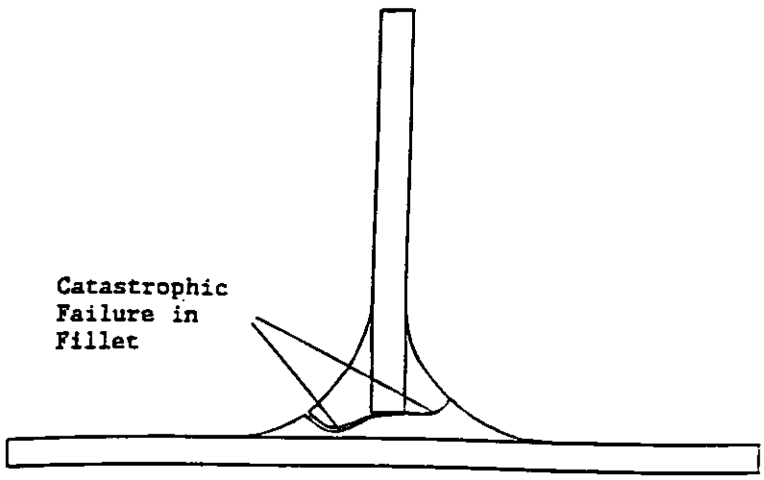


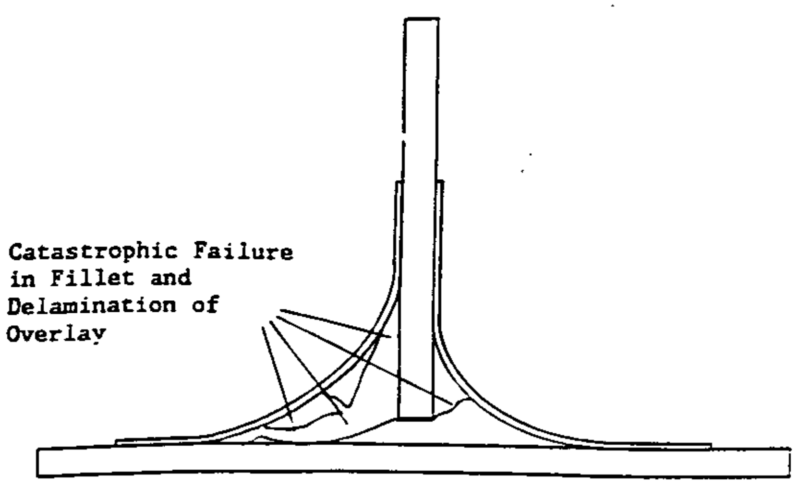
Figure 6: Failure Configurations in Various Samples



a) Configuration A



b) Configuration C



c) Configuration F

Charles Smith Memorial Conference

Recent Developments in Structural Research
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Design of Joints and Attachments in FRP Ships' Structures.

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16th June 1992.

ABSTRACT

This paper examines the problems of forming efficient joints between the major structural components of fibre-reinforced plastic (FRP) ships and boats. Joint types addressed by the paper are concerned primarily with stiffened single skin construction ships:

(i) *Tee-joints between panels, for example between watertight bulkheads and shell plating.*

(ii) *Attachment of top-hat stiffeners to plating.*

The development of numerical models to assess failure modes is discussed and the results of finite element analysis and validation testing of a range of alternative configurations is described. Optimised joint designs capable of achieving higher strength at reduced weight are presented. These new concepts would also facilitate the use of pre-formed stiffeners and mechanised production equipment for rapid rates of joint formation and therefore they offer the possibility of significantly reduced labour hours and ship build cost.

1. INTRODUCTION

Most fibre reinforced plastic (FRP) ships and boats are fabricated using an open-mould, wet lay-up process. This involves the laying up of individual layers of fibre reinforcements onto a mould surface to be consolidated with a thermosetting resin. The resin cures at room temperature and, together with the fibre reinforcements, forms a self-supporting laminate which can then be de-moulded and handled like any other structural material.

In the assembly of FRP ships' structure from component parts, it is inevitable that many bonded joints will need to be formed between laminates which are fully cured. Examples are bulkhead-to-shell and floor-to-tanktop connections. Currently such joints are formed by laminating strips of reinforcements either side of the parts to be joined, forming a double angle tee-joint.

To fabricate stiffened panels such as decks, bulkheads or the hull shell, the layup of the shell is first completed and allowed to cure before stiffeners are laminated directly onto it in-situ. Stiffeners are usually of the top-hat type, formed individually by placing a foam former or trapezoidal section onto the plating and laminating strips of fibre reinforcement over it, with the edges of the cloth landing on the plating to form attachment flanges on either side.

Figure 1 shows typical arrangements for tee-joints and top-hat stiffeners for naval and commercial applications. Whilst these joints are straightforward to form manually, they are labour intensive and therefore expensive. Also, in the case of tee-joints, boundary angles account for about 10% of the structural weight of the ship (whereas only about 2.5% is typically allowed for all weld metal in steel vessels). In a similar vein, top-hat stiffeners account for around 30% of a ship's structural weight.

Figure 2 shows the structural configuration of a 50m long FRP minehunter, prior to fitting of the main deck. The extent of tee-joints and stiffener connections in the internal hull structure can be clearly seen.

A key characteristic of such joints is that, because of a lack of continuity of reinforcing fibres across the joint, they are susceptible to failure by peel or delamination well before the ultimate in-plane laminate material stress is reached. Furthermore, their dependence on interlaminar properties make them somewhat sensitive to material imperfections such as voids and to minute changes in geometry in the laminate. Because of such sensitivity in structural performance and the weight/cost implications involved, it is important to ensure that the design and production of such joints is carefully carried out (1).

The purpose of this paper is three-fold. Firstly, key features to be considered in joint design are outlined. Secondly, an attempt has been made to understand failure mechanisms in the joint. Finally, important material and geometry variables affecting joint strength and flexibility are identified.

2. CURRENT PRACTICE.

2.1 Material Aspects.

A wide range of materials is available for marine use. Fibre reinforcements include E and S Glass, high strength and high modulus carbon and aramids such as Kevlar. Resin choice could be from polyesters, epoxies, phenolics or vinylesters. A full review of material properties and relative advantages and disadvantages is available from published sources (2,3).

In production ships and boats, E-Glass reinforced polyester still dominates due to its low cost, suitability to manual, open-mould, wet lay-up process and room temperature curing characteristics. The scope of the work outlined in the rest of this paper is restricted to this natural choice for construction of the parent laminate. Variation of parent laminate material or lay-up was deemed to be outside the scope of this study, being invariably determined from global structural considerations. All base plates refer to laminates consisting of glass woven rovings in a matrix of polyester resin. Table 1 lists the main engineering properties of such layups.

With reference to joint configurations, the primary feature of influence is compatibility of properties between the cloth/resin plies and the resin itself. The cloth/resin plies, i.e the laminates, have a low modulus and are relatively flexible. Polyester resin though is brittle and may be subjected to a stress raising effect at the interfaces with reinforcing fibres. A consequence of this is that through-thickness tensile strength (@ 10 MPa) is low compared to the resin strength (@ 50-70 MPa), with the latter being associated with a strain to failure of about 3% (4).

Because some of the extreme loads lead to high strains in the region of the joints, it is important to address the question of resin flexibility. A vastly improved capability in this context is achieved through the use of compliant resins. These are urethane acrylate resins in a styrene monomer which are chemically compatible with polyester resins and may be blended with them. The ultimate strengths are lower than those of polyester resin but elongation to failure is much greater, as shown in Table 2.

2.2 Production Aspects.

A feature of current boundary angle and stiffener production techniques is the exclusive use of the manual lay-up process. This is arguably a satisfactory situation when the work takes place on open flat panels in the down-hand direction, where access to the joint is easiest and good ventilation of the working area is possible. The internal structure of ships is built in a number of separate modules which can be turned during fabrication work so that downhand laminating is maximised (Figure 3). However, much of the later assembly work takes place inside the ship or boat hull, with laminators working from staging on the side of the hull or inside enclosed spaces. One of the most unpleasant tasks to be faced is laminating overhead joints inside a small enclosed space such as a fuel or fresh water storage tank. Here the physical access problems are compounded by the requirement to wear breathing apparatus and work under temporarily erected lights. Although ventilation trunking can be directed into tanks to remove styrene enriched air, it is only a partially effective measure. Ironically the tanks are some of the most highly loaded parts of a ship's structure in service due to internal fluid pressure and so, as well as having the heaviest laminate build up, the achievement of good levels of workmanship is of paramount importance.

Clearly, therefore, any improvements in tee-joint design must be aimed at using boundary angle laminates more efficiently and overcoming the current practice which requires a heavy build-up of thickness where the applied load is highest. For this reason tank joints were selected as the subject of investigation, with the intention of reading results across to other, less critical parts of the structure.

Stiffener production presents less of a problem than tee-joints, since they are added to panels and hull at an early stage before other structure adds to the complexity. However, the philosophy of creating FRP stiffeners in-situ is far removed from that of the steel shipbuilding world, where pre-formed stiffeners of a variety of standard sections are supplied to the shipyard, cut to length and welded into place. Not surprisingly, stiffener lay-up productivity is overall around half that of plating.

The current method of fabrication of a top-hat section FRP stiffener places a limitation on the geometry variables affecting the strength of its attachment to a panel. The flange thickness is by default the same as that of the stiffener webs. The web thickness is determined by the stiffener's requirement to act as a beam with the web supporting shear loading as the beam is placed in bending. Therefore the web thickness will drive the flange thickness, rather than vice versa. However, consider an alternative configuration whereby a pre-formed stiffener comprising an upturned U-section is connected to the panel by boundary angle type joints either side - see Figure 1. All the joint geometry variables considered for tee-joints such as fillet material, fillet radius, overlay material and overlay thickness will then apply without affecting the section modulus of the stiffener.

Another production-related influence on the jointing problem is the recent development of portable equipment for mixing and dispensing thixotropic resins such as urethane acrylate. It is most important when mixing such material not to introduce air pockets as air will remain trapped . This will not rise to the surface in the same manner as with low viscosity liquid resins. The equipment was developed for this reason and consists essentially of a resin container, pump, helical mixing device and specially shaped applicator. Access is possible to any part of the hull and the equipment is used to dispense a resin fillet of the required radius in a continuous operation at the rate of about 1 metre/minute. In use it can be likened to fillet welding of steel structure. From the operators point of view, it is cleaner and many times faster than applying laminate to the joint using the hand lay-up technique. Such an approach could have significant implications for the way in which stiffened single skin FRP ships are built.

A new production process is being developed by Vosper Thorneycroft to enable large high quality FRP mouldings to be cost effectively produced. Originating in the U.S.A., the process is a modified form of vacuum-assisted resin transfer moulding, which produces laminates of extremely high and consistent strength and quality. Using this process for the production of stiffener sections and the revised jointing method for attaching stiffeners to plating, the following advantages over the current manual layup method are apparent:

(i) High fibre volume fraction laminates can be produced, resulting in stiffener laminate strength and modulus approaching twice that of the hand layup plating, using the same constituent materials.

(ii) Producing stiffeners separately from plating means that most of the laminating work can take place in ideal workshop conditions rather than inside the hull where access is more difficult.

(iii) It is estimated that, if this approach was applied throughout a ship, a saving of 30% in the weight of stiffeners can be achieved, including avoiding the need for foam formers which become redundant once the laminate cures, but nevertheless have to remain built in to the structure. This method of producing and attaching stiffeners is most suited to flat panels, such as bulkheads, decks and superstructure, and plating of moderate curvature.

A feature of the analysis carried out in Section 4 is to ascertain the viability of this concept, validate its performance against "traditional" construction and identify failure mechanisms under load.

2.3 Design and Analysis Procedures.

One of the earliest approaches to GRP structure design is outlined in the Gibbs and Cox manual (5). This gives recommended arrangements of various joints with simple design examples. 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 compatible with strength requirements". However, no specific procedures concerning joint design are elaborated. Early work in the U.K. centred around the design of GRP minehunters (6) and this formed the impetus to the drawing-up of naval engineering standards. These rules for naval ships are based primarily on extensive experimental work. 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 non-naval craft, design guidance is sought primarily from classification society rules. Lloyds rules (7) state that, for top-hat stiffeners, the "width of the flange connection to the plate laminate are to be 25mm + 12mm per $600 g/m^2$ of reinforcement in the stiffener webs, or 50mm 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 "50mm + 25mm 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.

ABS rules (8) state that, for top-hat stiffeners, the minimum overlap on the plating should be 20% of the web depth or 50mm, whichever is greater. With regard to boundary angle connections the rules cater for three types of connections, namely single-skin to single-skin, sandwich to sandwich, and sandwich to single-skin. There is a minimum thickness requirement in each case. With regard to the first of these, boundary angle is to be "one-half the thickness of the thinner of the two laminates being joined".

DNV rules (9) specify first principles based calculation procedures to determine section moduli of top-hat sections and scantlings of plate laminates. However, no explicit guidance is given with regard to lay-up of the boundary angles or the flanges of the top-hat stiffeners.

It can be noticed from the above, that production considerations, especially concerning advances in material technology, have not been explicitly accounted for.

Specific and detailed analytical studies in context of joints and top-hat stiffeners are surprisingly rare. An early approach for marine applications was through the use of plane strain F.E. analysis (10). A detailed stress pattern was derived, though the elements used may not have permitted a full coverage of all composite material related properties. Gillespie and Byron-Pipes (11) conducted a detailed analysis of a spar-to-wingspan joint in an aircraft. Analytical results comprising interlaminar normal stresses, interlaminar shear stresses and inplane bending stresses are presented as a function of various geometrical parameters. More recently, Shenoi and Violette (12) have examined sandwich joints. They sought to identify, through physical testing and numerical modelling, the impact of different configurations and geometries on the behavioural efficiency of the joint. Such an approach was felt to be the most appropriate in the case of single-skin as well and formed the basis of investigations presented in the next two sections.

3. STUDIES OF TEE-JOINTS.

3.1 Experimental Programme.

The variables influencing the design of tee-joints are identified in Figure 1. A test programme was devised to study the influence of the most critical variables. The variables included in this program are as listed below:

- Fillet radius,
- No. of plies in boundary angle,
- Material make-up of boundary angle plies,
- Edge gap between the web and flange of the tee,
- Shape of the edge of the wedge.

The variations have been compared with "standard", NES 140 prescribed, specimens. The details of the specimens are outlined in Table 3.

The test specimens were subjected to a displacement controlled pull-off of the web at 45° to the base (flange) plate and restrained as shown in Figure 4 (13). 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. Loads and deflections were systematically recorded. The load/deflection curves for the samples are shown in Figure 5.

Sample A failed initially by delamination within the boundary angle corner on the side in tension - see Figure 6a. This delamination occured within the third ply of the overlaminate. The load carried by the joint dropped away almost to zero. 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 delamination had occured. Final failure occured when the remaining boundary angle plies delaminated and the fillet failed.

Sample B failed in a similar manner to sample A. The delamination occured in a similar region and, as before, the stiffness of the joint and its ability to withstand load seemed little altered by this initial failure.

Sample C failed catastrophically in the fillet. The point of initiation of failure was impossible to judge due to the speed at which it occured. Figure 6b shows the position of the cracks in the fillet.

Sample D also failed catastrophically in the fillet. However this was preceded by large cracks appearing between the fillet and the tee piece. As these developed the stiffness of the joint was reduced and large deflections occured. At a deflection of approximately 45mm the cracks ceased to propagate and the stiffness increased back to the original value. At this stage one of four samples failed completely while three others failed at significantly higher (75%) loads.

Sample E failed in a similar manner to sample C by catastrophic failure of the fillet. This was not preceeded by any significant cracks appearing in the fillet although the load/deflection curve does show a gradual reduction of stiffness towards failure.

Sample F failed catastrophically by complete delamination of the overlay on the face in tension and splitting of the fillet - see Figure 6c. These two mechanisms occured too rapidly to distinguish their order. These samples showed approximately constant stiffness to failure and withstood the highest loads of all samples tested.

Sample G also failed catastrophically with pieces flying away from the destroyed fillet. Some creaking could be heard at relatively low loads (@ 50% of failure) but there was no visible damage at this stage.

Samples J,K,L and M failed in a similar manner to F. The load/deflection curves in these cases were approximately linear.

To confirm the applicability of the above results to the original problem, that of tank boundary angles, tests were conducted using a full scale tank as used previously (14) but with the top boundary angles replaced with the new design. A maximum head of 9m of water was applied to the tank with no damage being sustained, compared to the previous maximum head of 5m.

With an applied head of 5m the maximum deflection of the tank top was 13mm, the same as for the previous test. This increased to 23mm for the 9m head. Thus the new design of boundary angle does not affect the global properties of the tank, and yet can withstand a 75% increase in deflection without failure.

3.2 Numerical Modelling.

The numerical analysis was performed using the ANSYS finite element package (15). The models were made up of two elements across the width with one element through the thickness of the plates and two elements through the thickness of the boundary angle. Figure 7 shows a cross section through a typical model. The material properties are shown in Table 1.

An extensive examination of stress patterns in all sample variants was carried out with a three-fold objective. First, an analysis of the stress regimes was conducted with a view to correlating internal behaviour of the joints with experimental observations. Secondly, systematically varied designs, incorporating material and geometrical changes, were studied to verify their impact on design. Lastly, the results are interpreted in a manner so as to understand joint efficiency.

A) Study of stress regimes

Only three sample sets - B, C, and F - are selected for discussion here because they represent the possible types of joints in a production context. Sample B is a "traditional" fillet with a thick overlaminate, sample C is a pure fillet and sample F is a fillet with a very thin overlaminate.

First, consider sample B. Figure 8 shows, from the top, the through-thickness stresses in the outer laminate, the inner laminate, and the fillet respectively, for a relatively high applied load of 12.73KN. The maximum through-thickness stresses occur on the outer face of the inner laminates at positions marked 'm' and 'n', with a maximum value of 10.85MPa, indicating that the joint would have failed at this load. The maximum stress in the fillet is 6.39MPa. Since the fillet would be expected to withstand more than twice the stress compared to the laminate in the through-thickness direction, it can be seen that failure would initially occur at positions 'm' and 'n' within the laminate. At this load stage, the fillet is virtually "redundant".

To model the behaviour of the joint after this 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 9 shows the stress contour plots (in the same format as Figure 8) but with gap elements included. High values (up to 27.88MPa) of through-thickness stress occur within the overlaminate at the edge of the delamination, indicating that the laminates would continue to peel at these points. In the fillet, stresses have increased to a maximum of 8.07MPa, illustrating the transfer of load from overlaminate to fillet as delamination proceeds. The experimental results discussed in the previous section have shown that final failure occurs only when the fillet fails, and that the load carrying ability of the joint is little affected by these initial delaminations. This indicates that overlaminates that delaminate are too thick and the extra layers are being used inefficiently.

Next, consider sample C. This is made up of a large radius urethane acrylate fillet without any reinforcing overlay. The failure mechanism is thus relatively simple in that this occurs catastrophically once the stress levels in the fillet exceed the UTS value. Figure 10a shows the stress distribution within the fillet at the same applied load as in Figures 8 and 9. Stress levels are much higher than in sample B (up to 23.85MPa, again indicating that the joint would have failed at this load) because the fillet is carrying all of the applied load, whereas in sample B the majority of the load is carried by the overlaminate in the in-plane direction.

Figure 10b shows the same plot as in Figure 10a but additional arcs (labelled 1-4) are drawn on. The direction of the stress plotted on this diagram (the maximum principal stress) is approximately tangential to these arcs, and hence crack propagation will occur in a direction perpendicular to these. Initial crack formation will occur on arc 1 (ie. the outer edge of the fillet) in the region marked w-w since this is the region of highest stress. Once the crack opens it will propagate rapidly as there are no reinforcing elements to arrest the crack. On arc 2 the crack would be expected to pass through the region x-x and similarly, it will pass through regions y-y and z-z on arcs 3 and 4 respectively, and on around the tee to the limit of the tensile stress contours. This indicates a crack path as shown in Figure 10c, which correlates well with the experimental samples (see Figure 6b).

Sample F is similar to sample C in all respects except that it has a thin overlaminate of woven roving around the fillet and extending a short distance to either side of it. Sample F can also be described as being similar to sample B, except that the radius of the fillet is increased and the thickness of the overlamination is much reduced. Thus the failure mechanism could proceed in two ways - delamination followed by fillet failure, or catastrophic fillet failure only.

Figures 11 shows, for sample F, the maximum through-thickness tensile stress in the laminate and the maximum stress in the fillet respectively, at the same load as before. The maximum through thickness stress in the laminate is shown as 2.79MPa and the maximum stress in the fillet as 14.46MPa. This indicates that failure would tend to occur within the fillet and not by delamination. In this case, both elements of the joint are being exploited much more evenly and hence (judging from the stresses given above) show that this joint can withstand a substantially higher load than that being applied before failure will occur.

The above demonstrates that it is possible to get a qualitative assessment of failure paths using relatively simple models. Furthermore, it is evident that the paths are dependent upon layup considerations.

B) Impact of material and geometry variations.

The experimental programme and previous work (12) indicated that variations in geometry and material make-up of the joint will have an impact on joint behaviour. An attempt was therefore made to explore internal joint stresses as a "function" of such variations. Maximum stresses occuring in the joints at a constant load of 12.73KN were extracted and are summarised in Table 4.

The chosen designs were then compared, on the basis of these stresses, in context of five major features characterising geometrical and material differences. These are listed below.

1) Compare cases B and K: Both have the same fillet radius and are made of the same material. Overlaminate thickness of B (@ 15mm) is greater than that of K (@ 2mm).

2) Compare cases F and K: Both have the same overlaminate thickness and are made of the same materials. The radius of F (@ 75mm) is greater than that of K (@ 50mm).

3) Compare cases C and G: Both are pure fillets with the same material. The radius of C (@ 75mm) is less than that of G (@ 100mm).

4) Compare cases C and F: Both have the same radius and their fillet material is the same. However, whilst C is a pure fillet, F is overlaminated.

5) Compare cases F and L, and cases K and M: Both sets have the same radii and overlaminate thickness. However F and K use polyester/urethane acrylate mix while L and M use polyester only as the laminating resin.

The impact of these geometry and material variations on stresses is summarised in Table 5.

C) Joint efficiency.
The stress patterns in different joints and the different failure modes indicate that there is an "optimum" case where the maximum stress in the fillet and the through-thickness stress in the overlaminate both reach their respective limits at the same load. This, along with observations from Table 5, indicates there is a need for a measure of joint "efficiency" in terms of both overall joint strength/stiffness and internal component strength.

"Efficiency" in this context could be defined as the ability of the joint to remain integral and intact at as high a load and for as high a deflection as possible. This can be extended, for interpretation of internal stresses, as being equivalent to the peak stresses being as low as possible for a given load and deflection.

In the present context, this efficiency can best be explored by examining plots of peak joint stresses versus load and deflection, as shown in Figures 12 and 13 respectively. It is noticable that case F has the lowest stress levels for a given load and deflection. This indicates that the joint will be capable of withstanding higher loads and larger deflections than other design variants before failure. This feature is borne-out and confirmed by the experimental records.

4. TOP-HAT STIFFENER JOINTS.

4.1 Basis of Study.

The fundamental approach adopted here is similar to that in the case of tee-joints. This involved, initially, a validation of the numerical model with respect to some experimental results pertaining to samples fabricated as per current practice (see Figure 1 - current naval practice). This validation of the the approach was followed by a study of the impact on performance of a systematic variation of design features. The variations refer to the "new" production approach described in Section 2 involving connections of preformed inverted U-sections to base plate laminates through boundary angles. The variations considered were: radius of fillet (25-125mm); thickness of overlaminate (1-12 laminates); gap between base panel and stiffener (10-50mm): and the fillet backfill angle inside the stiffenner (0-45°). These are illustrated in Figure 14. The boundary conditions applied to the models are shown in Figure 15. Centre clamp loading has been shown by previous work (16-18) to be 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.

Figure 16 shows the load/deflection characteristics for the centre clamp condition. As can be noticed, the trends for the test results and F.E. results are similar in the case of specimens owing to current practice. Furthermore, the trend for the "new" design, incorporating advances in production techniques, is also similar. This preliminary result indicates that the "new" design does not affect the global properties of the top hat stiffener.

4.2 Effect of Geometrical Variations.

The design variants pertaining to the proposed new production process were then studied in detail. 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 analyses were conducted in the case of both centre clamp and two clamp edge conditions. The results were then compared on the basis of three criteria. These were the overall stiffness of the boundary angle (measured as the deflection at the point of load application), the highest value of principal stress in the fillet, and the through-thickness direct tensile stress in the overlaminate.

The results of the study are summarised in Table 6. 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 5 laminations.
- For two clamp loading, the stress in the fillet increases slightly as overlaminations are increased up to 4 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°. For two clamp loading, the stress in the fillet remains constant, and then rises slightly as the backfill angle increases to 45°. This is matched by an increase in stiffness at this angle.

4.3 Failure Modes.

By considering the stress regimes within the models, a qualitative picture of the failure modes can be seen. The experimental work described in References 16-18 showed that failure occurs in two ways, either by cracking in the fillet leading to peeling along the shell/boundary angle interface or by delamination in the boundary angle leading, again, to peeling along the same interface. In general, the first mode of failure is associated with centre clamp loading and the second is associated with two clamp loading. When using the flexible resins as the fillet material this failure occurred instantaneously so the position of the initial failure was difficult to determine.

Figure 17a shows the through-thickness stress in the web-to-shell boundary angle under two clamp loading for the design corresponding to current production practice. The tensile stress concentration that occurs as the boundary angle is opened out is clearly visible. This is the cause of the initial delamination mentioned above. A corresponding compressive stress concentration occurs under centre clamp loading (Figure 18a) but with a greatly reduced magnitude, indicating that there is less likelihood of initial delamination for this form of loading.

Figure 18b shows the inplane stress in the laminate and the maximum principal stress in the fillet for centre clamp loading. The contours clearly show the manner in which load is transferred across the joint as indicated by the arrow. Two clamp loading generates load transfer in the opposite manner as shown in Figure 17b. Failure will tend to propagate perpendicular to these lines of load transfer. For single clamp loading, this corresponds to failure across the fillet from the internal side in tension to the flange/base secondary bond. For two clamp loading, the layered nature of the overlaminate means that loading through the thickness leads to delamination between the layers successively nearer to the fillet. Once the final laminate has delaminated from the fillet the unsupported overlaminates will peel from the base.

Figures 19 and 20 are in the same format as Figures 17 and 18 but show the stresses in the new design. Figures 19a and 20a show that the through-thickness stress concentrations are no longer in the overlaminations, but are now in the web itself with reduced magnitude. Figures 19b and 20b show that the lines of load transfer remain similar in the new design and hence the failure modes will be similar. For single clamp loading, the stress in the fillet is very similar to the original design: so failure will occur at similar loads. In the case of two clamp loading, the reduced through-thickness stress indicated in Figure 19a will delay failure in this mode until higher loads are applied, but the stress in the fillet is increased due to the increased flexibility of the overlaminate. However, the fillet can withstand higher stress levels than the overlaminate in the through-thickness direction; so some increase in applied load at failure would be expected.

Refinement of the new design could improve performance. For example, if the gap between the web and the shell is increased to 30mm, the stress in the fillet in two clamp loading is reduced by 12%, resulting in a corresponding increase in failure load for this mode. A similar stress reduction can be achieved by increasing the fillet radius to 100mm but this would result in an increase in stress for single clamp loading. Thus it can be seen that if the failure stresses of the constituent materials are known then, by considering the stress regimes, the model can be optimised for a given loading condition, or the best compromise can be achieved when accounting for several loading conditions. Further experimental and theoretical work is necessary before a more complete understanding of the optimum design is reached.

5. CONCLUDING REMARKS.

The purpose of this paper has been to highlight design, production and materials related aspects concerning joints and attachments in FRP ships.

Currently, most production ships and boats use glass reinforced polyester as the fabrication material. This is because of its suitability to an open-mould, wet lay-up process. This is a simple process prone to variation in quality and is still labour intensive, especially in context of tee-joints and connections to top-hat stiffeners. Hence, improvements are being investigated with respect to portable resin injectors for use in boundary angles and resin injection moulding process for base laminates and top-hat stiffener sections. These offer potential performance, quality and cost benefits with respect to bonded connections and joints.

The "correct" design of tee-joints is very important with a view to improving structural efficiency. Current design methods do not have explicit ways of modelling variations in parameters and examining their impact on performance. The results of the experimental and numerical studies conducted by the authors have identified two critical variables. Firstly, it has been shown that increasing the thickness of overlamination, traditionally the criterion used for design, has a detrimental effect on joint performance. Secondly, it has been found that the radius of the fillet, traditionally given little or no consideration by current design methods (7,8), is critical to the performance of the joint. In general, increasing fillet radius enables the joint to withstand higher loads.

The design of top-hat stiffeners, which account for a significant proportion of a ship's structure, again has scope for improvement. Present methods (7,8) do not cater for different geometry and material-based design variables. The F.E. modelling conducted in context of this paper has been with a view to assessing the feasibility of producing this connection in a new manner. This involves placing a pre-formed, inverted U-section on to a base plate laminate and then connecting the two using a boundary angle. The results, when compared with past experiments on traditionally formed stiffeners, have shown that the performance of the joint using this new technique is at least as good as before. The analyses have identified trends in performance on varying different design parameters. As in the case of tee-joints, the critical variables are radius of the fillet and number of overlaminations. Further work is necessary to identify designs with best performance and minimum weight/cost.

The key conclusion to be drawn from this work is that there is scope to reduce the number of overlaminations in the case of bonded joints. This will lead to a significant reduction in structure weight and production cost, while at the same time, in most cases, leading to improved structural capability. More experimental work, especially with respect to top-hat stiffeners, is necessary before final implementation in a production context. There is also 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. This theoretical work is currently underway and will be reported in a separate forum.

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TABLE 1
Typical Laminate Material Properties.

Property	Tensile Strength (MPa)	Tensile Modulus (GPa)	Resin Ratio (Weight)	Poissons Ratio
WR/Polyester Warp	238	13.22	45.5	0.178
Weft	175	12.33	45.5	0.152
CSM/Polyester	96.5	6.89	70.0	0.13
WR/CR1200 Warp	183.3	6.37	45.5	-
Weft	188.9	3.93	45.5	-
CSM/CR1200	110.4	3.02	70.0	-

TABLE 2
Typical Cast Resin Properties.

Property	Tensile Strength (MPa)	Tensile Modulus (GPa)	Elongation at Break %	Hardness
Urethane Acrylate	26.0	0.5 (initial)	100	65 (Shore D)
Polyester	75.0	3.5	3.5	42 (Barcol)

TABLE 3
Details of Tee-Joint Test Specimens.

Specimen	Boundary Angle Thickness (mm)	Fillet Radius (mm)	Fillet Overlay	Resin for B.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

TABLE 4
Maximum Tee-Joint Stresses ocurring at Constant Load.

Configuration	In-Plane Stress in Laminate (MPa)	Through-Thickness Stress in Laminate (MPa)	Principal Stress in Fillet (MPa)
B	109.14	10.85	6.49
C	-	-	23.85
F	135.81	2.79	14.46
G	-	-	23.94
K	249.47	9.10	18.81
L	260.49	8.05	10.87
M	301.73	11.65	17.34

TABLE 5
Impact of Variations on 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	Decreases

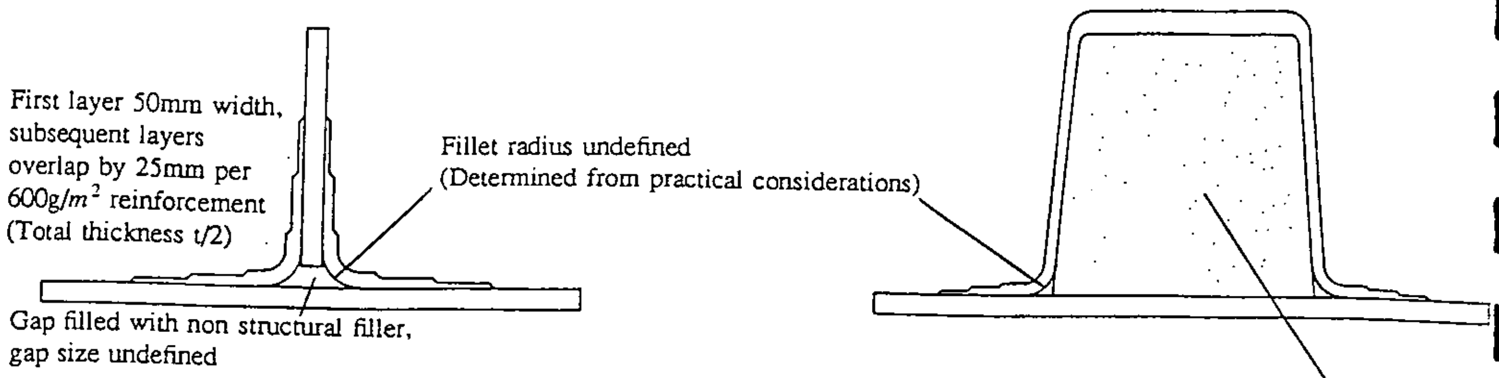
TABLE 6

a) Top-hats: Impact of Geometrical Variations - Centre Clamp.

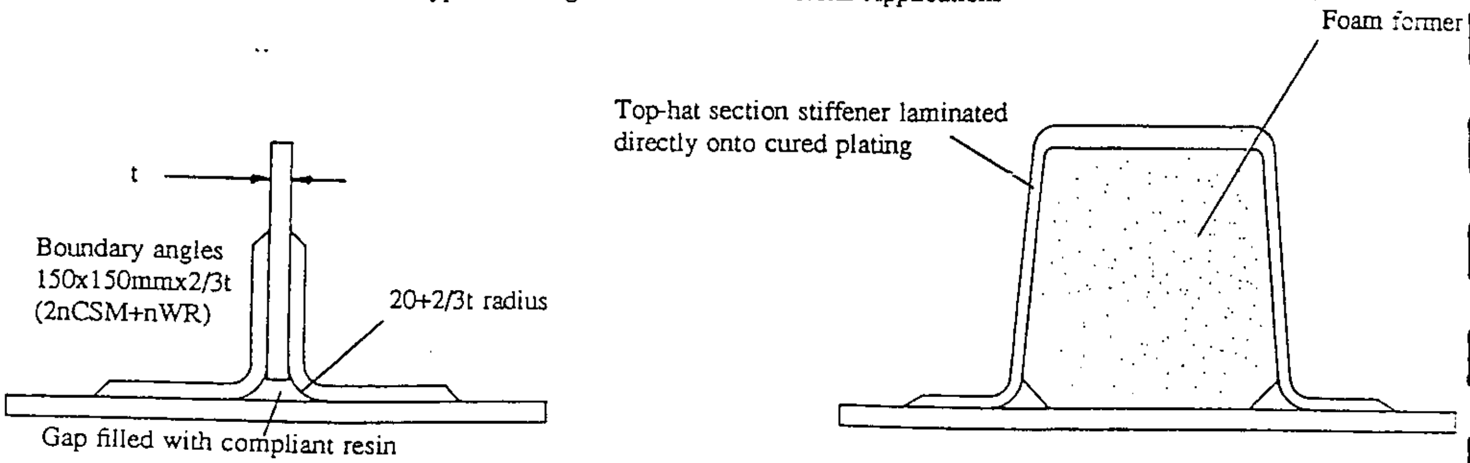
Increase in Property	Effect on Stiffness	Effect on Stress in Fillet	Effect on Stress in overlaminate
Radius	Marginal Increase	Minimum at 75mm	Decrease
Overlaminate Thickness	Marginal Increase	Increase	Increase
Gap	Marginal Increase	Decrease	-
Backfill Angle	Marginal Increase	Decrease	-

b) Top-hats: Impact of Geometrical Variations - Two Clamp.

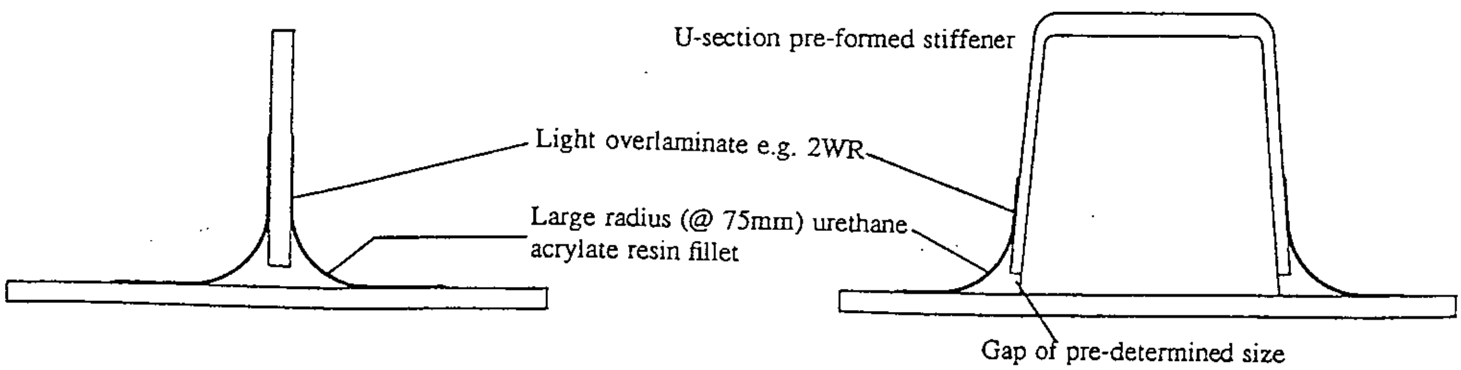
Increase in Property	Effect on Stiffness	Effect on Stress in Fillet	Effect on Stress in overlaminate
Radius	Increase	Maximum at 75mm	Decrease
Overlaminate Thickness	Increase	Decrease	Increase
Gap	Maximum at 30mm	Minimum at 30mm	-
Backfill Angle	Marginal Increase	Marginal Increase	-



Typical Configurations for Commercial Applications



Typical Configurations for Naval Applications



"New" and Alternative Configurations.

FIGURE 1 Tee-Joint and Top-Hat Stiffener Configurations.

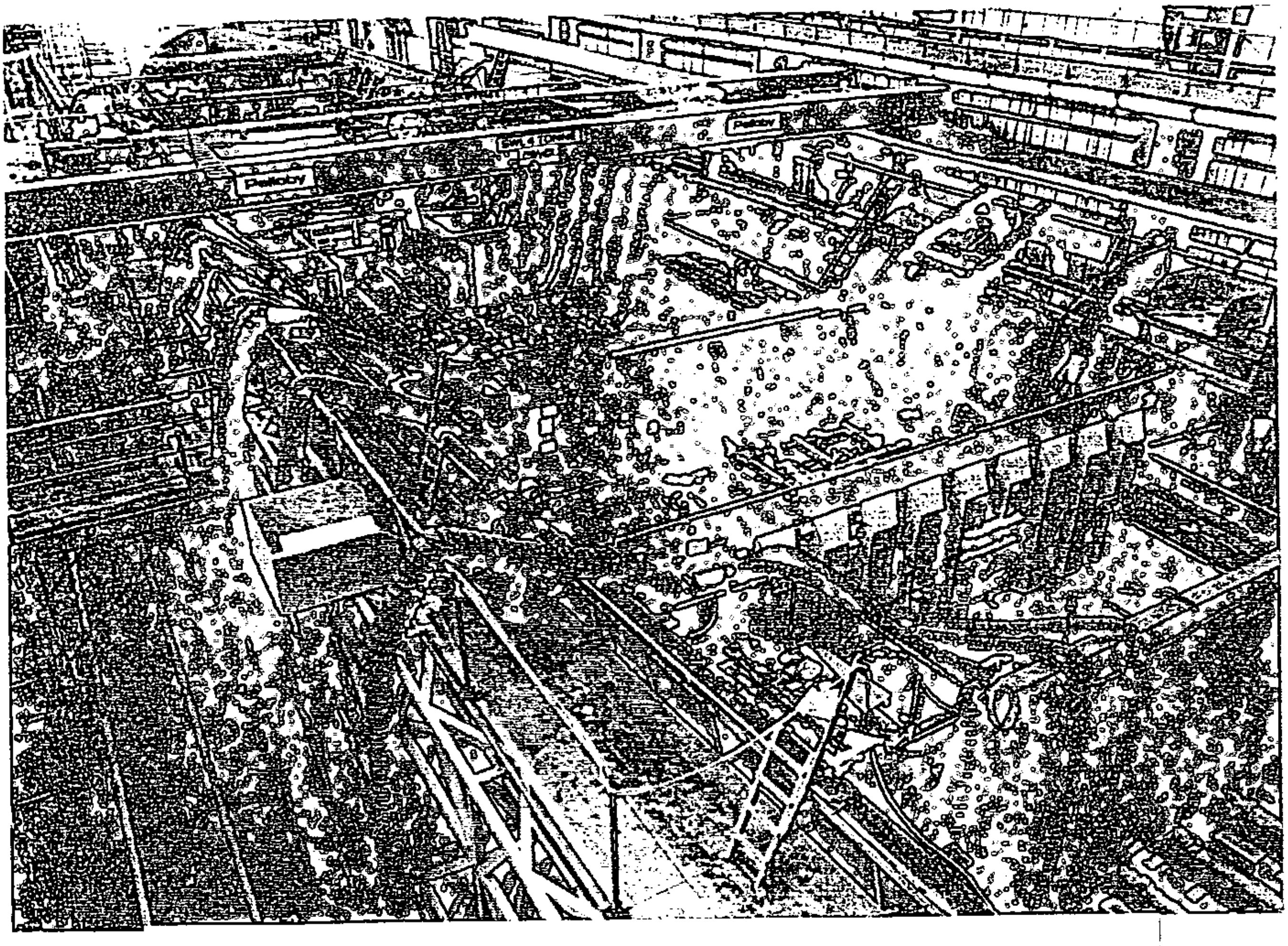


FIGURE 2 Structural Configuration on Typical FRP Ship Hull

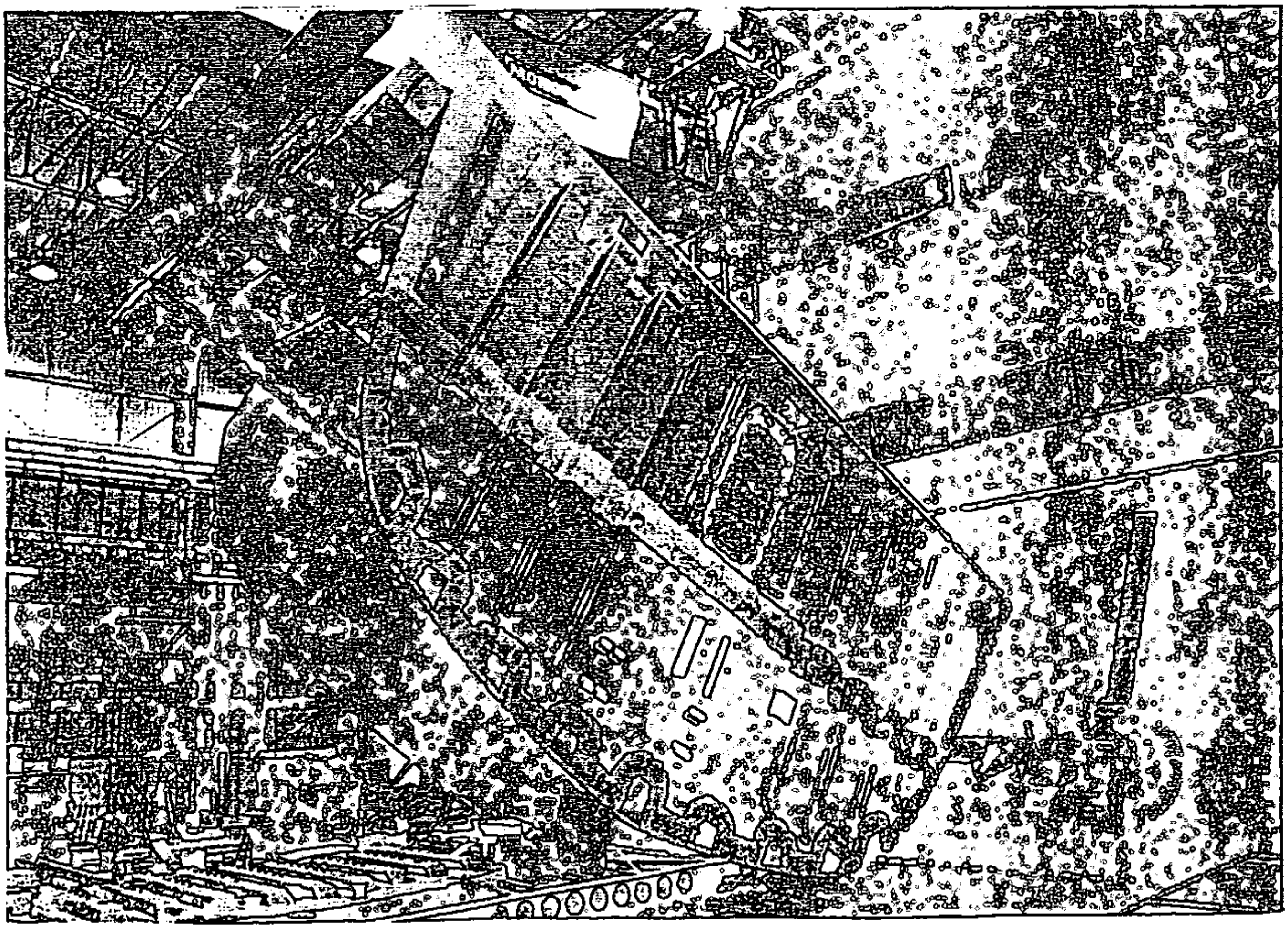


FIGURE 3 A Typical Panel Module

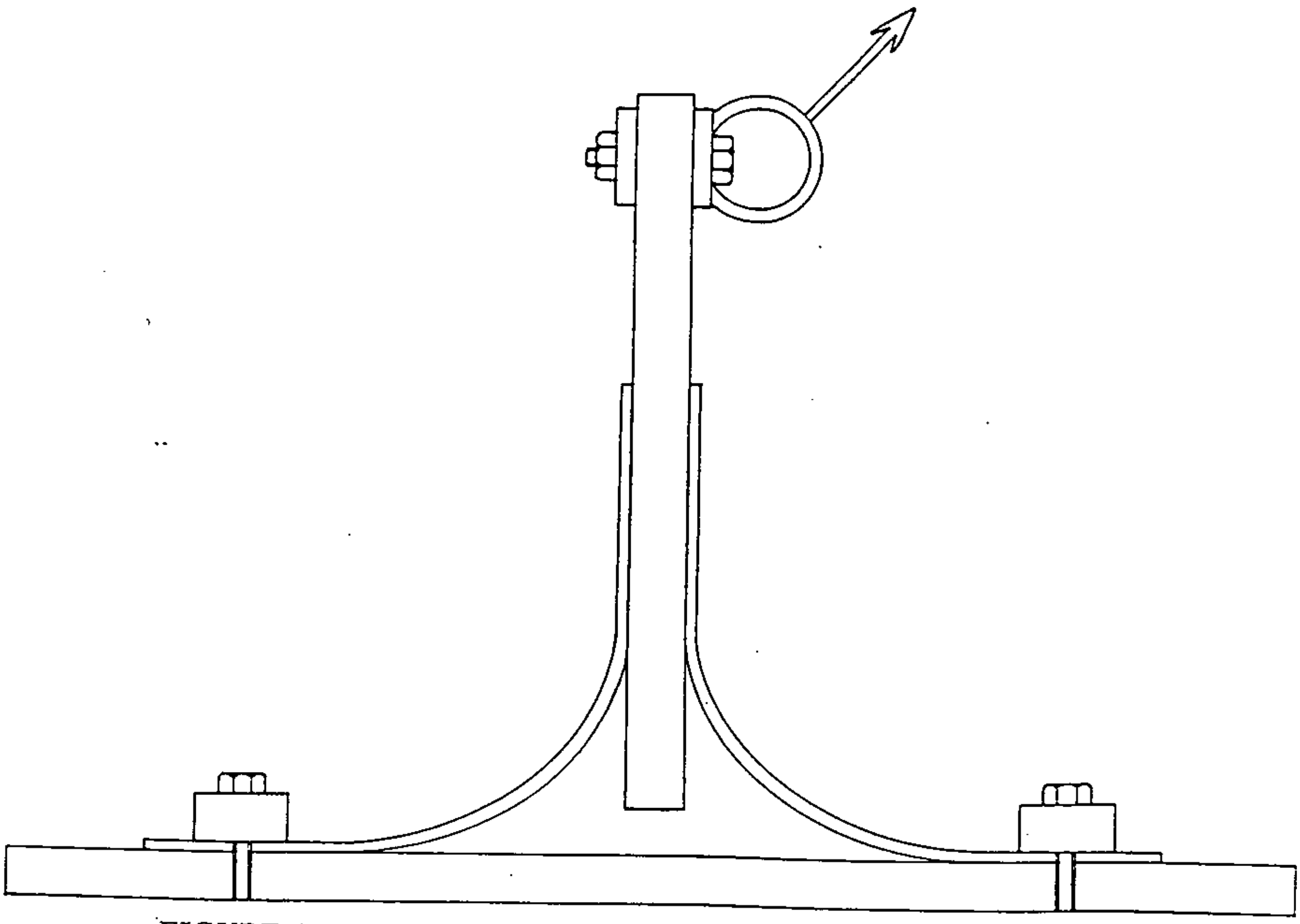


FIGURE 4 Edge Restraints and Loading Method for Tee-Joint Test.

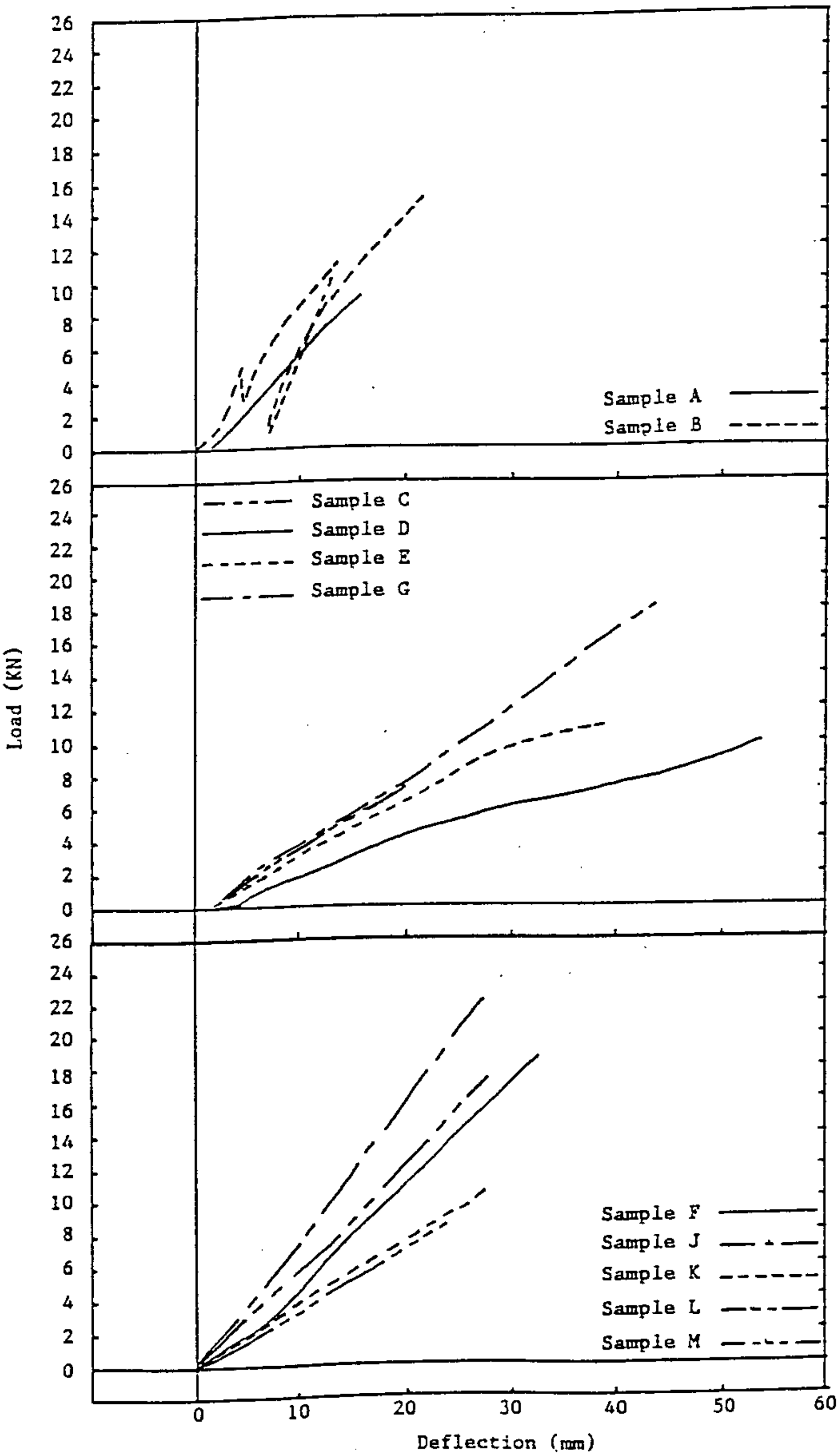
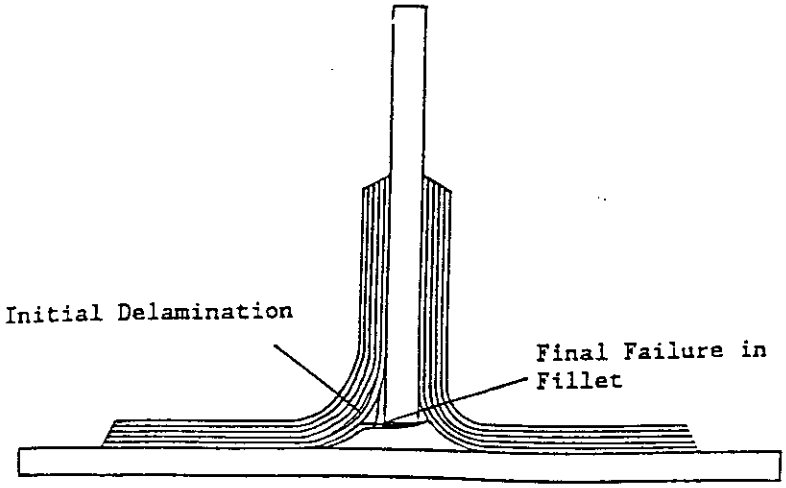
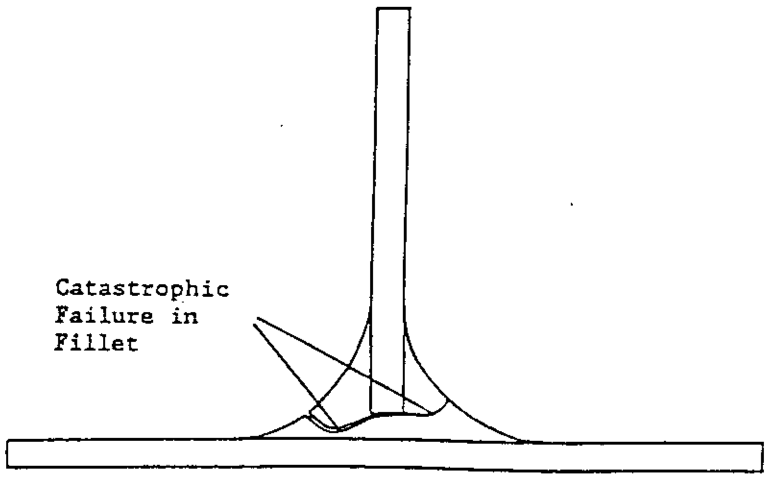


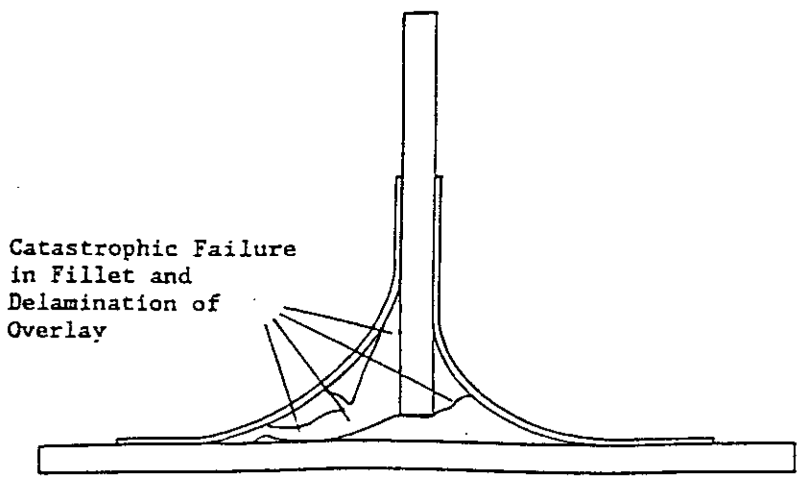
FIGURE 5 Load-Deflection Plots from Tee-Joint Tests.



a) Configuration A



b) Configuration C



c) Configuration F

FIGURE 6 Failure Configurations in Tee-Joint Tests.

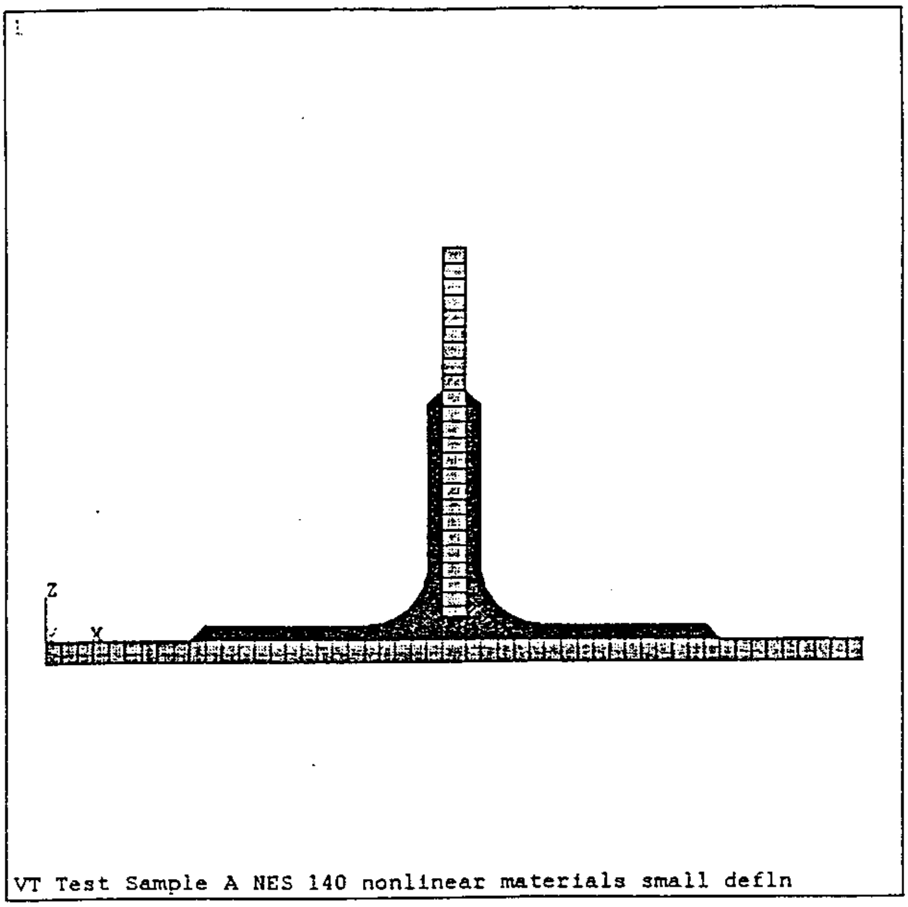


FIGURE 7 Typical F.E. Model of Tee-Joint.

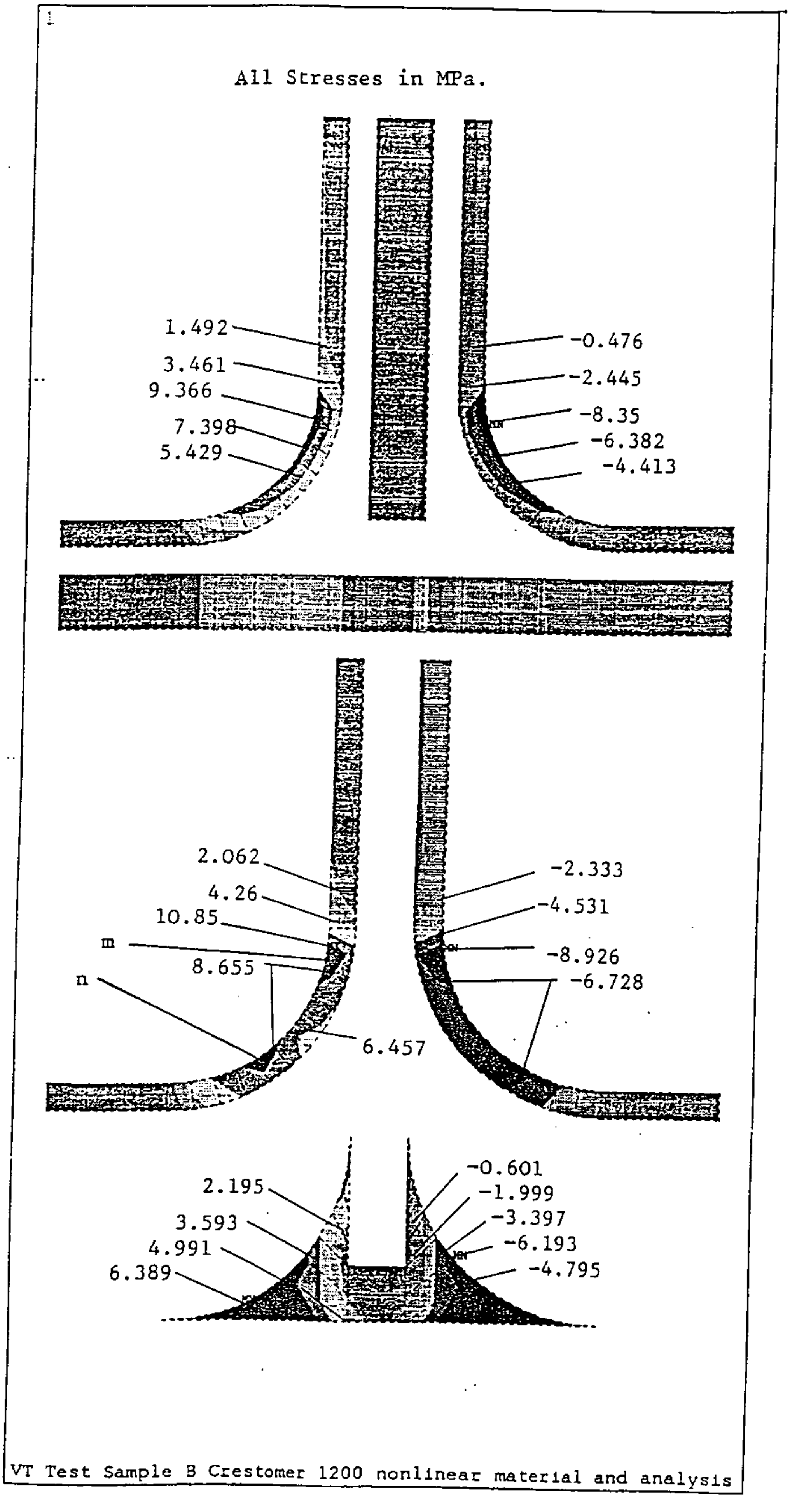


FIGURE 8 Stress Contours within Tee-Joint Sample B.

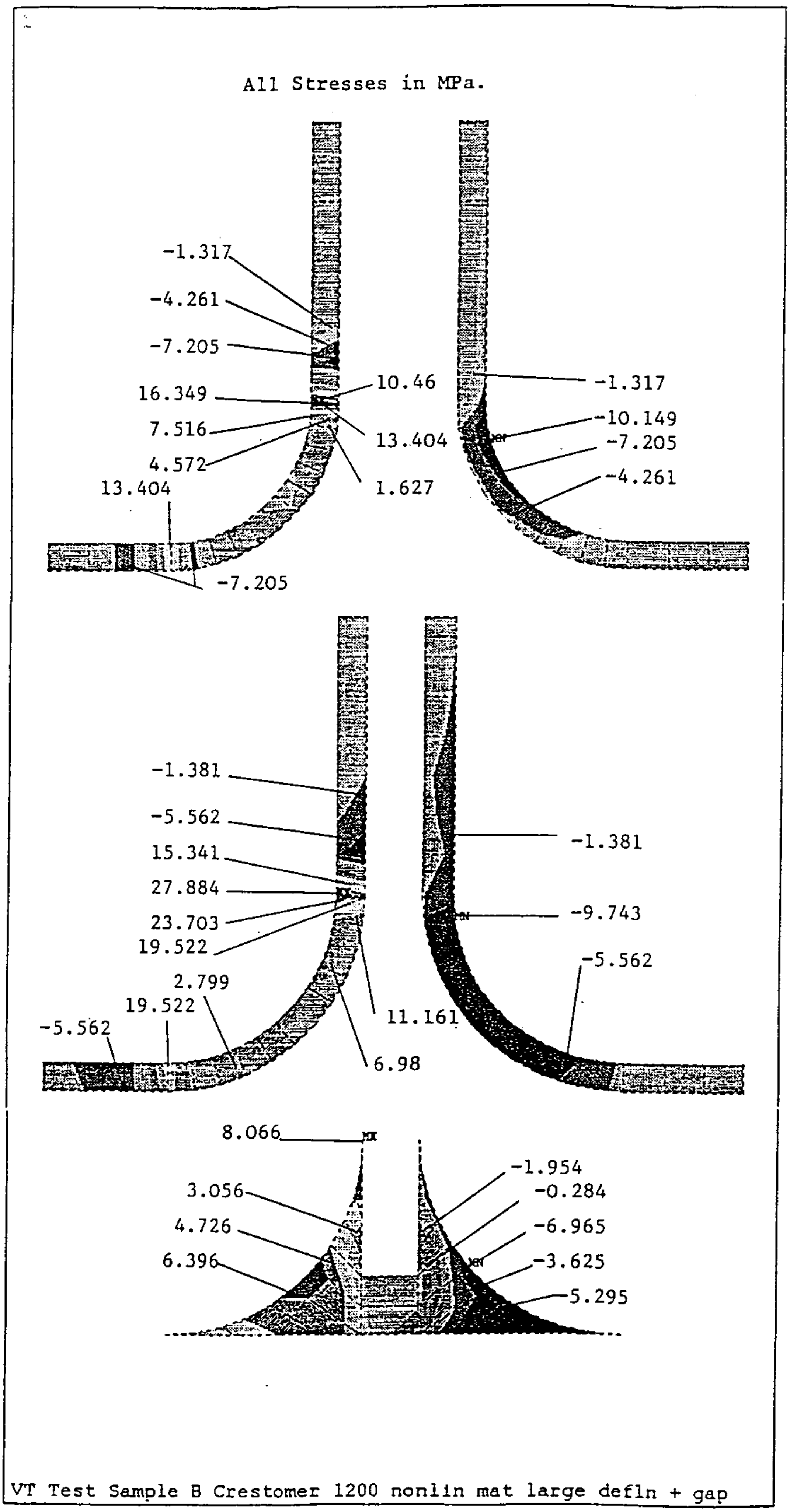
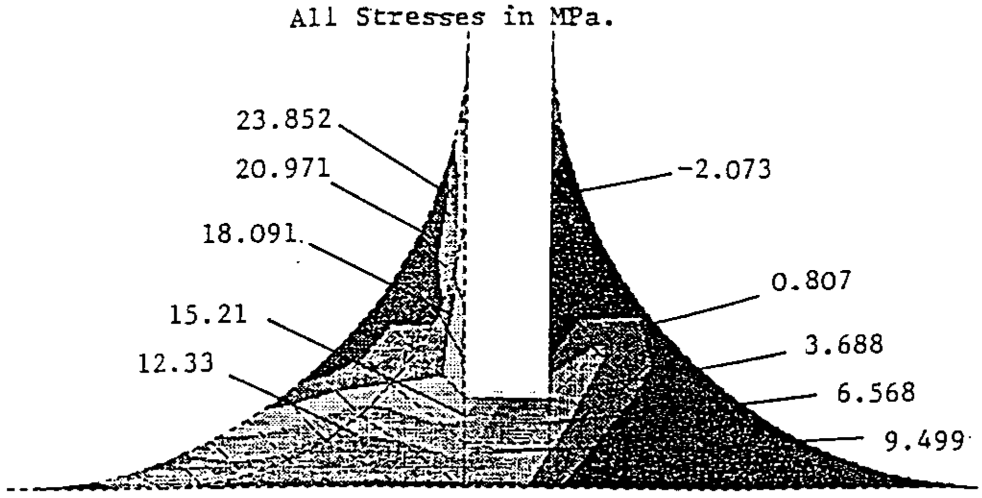
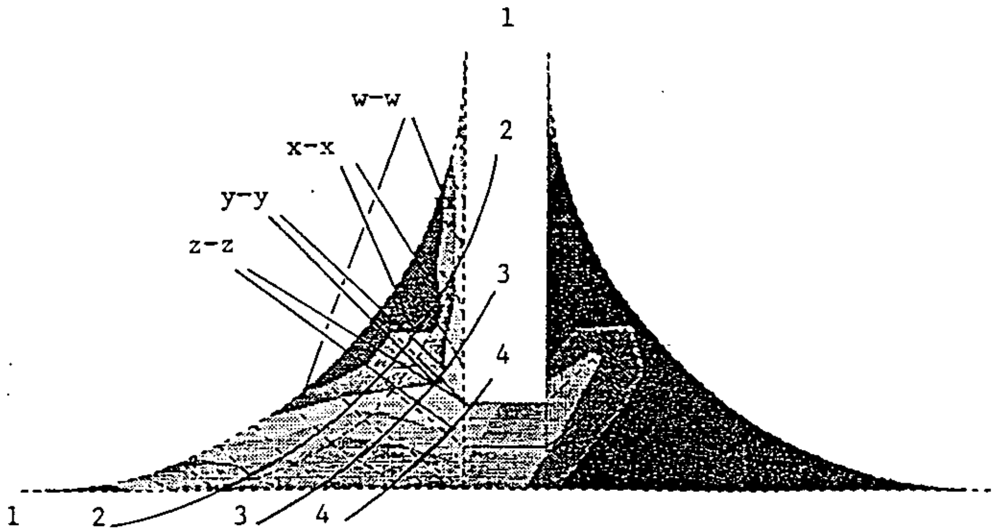


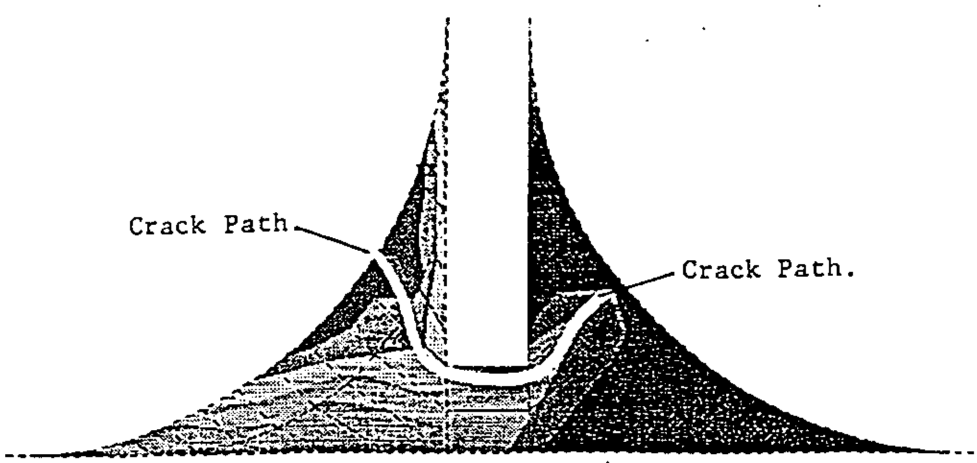
FIGURE 9 Stress Contours within Tee-Joint Sample B - With Gap Elements.



a: Stress Contours Within Sample C.



b: Regions of High Stress Concentrations in Sample C.



VT Test Sample C Crestomer 1152

c: Indicated Route of Crack Propagation in Sample C.

FIGURE 10 Stress Contours within Tee-Joint Sample C.

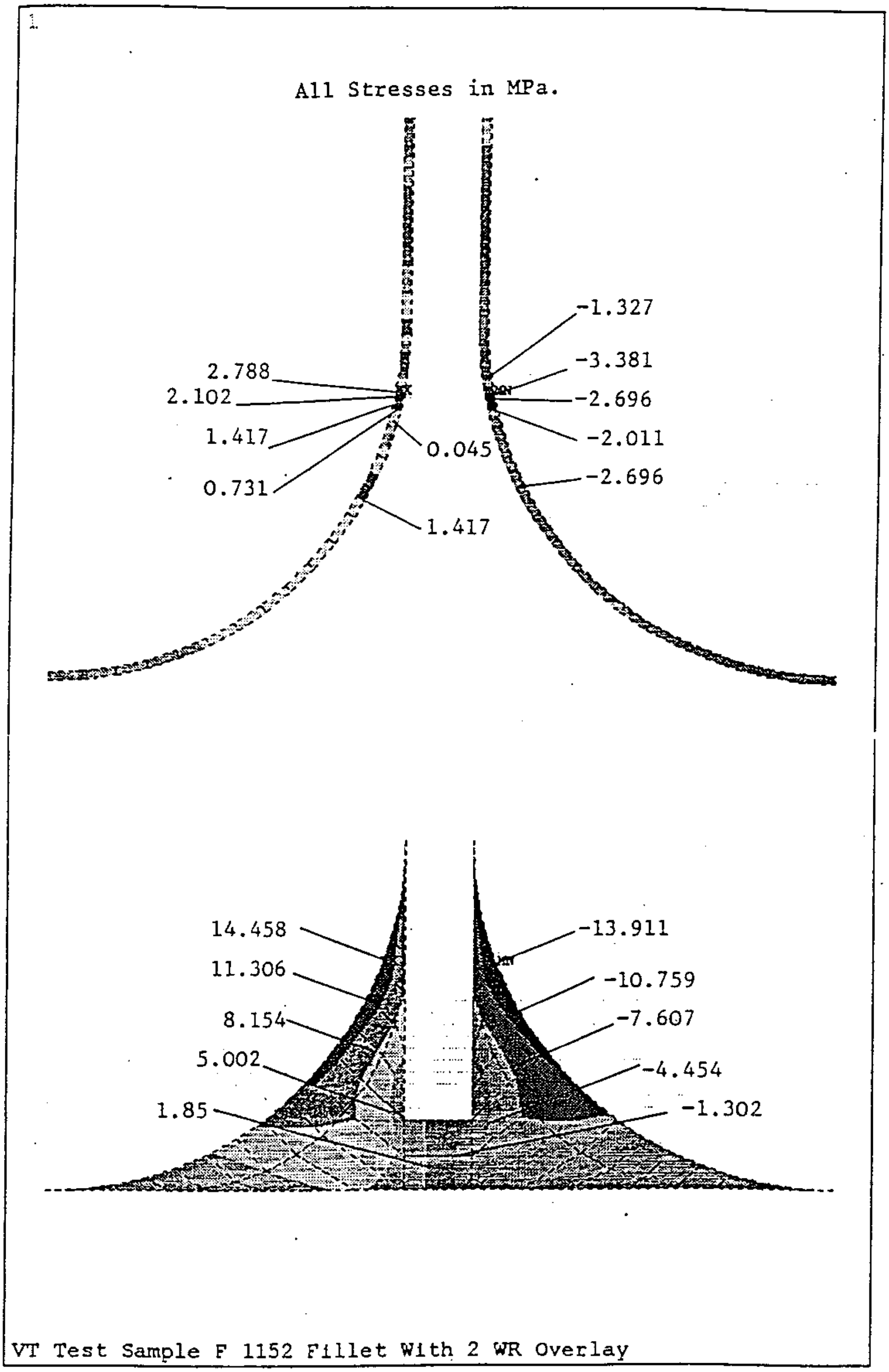


FIGURE 11 Stress Contours within Tee-Joint Sample F.

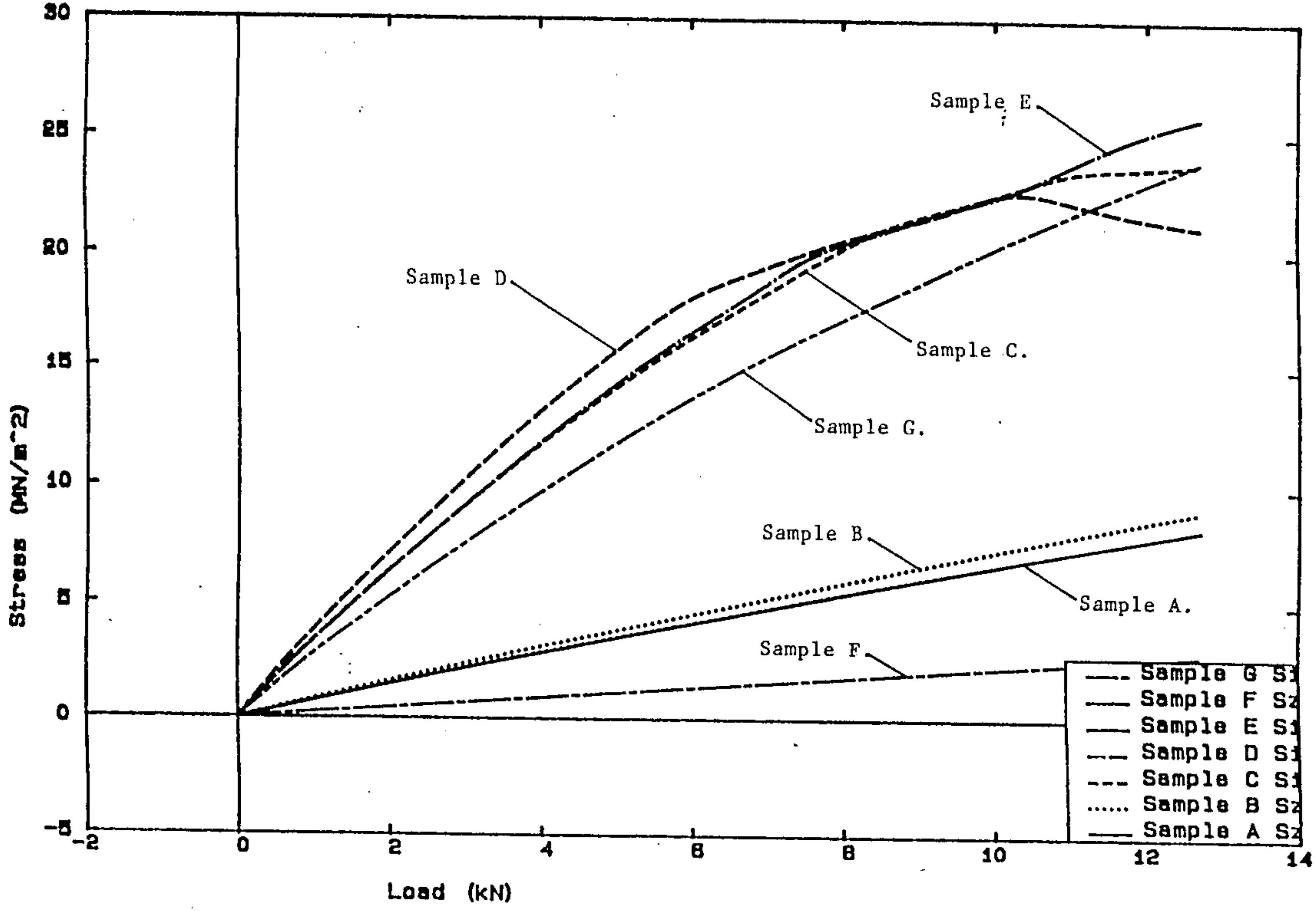


FIGURE 12 Peak Stresses in Tee-Joint versus Load.

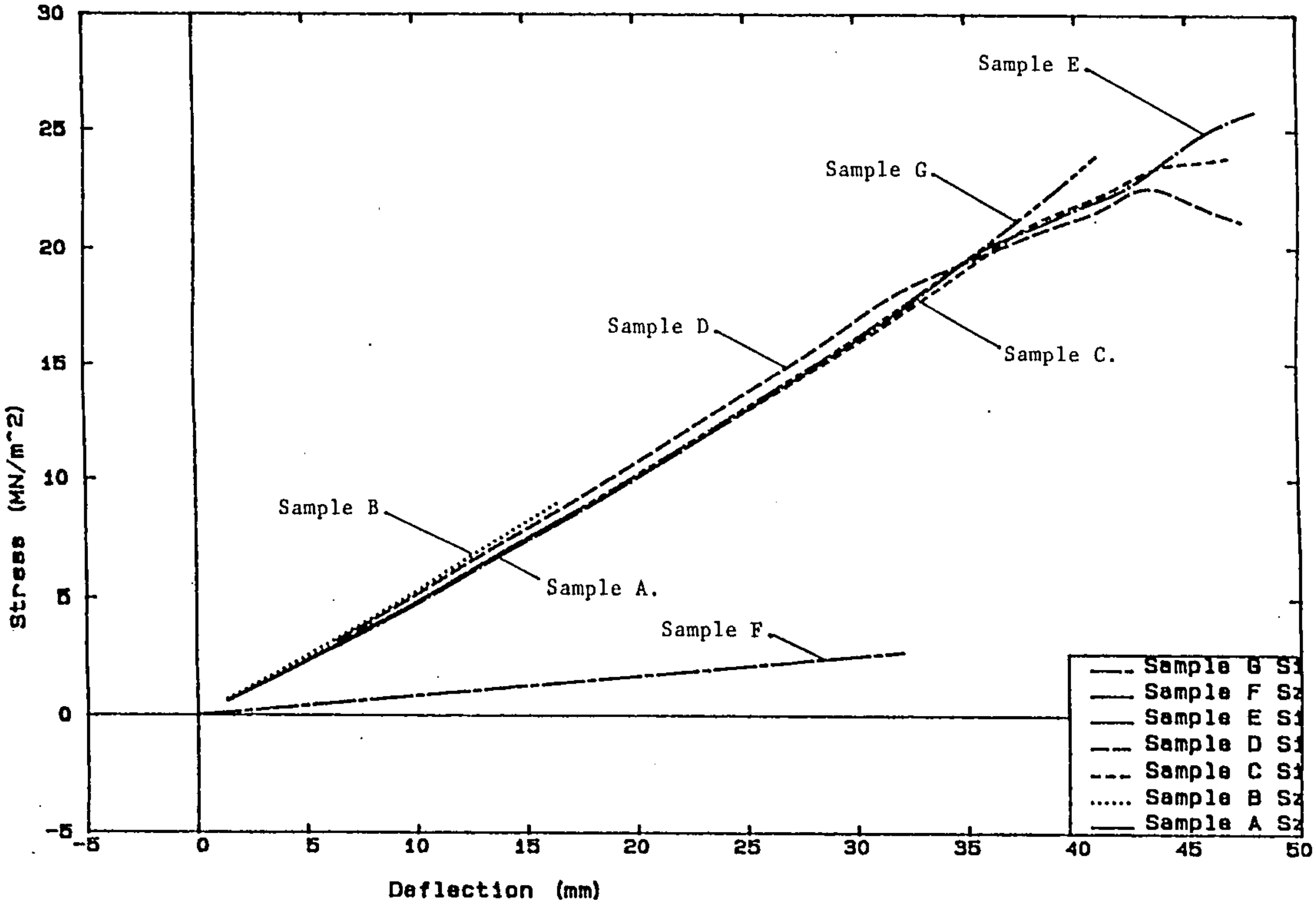


FIGURE 13 Peak Stresses in Tee-Joint versus Deflection.

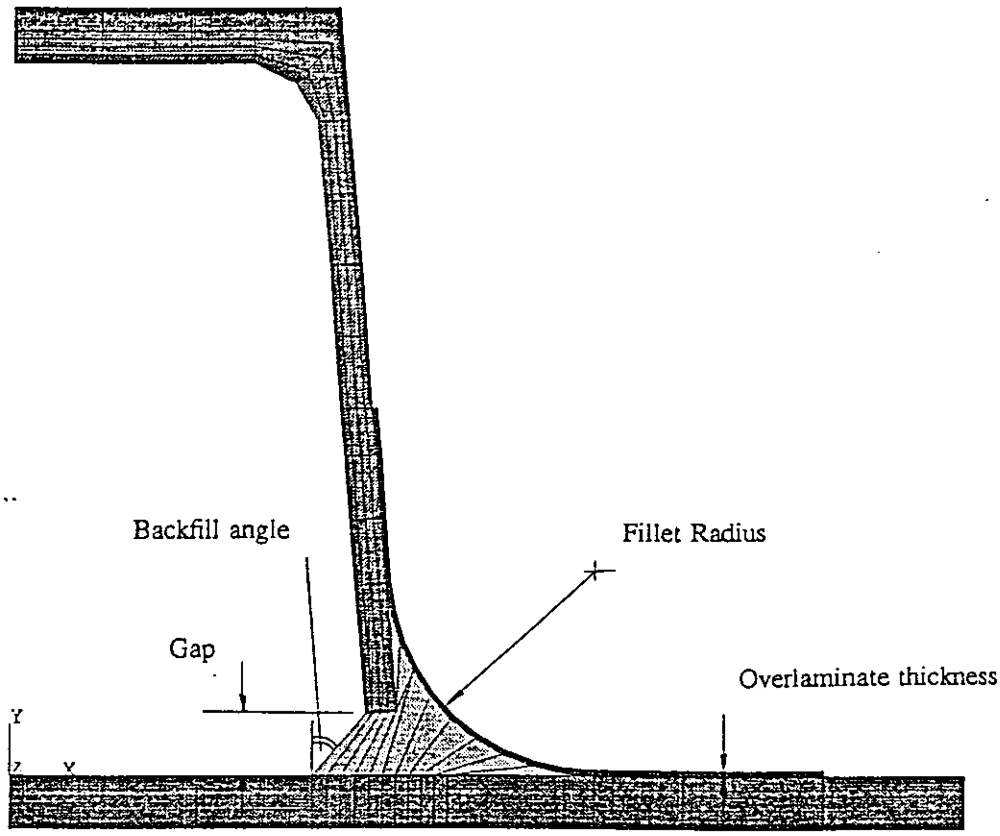


FIGURE 14 Design Variations Considered for Top-Hat Stiffeners.

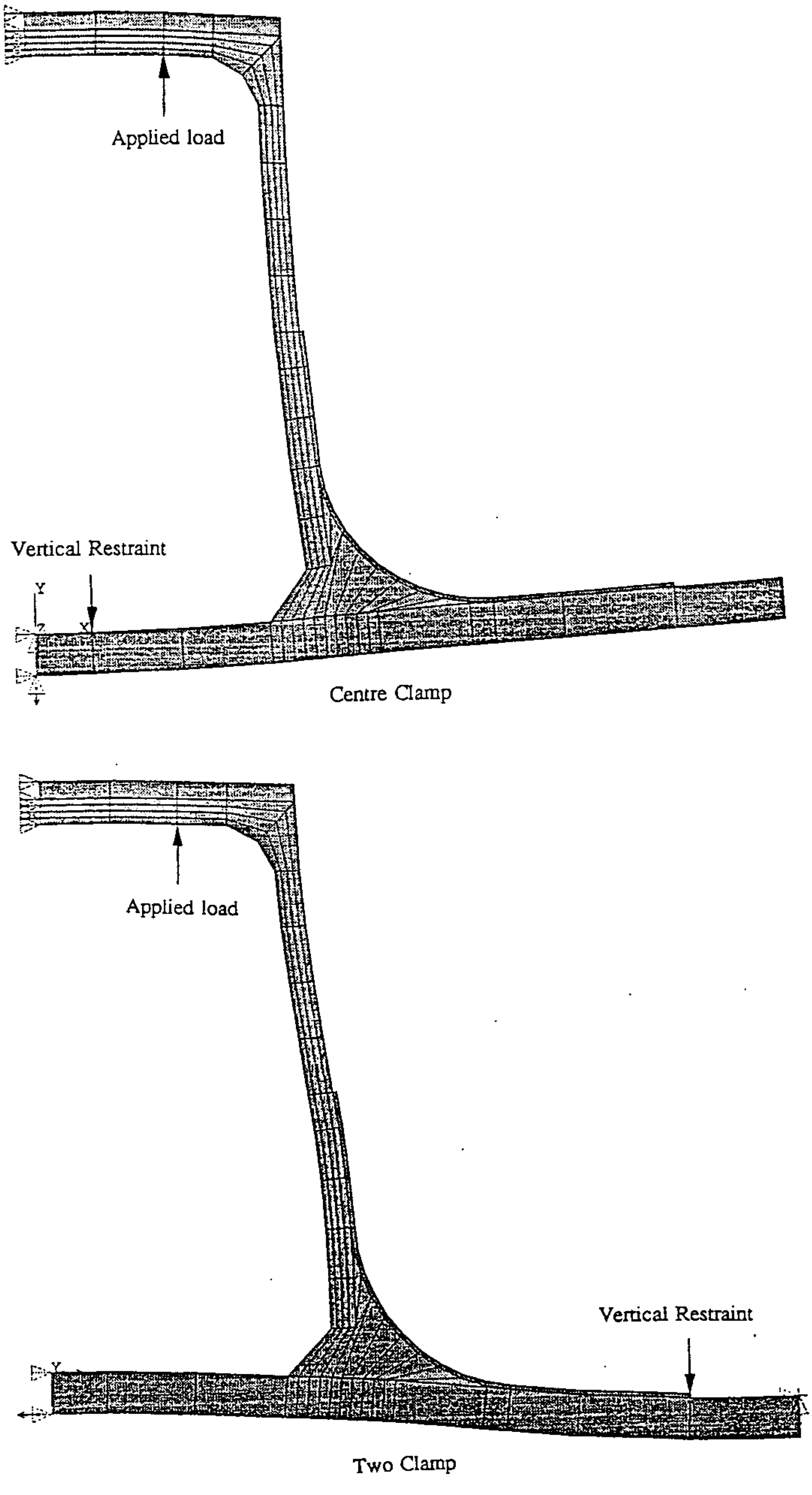


FIGURE 15 Boundary Conditions For Top-Hat Stiffeners.

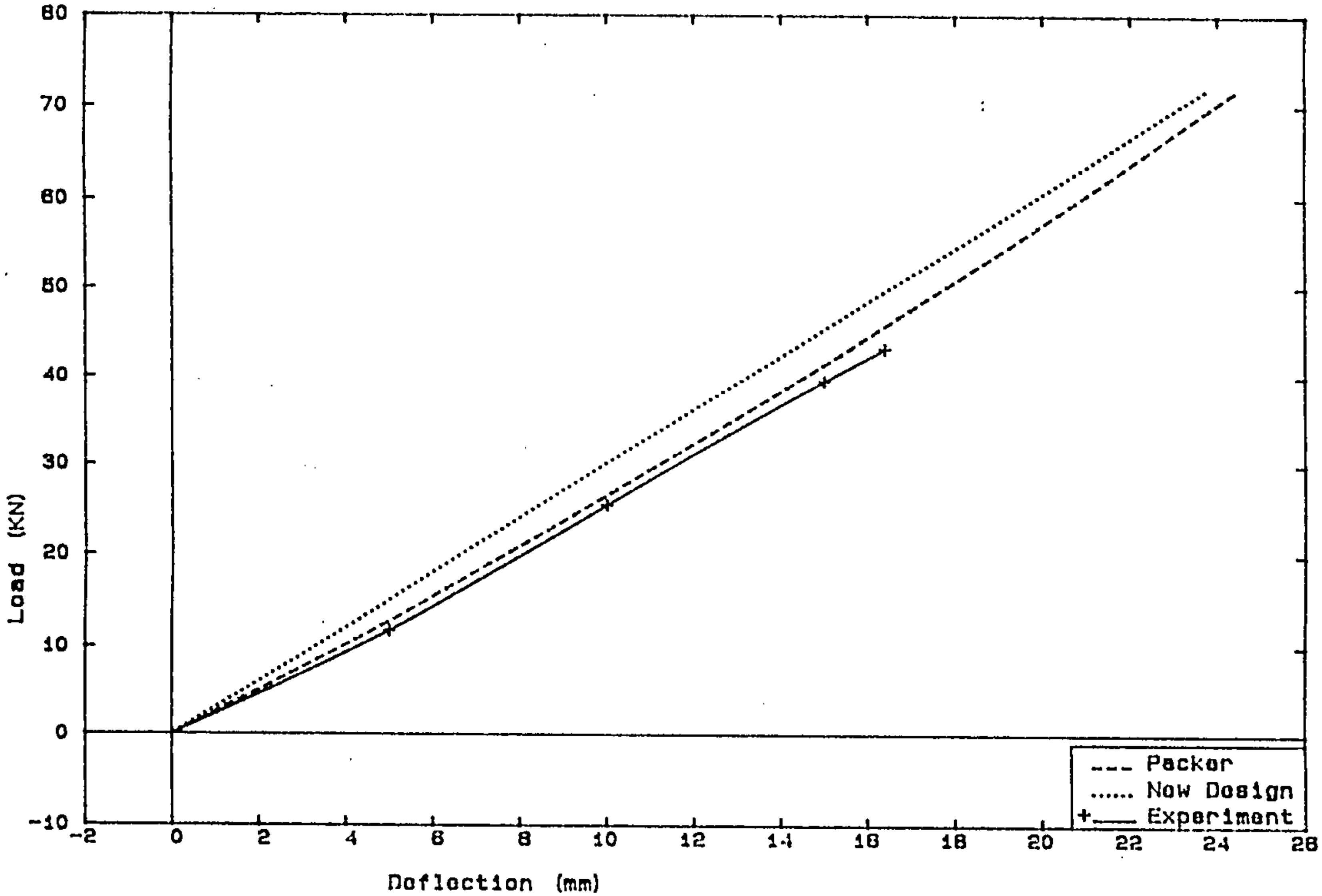
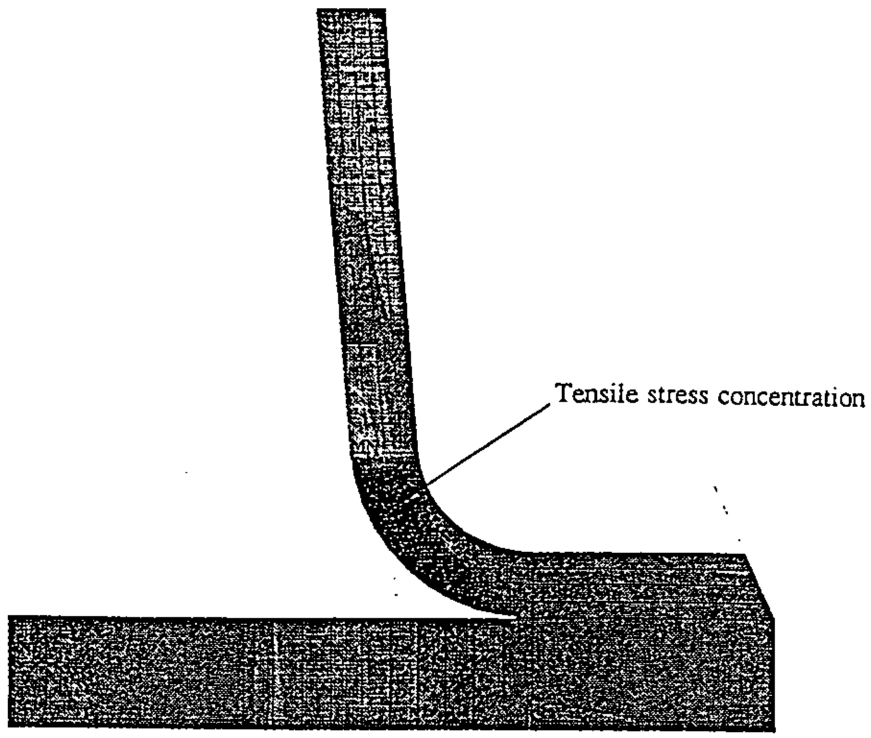
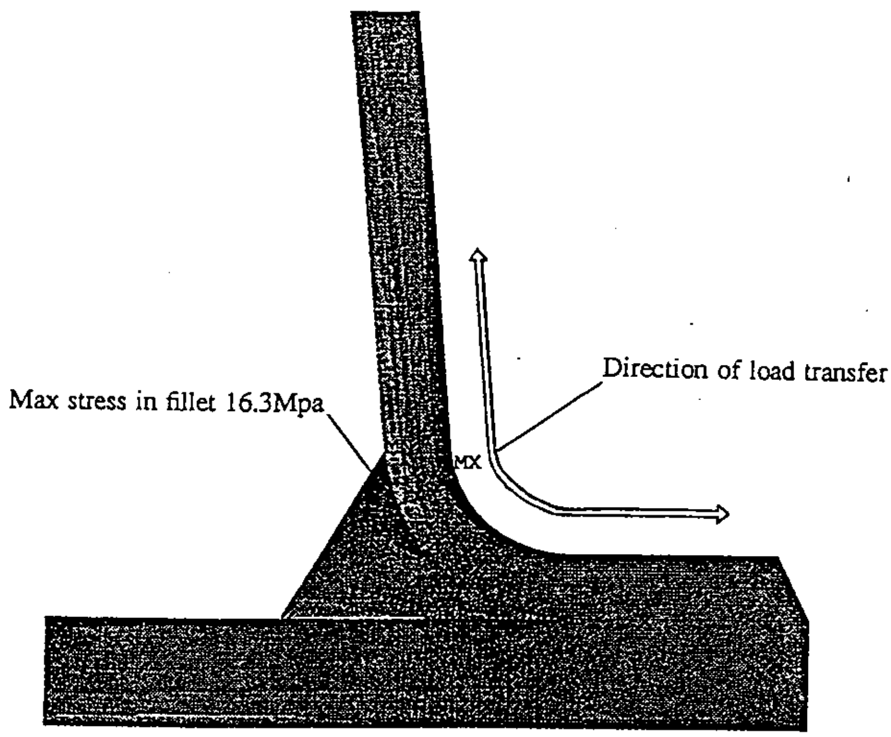


FIGURE 16 Load-Deflection Curves for Top-Hat Stiffeners.

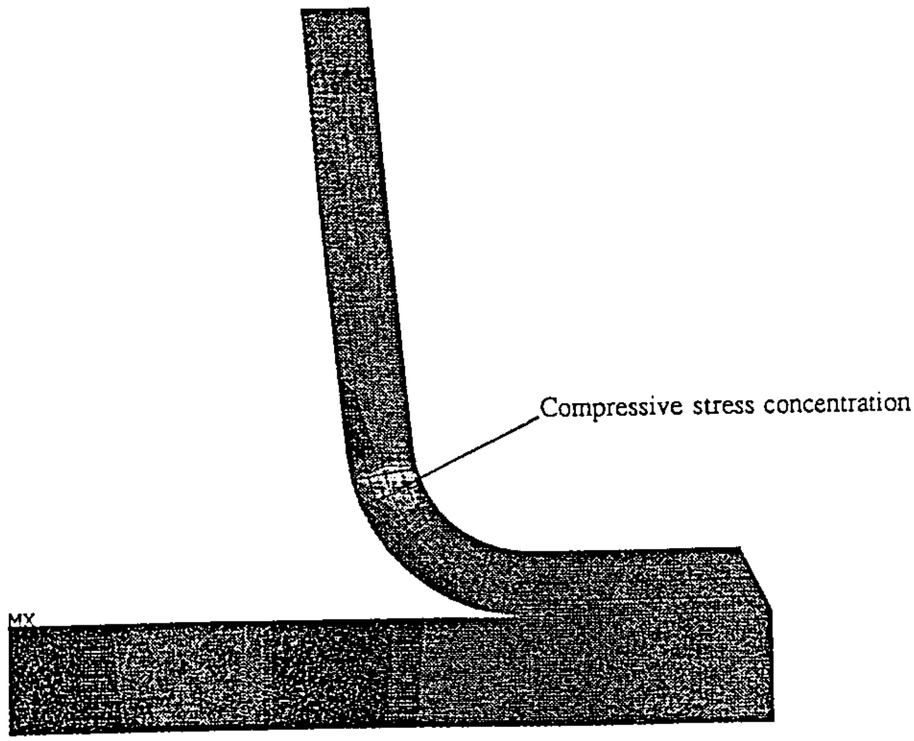


a) Through-Thickness Stress in Overlaminate

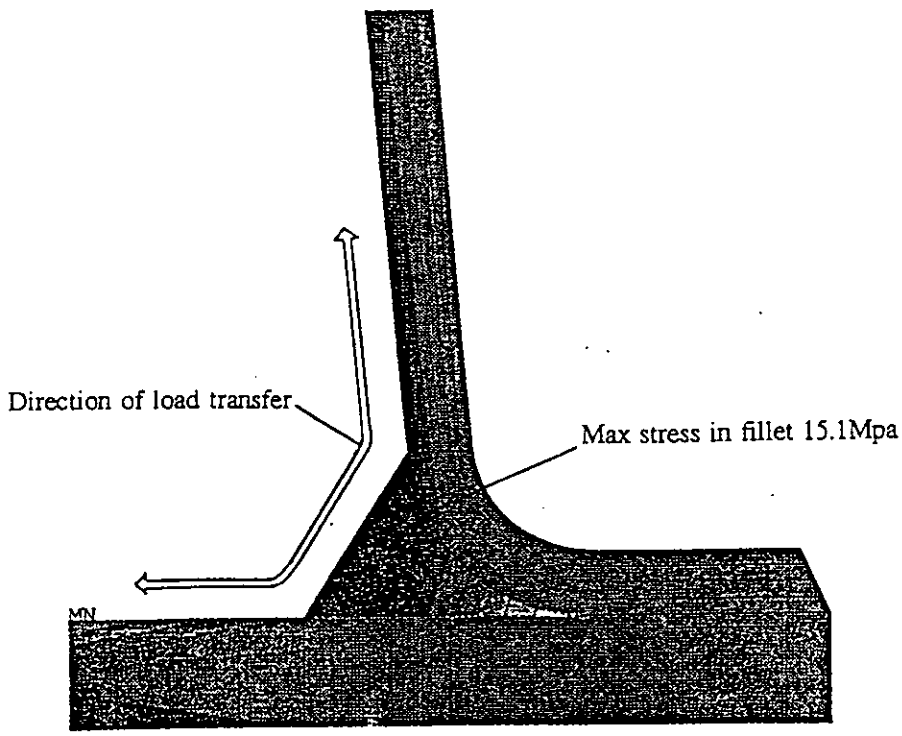


b) Inplane Stress in Overlaminate
Principal Stress in Fillet

FIGURE 17 Top-Hat Stress Pattern - Current Practice, Two Clamp.

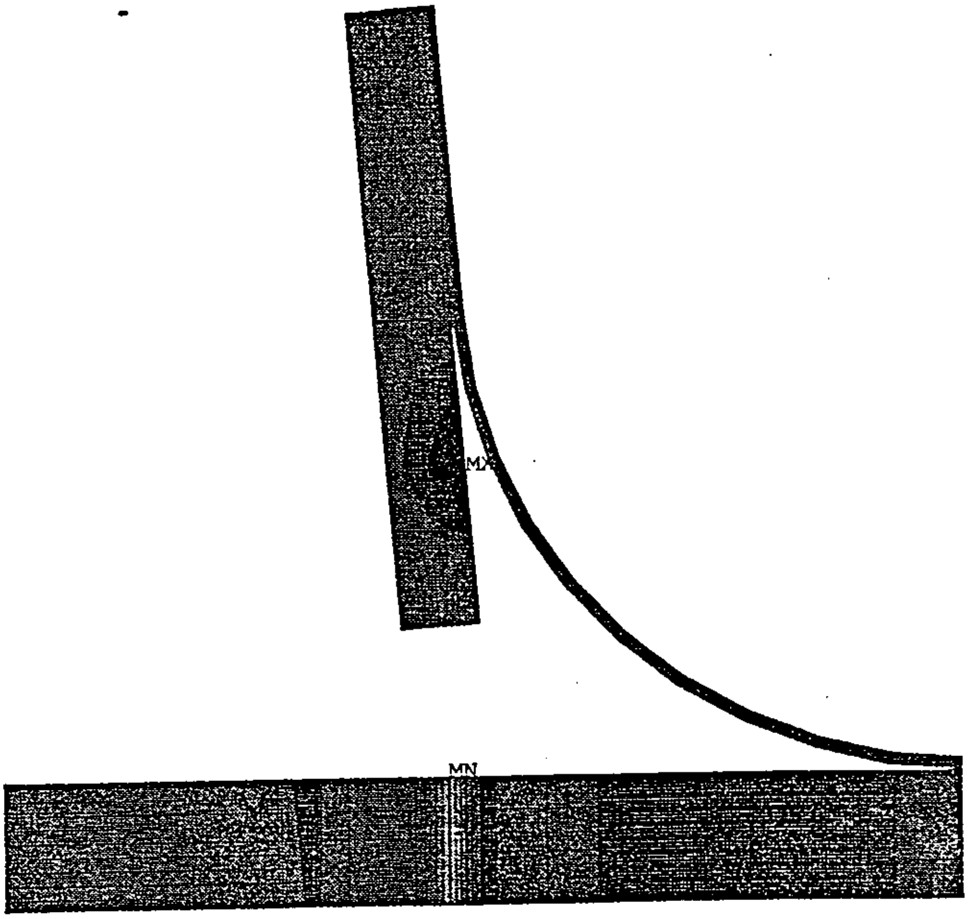


a) Through Thickness Stress in Overlaminate

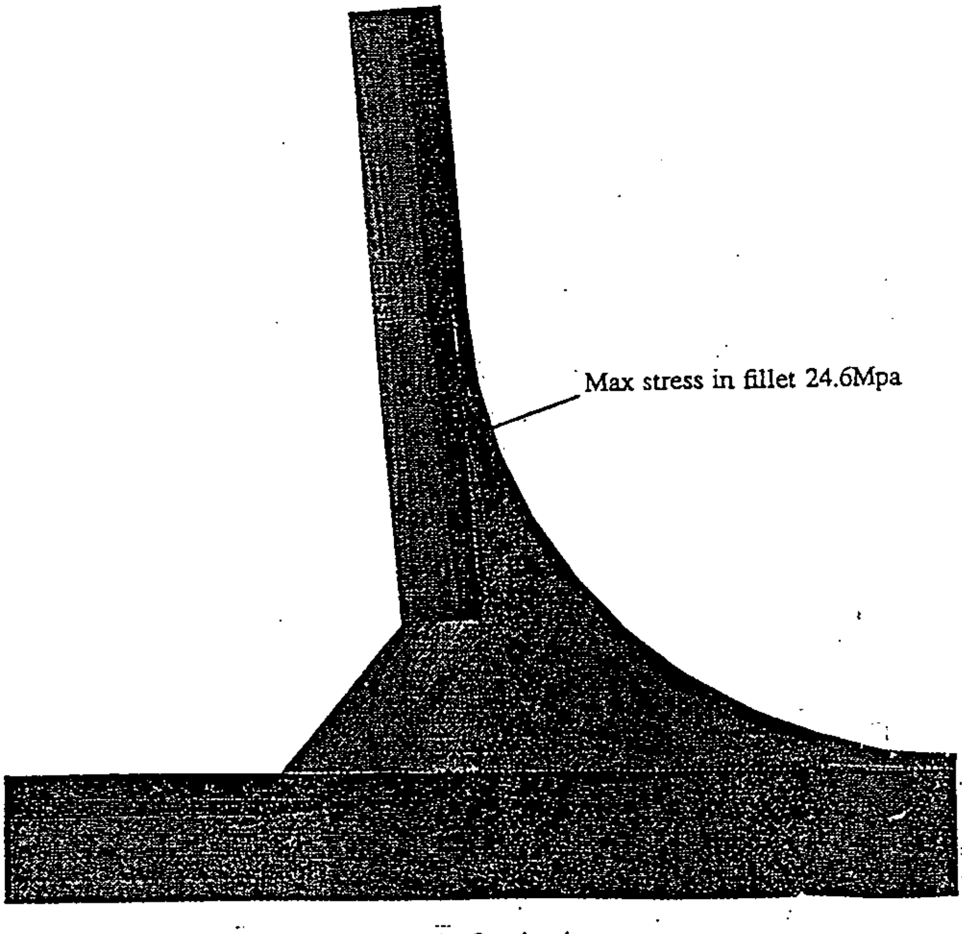


b) Inplane Stress in Overlaminate
Principal Stress in Fillet

FIGURE 18 Top-Hat Stress Pattern - Current Practice, Centre Clamp.

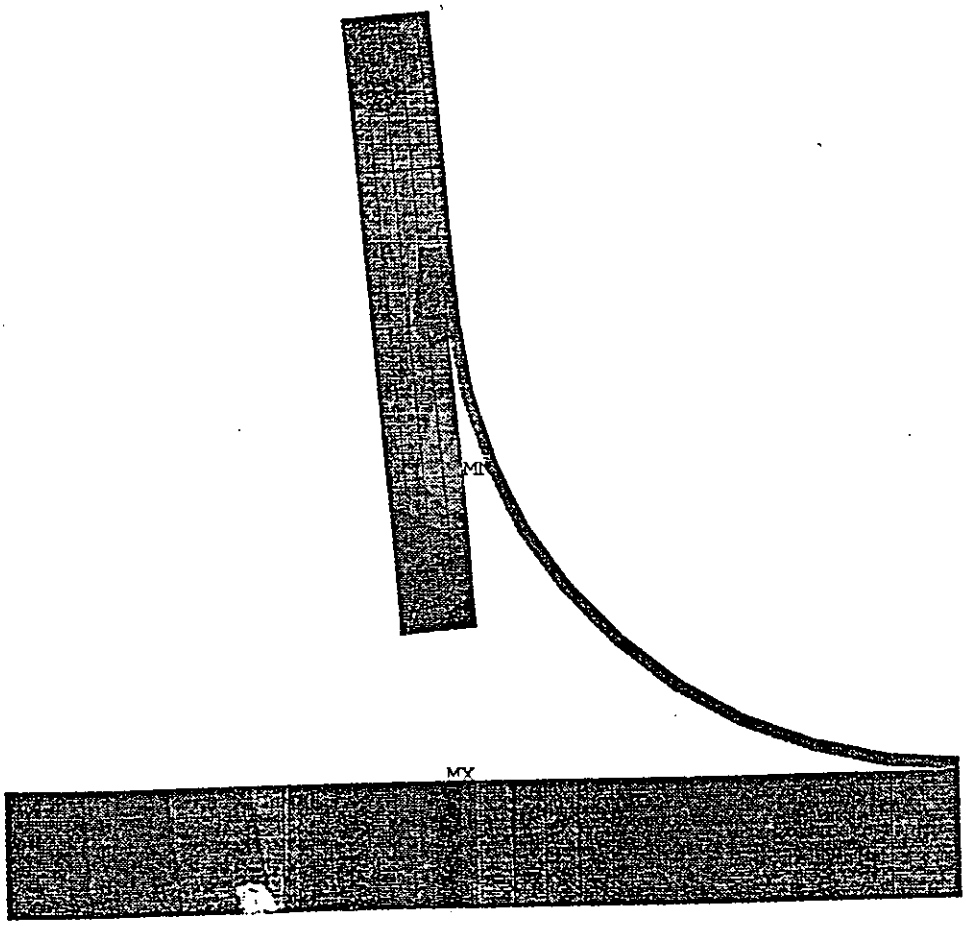


a) Through Thickness Stress in Overlaminate

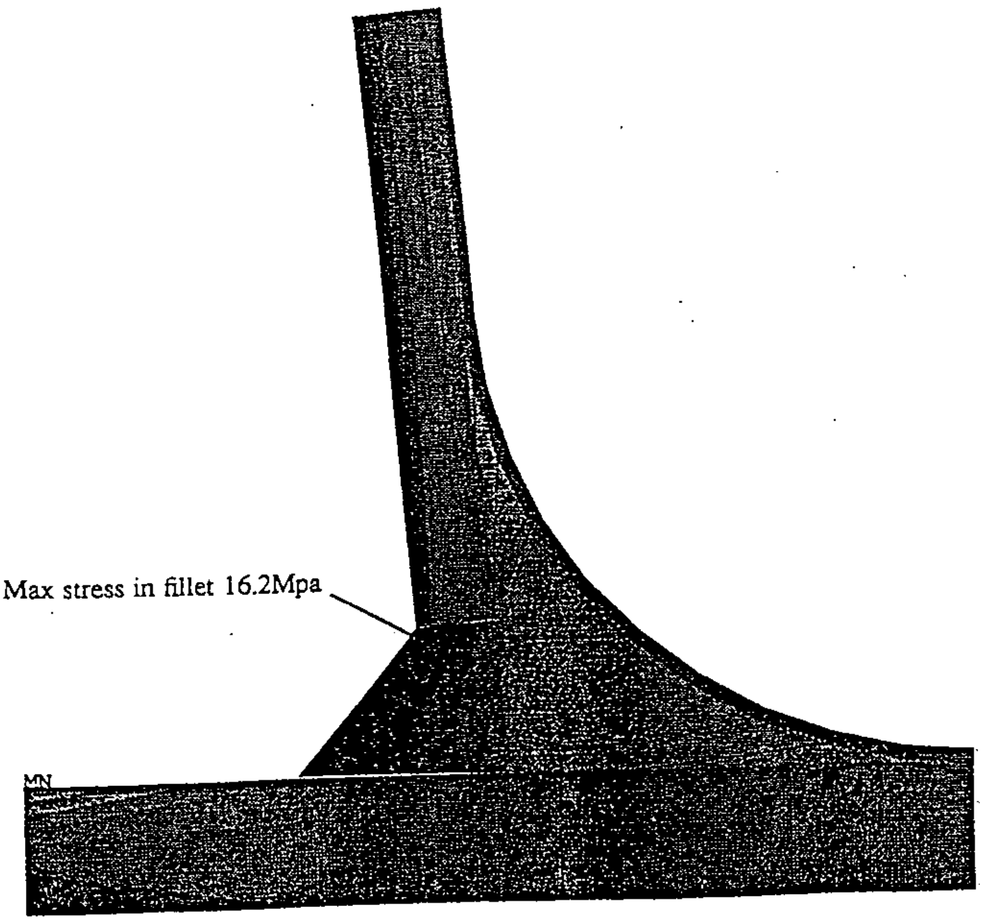


b) Inplane Stress in Overlaminate
Principal Stress in Fillet

FIGURE 19 Top-Hat Stress Pattern - New Design, Two Clamp.



a) Through Thickness Stress in Overlaminate



b) Inplane Stress in Overlaminate
Principal Stress in Fillet

FIGURE 20 Top-Hat Stress Pattern - New Design, Centre Clamp.

5 PRACTICAL DESIGN OF JOINTS AND ATTACHMENTS

5.1 BACKGROUND

5.1.1 Need for Joints

Joints become necessary in a structure for three main reasons. These relate to production/processing restrictions, the need to gain access within the structure during its working life, and repair of the original structure.

Typical production/processing restrictions arise because of the need for:

i. Large and complex structures which cannot be formed in one process thereby needing several components to be joined 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.

ii. 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.

Considering access and repair considerations, if components within the structure require regular servicing then 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 jointing method can be considered to be permanent.

5.1.2 Requirements

To fulfil its role a joint must meet one major requirement; the integrity of the overall structure must not be impaired by the presence of the joint. "Structural integrity" can be defined in several ways, depending on the particular application, and can include one or more of the following:

a) Strength - in tensile or compressive, shear or through-thickness directions. A joint must be at least as strong as the surrounding structure.
b) Stiffness or flexibility - if there is a differential in stiffness between a joint and the surrounding structure then stress concentrations will occur in the joint, the surrounding structure, or both, depending on the particular geometry and loading.
c) Water- (or air-) tightness - if the purpose of the structure is to retain (or prevent the ingress of) a fluid of one sort of another then obviously any joints on the surface of the structure need to be equally secure.

Another requirement which needs to be considered at the design stage is the economics of producing the joint. In large and complex structures the joints can constitute a significant proportion of structure weight and their manufacture can be expensive. It is necessary to ensure that material and labour requirements are kept to a minimum and the technology used to produce the joint is compatible with that used elsewhere in the structure. Care should be taken to balance the economic advantages of reduced weight against the possible additional costs incurred in producing a high performance joint.

5.1.3 Types of Joints

The range of joints used in both single skin and sandwich structures can be split into two basic types, namely in-plane and out-of-plane joints.

A. In-plane or Butt Joints:
   These are used whenever two plate elements need to be joined. As shown in figure 5.1, these can be either scarfed joints or lapped joints. Scarfed joints are adhesively bonded. They can be symmetric or asymmetric and can be used to bond sandwich panels providing the skins are sufficiently thick. Stepped-lap-joints may also be considered in this category. Lapped joints

may be bolted, adhesively bonded, or both. They can be single- or double-lap, and can be used to bond sandwich panels, especially when the skins are thin.

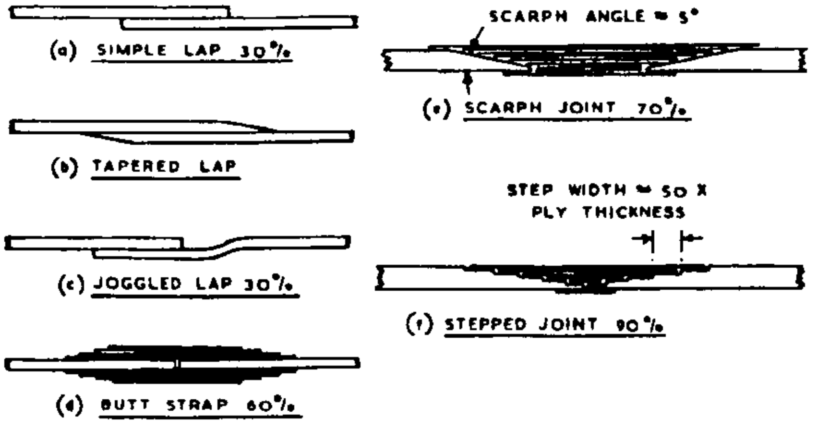


Figure 5.1. Typical arrangements and efficiencies of in-plane joints.

B. Out-of-plane Joints:
   Typical examples are shown in figure 5.2 and include frame-to-shell connections and bulkhead-to-shell connections. The former mainly employs top hat stiffeners which are usually moulded in situ onto the cured shell. It is also concerned with extruded/pultruded section stiffeners which are preformed and bonded in place using adhesives and/or boundary angles. Bulkhead-to-shell connections differ from the top hat stiffener frame-to-shell connection in that both elements are formed before the joint is fabricated, and access is available from both sides of the joint.

5.1.4 Bonded vs Bolted Connections

The choice depends very much on the application being considered. A bonded connection provides a greater surface area to transmit load. This ensures that all fibres at the joint interface are used to carry load so that stress concentrations are reduced. They are cheaper and easier to produce and can be formed from one side of the panel. However, some environmental control is usually required during the construction process. Another shortcoming is that when initial failure occurs in a purely bonded joint it can propagate easily since there are no fibres

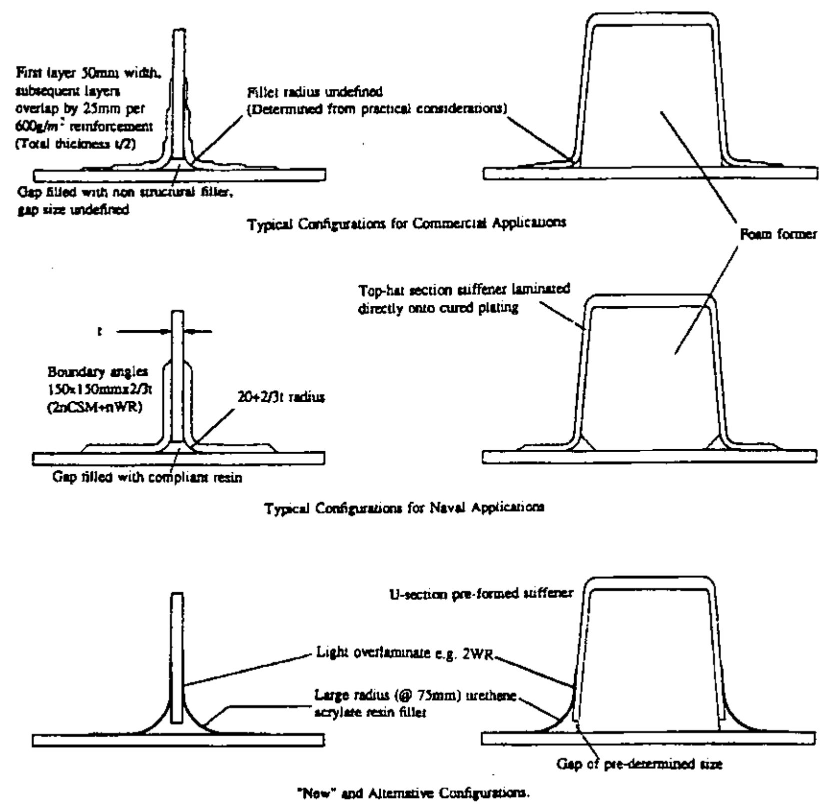


Figure 5.2. Typical configurations of bulkhead-to-shell joints and top hat stiffeners.

across the joint to act as crack arrestors. Such joints are permanent and cannot easily be removed.

Bolted connections provide a strong link across the joint interface; they are easily removed, and can usually be formed under adverse environmental conditions. When used in conjunction with an adhesive, the bolts can act as crack arrestors in the event of initial failure. However, since the load is transmitted through a small area, stress concentrations occur that can lead to early initial failure. They require access from both sides of the joint, are heavy and can be expensive to produce.

The performance of both bonded and bolted joints is sensitive to the lay-up of the composites being joined, particularly if they are formed using a large proportion of unidirectional fibres. This must be considered when using single-lap or stepped-lap-joints and, to a lesser degree, other forms. Typical marine composites contain high proportions of woven or randomly oriented fibres, which are less sensitive to the jointing method used.

For bonded joints, the geometry should be arranged so that the layer at the joint interface contains fibres that are oriented in the direction of the local loads. This can usually be arranged and thus is not an initial design consideration.

For bolted joints, the combination of layers through the thickness of the laminate should ideally be arranged so that the full bearing strength of the laminate is achieved before pull-out or pull-through. This needs to be considered at the initial design stage.

5.2 IN-PLANE OR BUTT JOINTS

5.2.1 Features and Purpose

As mentioned above, these joints are used to join structural elements together, attach removable panels to the main structure and make repairs to a structure. They are used in two basic forms, namely scarfed and lapped. The decision as to which form to use depends upon the particular application and the thickness of the adherands being joined. The relative advantages and disadvantages of different joint types are summarised in table 5.1.

Sandwich panels generally have thinner skins which can make scarfing much more awkward. However, even when the panels are in bending, the loading on the skins remains in-plane, as the panels are much stiffer than the individual skins. So the bending and stress concentration effects associated with lapped joints are much reduced.

A considerable amount of work has been carried out on the analysis of butt joints in the past, both experimentally and theoretically, for the case of in-plane loading. This work has been thoroughly reviewed elsewhere [2,3] so that task will not be repeated here. Only the influence of different design variables and analysis methods are briefly outlined in the following three sections.

5.2.3 Design Variables

Assuming that the magnitude and mode of loading is known, a series of design variables can be identified for any given joint configuration. Firstly, considering the structural elements being joined, the variables that need to be considered are:

a) Fibre type and form

Table 5.1. Summary on in-plane joint types.

Joint Type	Advantages	Disadvantages
Symmetric scarf	Best in flexure Good in tension Uses less material Similar flexibility to original structure	Expensive in labour Difficult on thin laminates Needs access from both sides Permanent Joint
Asymmetric scarf	Good in flexure Good in tension Needs access from on side only Similar flexibility to original structure	Expensive in labour Difficult on thin laminates Uses more material Permanent Joint
Double-lap	OK in flexure Good in tension Easy on thin laminates Cheaper in labour Removable joint if bolted	Needs access from both sides Uses more material Stiffer than original structure
Single-lap	Needs access from one side only Cheapest in labour Easy on thin laminates Removable joint if bolted	Poor in flexure Poor in tension Uses more material Stiffer than original structure

b) Resin type
c) Fibre orientation/stacking sequence
d) Laminate material properties (moduli, UTS, shear strength, etc)
e) Laminate thickness

Next, considering the type of connection used, for bolted connections design variables include (figure 5.3):

a) Bolt diameter
b) Hole size and tolerance
c) Clamping force
d) Washer size
e) Pitch of holes
f) End distance
g) Side distance
h) Back pitch if several rows are used
i) Joint type (single or double overlap)

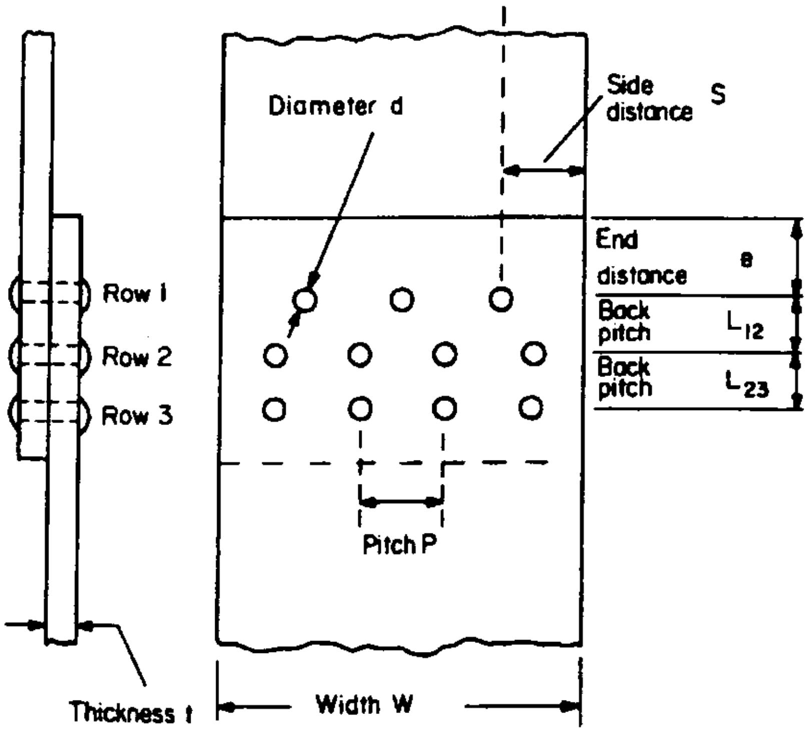


Figure 5.3. Definition of design variables for bolted joints.

For bonded connections two further subsets of variables exist, namely those that apply to scarfed or stepped-lap-joints and those that apply to lapped joints. The design variables which apply to scarfed or stepped-lap-joints include:

a) Angle of scarf or length of cutback
b) Depth of cutback (influenced by the stacking sequence of the

structural elements being joined, as mentioned in section 5.1.4 above)
c) Symmetric or asymmetric joint
d) Lay-up of the joining composite

Those which apply to lapped joints include:

a) Single- or double-lap
b) Length of overlap
c) End profile of overlaps
d) Thickness of overlaps
e) Adhesive thickness
f) Adhesive material properties (strength and stress-strain characteristics)

The above lists are by no means exhaustive but are sufficiently detailed to cover most applications. Care must be taken at the design stage to account for the sometimes complex mechanisms that can arise in even the simplest of joints under different loadings. These mechanisms may require the consideration of other variables which may not appear initially to be relevant. For example, consider the case of a bonded single-lap-joint. When load is applied in-plane, the joint will deflect as shown in figure 5.4. This results in a complex stress

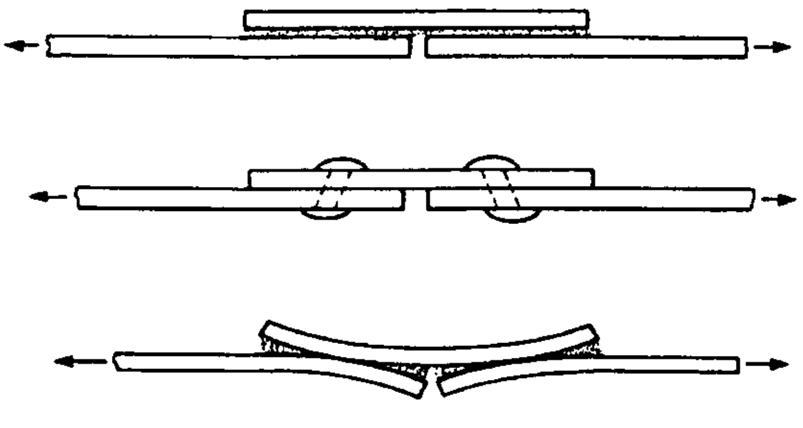


Figure 5.4. Deflection characteristics of a single-lap joint.

state that includes through thickness tensile stresses in the adhesive and the adherands, shear in both the longitudinal and through thickness directions, and longitudinal stress in the adherands. The stresses in the adhesive vary through the thickness. This reinforces the need to have well identified analytical and modelling techniques which are able to cater for such mechanisms.

Clearly, if a modelling technique is to account for all the above mechanisms and design variables it will be complex and cumbersome, and thus unsuitable for use in a design context. Many of the out-of-plane material properties that are required for these techniques are unlikely to be available for a given laminate, necessitating an extensive materials testing programme prior to design. This is impractical in most circumstances. Hence it is necessary to eliminate many of the variables when considering a particular problem. In previous work, the behaviour of a few basic joints in a limited range of materials has been analysed and the influence of the more important variables has been determined. These results can be used in a qualitative sense to predict the behaviour of other joints.

5.2.4 Modelling Techniques for Bolted Joints

Consider the case of a bolted single-lap-joint loaded in-plane. A simple modelling technique is to take the bearing stress of the composite (σ_b), its thickness (t), and the diameter (d) and number of bolts (n) used;

$$\text{Load (P)} = \sigma_b \; t \; d \; n. \qquad \text{For a unit width Capability of joint.} \qquad (5.1)$$

Generally, the load capability and thickness of the composite being joined will have already been determined from other structural considerations, at least on a preliminary basis. If the criterion is to reduce the number of bolts used, it is necessary to achieve as high a bearing stress and as large a diameter as possible. These variables are, however, linked, both with each other and with other design variables. Figure 5.5 shows that bearing stress reduces as d/t increases if no lateral restraint is applied to the joint, and remains constant if a lateral restraint is applied. It is further suggested that d/t should not exceed 1 if full bearing strength is to be achieved. When considering small diameter bolts the shear strength of the fastener must be considered. Reference [6] provides data sheets on fastener performance in several applications but the simplest formula would be;

$$\text{Shear Strength of Fastener} = P \; n \; \pi \; (d/2)^2 \qquad (5.2)$$

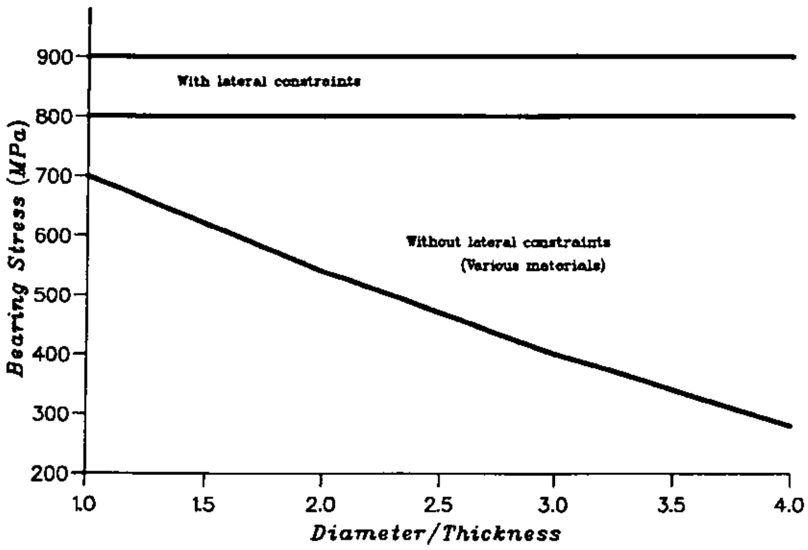


Figure 5.5. The effect of d/t ratio on bearing strength (Data from [5]).

Thus an upper and lower limit on d can be arrived at. All that remains is to ascertain the achievable bearing stress for the given joint. Unfortunately this is rarely found in material data since it is so dependent on geometrical variables. Maximum values for bolted connections have been quoted [2] as being between 800 and 930 MPa for CFRP, 550-700 MPa for unidirectional GRP, and 200 - 600 MPa for woven or CSM GRP. Generally, failure can occur in three ways by shear, bearing, or tension. Figure 5.6, for example, shows the relationship between bearing stress and lay-up in CFRP. Quasi-isotropic laminates will fail in bearing mode at the highest loads. Other factors affecting bearing strength are clamping pressure, joint width or pitch, end distance, and load direction. Curves similar to figure 5.6 can be found in reference [2] for these factors. Stacking sequence has little effect on the bearing strength of bolted joints if the plies are homogeneously mixed through the laminate. If the laminate is blocked (i.e. all layers of a given orientation are grouped together) significant reductions of up to 50% can result [7]. It is also important that the bolt is a good fit in both the hole and any washers used.

A reasonable prediction of bearing strength can be made by considering the effects of the criteria mentioned above. By completing a simple iterative process using the two equations, the joint parameters can be derived, having accounted for the laminate properties (type,

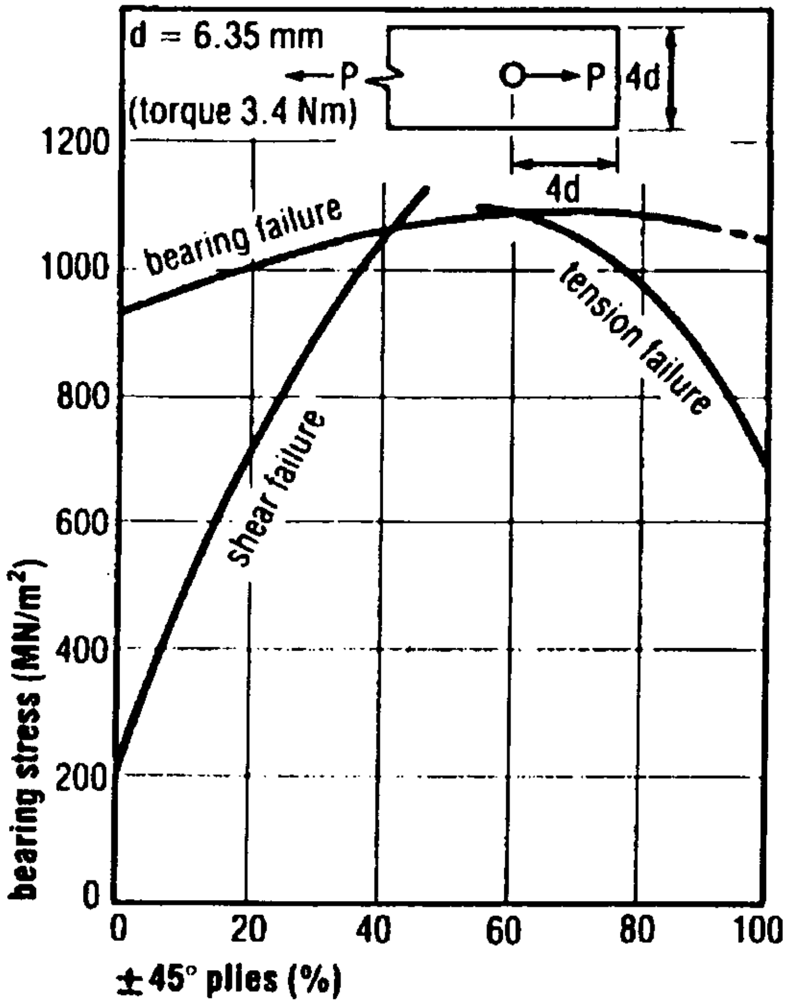


Figure 5.6. Influence of fibre-orientation on failure mode of bolted joints in 0/+-45 CFRP. (From [4]).

orientation, material properties, thickness), bolt diameter, clamping force, pitch of holes, end distance, and side distance. It should be borne in mind that the relationships illustrated above are derived experimentally from in-plane loading and, whilst they do account for out-of-plane deflections, care must be taken when extrapolating any results from one configuration to another.

The same principle can be applied to a double-lap bolted joint,

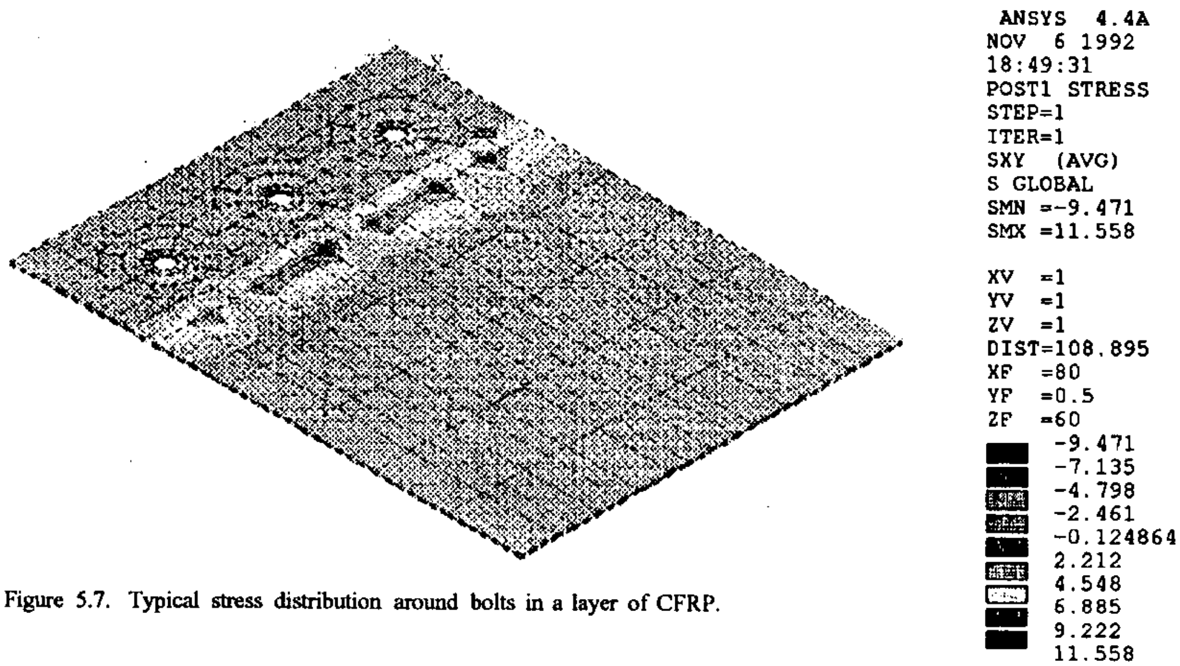


Figure 5.7. Typical stress distribution around bolts in a layer of CFRP.

since each lap is usually half the thickness of those in a single-lap-joint. The reduced out-of-plane deflections in a double-lap-joint means that failure is more likely to occur in bearing or tension and hence this is a better jointing method particularly for highly anisotropic laminates. One change to the above equations is that the shear stress in the fastener is half that found in the single-lap-joint, and hence a different optimum arrangement will be arrived at.

If greater detail is required or more complex out-of-plane loading is anticipated, then it may be necessary to carry out a numerical analysis of the joint using, for example, the finite element method. This method allows a layer by layer analysis of the stress distribution within a joint. Many commercial packages are available and, with modern graphical interfaces, are as quick and easy to use as the classical analytical methods without being constrained by the assumptions these methods make. Figure 5.7 shows a typical stress contour plot of a bolted single-lap-joint being subjected to flexure. Extensive experimental and numerical analysis in CFRP [8] has shown the performance of the joint to be sensitive to stacking sequence, fastener diameter, pitch, and clamping pressure. These variations are non-linear in nature and it is recommended that each problem of this type is analysed independently.

5.2.5 Modelling of Adhesively Bonded Joints

Several classical analytical methods exist for analysing bonded joints between isotropic plates loaded in tension [3]. When applying these methods to composites care must taken to account for the anisotropic nature of the adherands and the variation in modulus through the thickness. Consideration should also be given to thermal loadings resulting from the curing cycles. Generally, the simpler the analytical method, the less accurate it will be when applied to single-lap-joints due to the out-of-plane displacements described in section 5.2.3 above. However, for the scarfed, stepped-lap, and double-lap joints where these out-of-plane displacements are less significant, reasonable correlation between analytical results and experiments can be achieved [9-11] providing any non-linear behaviour of the adhesive is accounted for.

For practical design purposes, however, even the simplest shear lag theory is impractically complex and of limited application since the assumed boundary conditions are not modelled in real life. Of much greater use to the designer are the guidelines produced by various classification societies and other sources [12-16], and the results of parametric studies that have been completed. Significant features are outlined below:

a) The stresses in the adhesive reduce and become more uniform as the adhesive modulus is reduced, the adherand stiffness is increased and the overlap length is increased.
b) Tapering the ends of the adherands on double-lap-joints and including an adhesive spew fillet on single-lap-oints reduces the maximum shear and direct stresses in the adhesive.
c) The adherands at the joint interface should be identical if possible. If this is not possible then the in-plane and bending stiffnesses should be matched.
d) If a single-lap-joint is restrained to prevent rotation, and hence the formation of peel stresses (e.g. on sandwich skins), its strength is improved and thus it can be applied to thicker adherands. Similar improvements can be made if a through-thickness clamping force is applied at the ends of the joint (see figure 5.8).

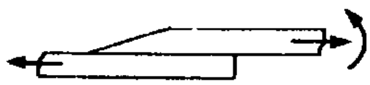
FM-400 bonded joints (width = 1.00 inch)	failure mode	joint eff
 control	peel and shear failure	0.52
 one rivet	shear and patch net tension failure	0.73
 undercut and one rivet	shear and rivet head pull-through failure	0.78

Figure 5.8. The effect of detailed design in reducing peel stresses.

e) Plies at the joint interface should be aligned in the direction of the applied load where possible.
f) The length of overlap in a double-lap or stepped-lap joint must be long enough to allow the "elastic trough" to develop (see

figure 5.9). This is the name given to the region where the shear stress in the adhesive is below its elastic limit. Any increase above this results in little increase in strength.

g) The adherands in a scarf joint should ideally have knife edges. Finite tip thicknesses cause strength reductions of up to 25%.

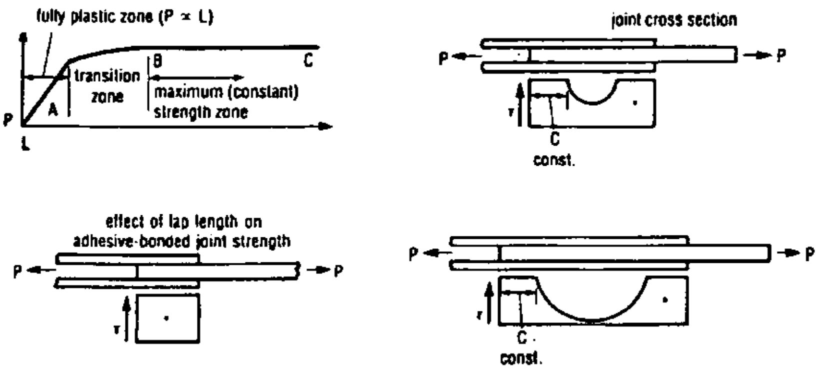


Figure 5.9. Effect on lap length on strength and adhesive shear stress distribution (*) of bonded double-lap joints.

The finite element method has been used extensively to analyse bonded joints. This allows analyses to be completed without the need to make the simplifying assumptions necessary for the classical analytical methods, and can be used to tackle problems that cannot be solved using classical analyses. An example of a simple plain strain analysis of a double-lap-joint is shown in figure 5.10. This illustrates the high peeling and shear stresses at the end of the lap.

5.3 OUT-OF-PLANE JOINTS I: FRAME-TO-SHELL CONNECTIONS

5.3.1 Types of Connections

As mentioned in section 5.1.3 above, the principal type of stiffener used in marine composite structures is of a top hat configuration. It is usually formed in situ over a former onto the cured panel or shell. This imposes some limitations on the design of the joint between the stiffener and the panel (see section 5.3.2 below). Nevertheless several different designs have been developed, principally in an attempt to prevent stiffener debonding under extreme loads. One recent development being considered [17] is that of preforming top hat sections and bonding these cured sections onto the shell. The major

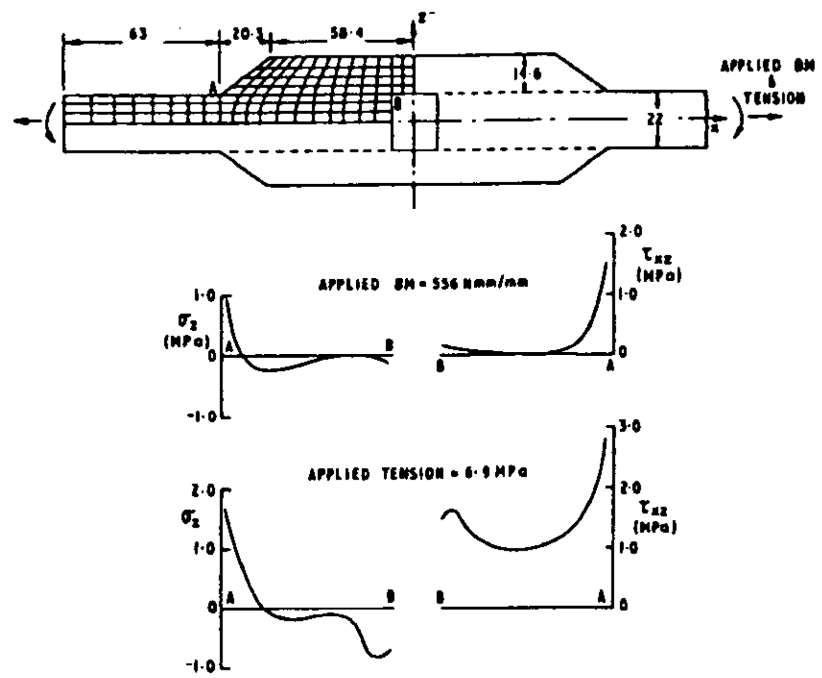


Figure 5.10. Stress analysis of a double-lap joint.

benefits of this development are much reduced production costs and greater flexibility in the design of the joint.

5.3.2 Design Variables

Taking a typical example of a top hat joint, given in figure 5.2, the variables available to the designer are:

a) Radius of fillet
b) Backfill angle of fillet
c) Fillet material
d) Thickness of overlaminate
e) Length of overlap of overlaminate
f) Lay-up of overlaminate
g) Overlaminate resin.
h) Number, position, and make-up of additional plies
i) Number, position, and size of reinforcing bolts

However, the current practice of forming the stiffeners in place means that the variables associated with the overlaminate (namely items d, f and g above) are fixed since the overlaminate around the joint is just an extension of the web of the stiffener, whose dimensions are governed by global stiffener requirements. The development of preforming the top hat section removes these constraints since the overlaminate is separate from the structural elements being joined. For economic reasons additional plies and reinforcing bolts (items h and i) should be avoided if possible.

If the stiffener is preformed and bonded in place these variables remain flexible, and the gap between the stiffener and the shell becomes an additional variable. However, the ability to control the backfill angle of the fillet is lost.

5.3.3 Modelling Techniques

As far can be ascertained no analytical methods exist to model the complex behaviour of a joint of this nature. Empirically based design guidelines are available [12-16] but these consider only a limited number of variables.

A considerable amount of experimental work has been conducted, especially for naval vessels, to determine the effects of some of these design variables [17-20]. When considering this work it should be borne in mind that the driving design criterion for these vessels is that they should be able to withstand high levels of explosive loading. This loading applies high through-thickness tensile forces to the joint which will not be seen in other applications. Initial work showed that an all polyester construction was prone to early failure in this mode. By reinforcing the joint with bolts the ultimate performance is much improved but the bolts do not prevent initial failure.

Further work investigated the use of stitched cloth and the use of compliant resins at the joint interface and for the fillet material. It was found that increasing the compliance of the joint region markedly improved the resistance of the joint to initial failure. This theme has been pursued further leading to the removal of the expensive bolted reinforcement.

A stress analysis of a frame-to-shell connection was made using 2-D finite elements [21]. The length of overlap was varied as well as the addition of internal and external flanges. Several different loadings arrangements were applied. The conclusions drawn were that the magnitude and distribution of the tensile debonding stresses were very sensitive to the form of the applied load. For loading arrangements that caused high tensile stresses in the root of the joint (the point of initial failure observed in experiments) only the internal flange

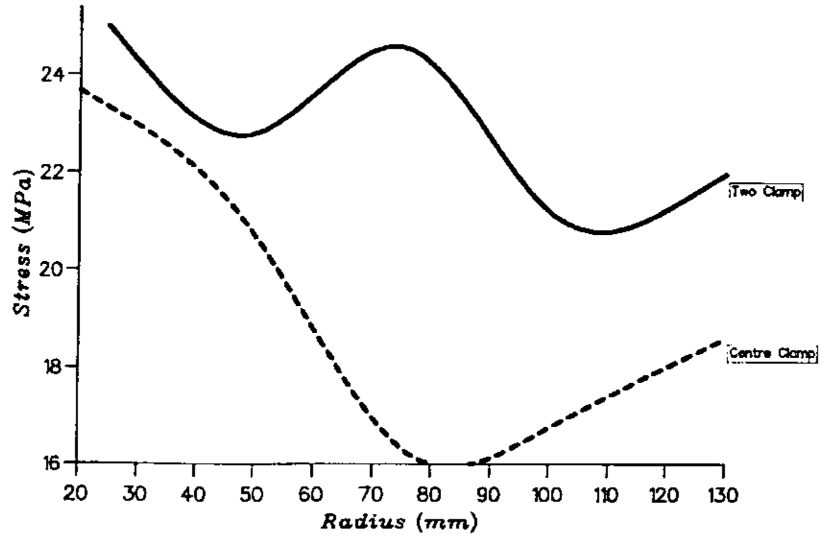


Figure 5.11. Effect of radius on stress in fillet.

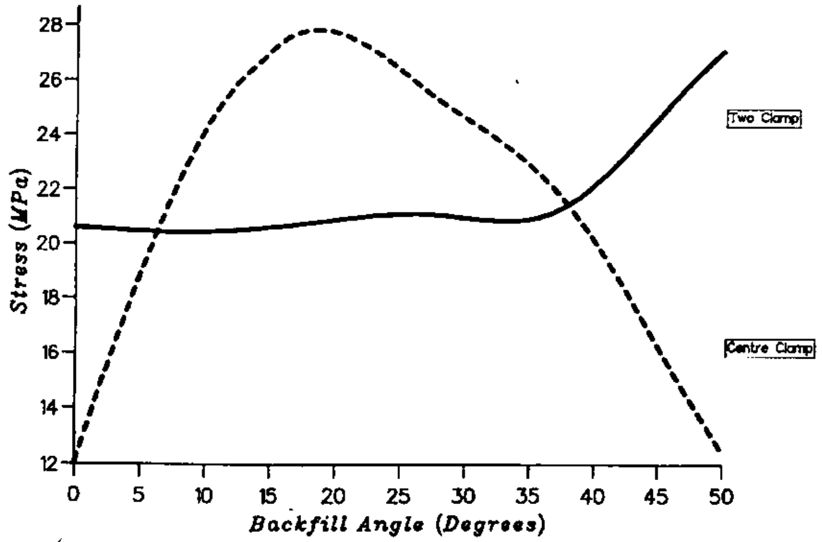


Figure 5.12. Effect of backfill angle on stress in fillet.

significantly reduced these stresses, i.e. the length of the flange overlap is not a significant design variable.

A parametric study has been completed [17] to assess the feasibility of the preformed top hat design and to determine the effect of different design parameters. Figures 5.11 to 5.15 show the variations of internal stresses relating to each variable. These results can be used to optimise the joint parameters for the particular load level anticipated.

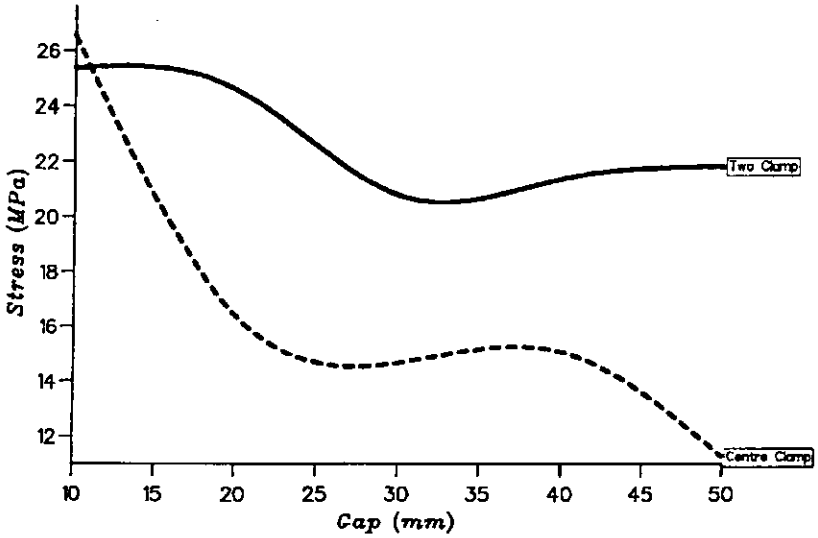


Figure 5.13. Effect of gap on stress in fillet.

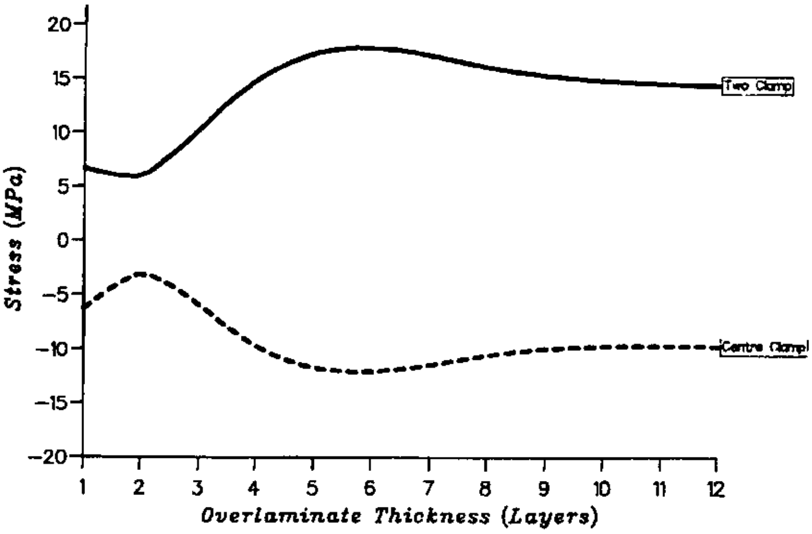


Figure 5.14. Effect of overlaminate thickness on through-thickness stress in overlaminate.

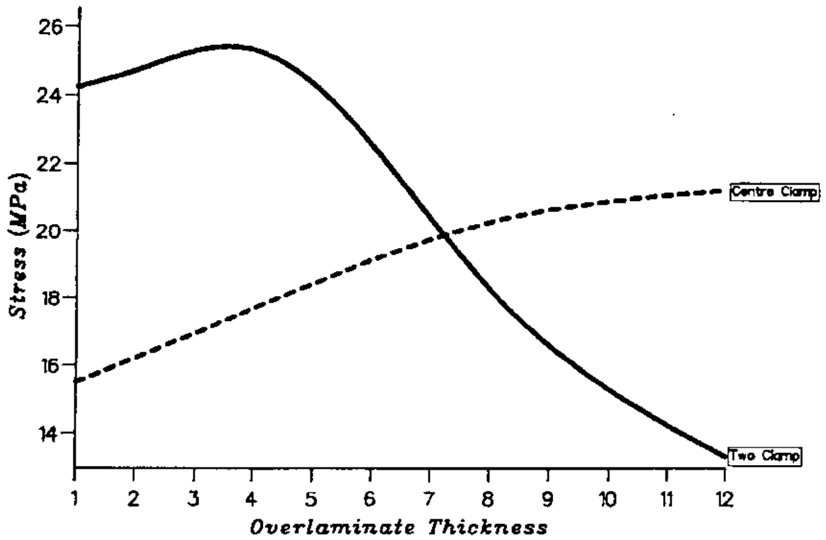


Figure 5.15. Effect of overlaminate thickness on stress in fillet.

5.4 OUT-OF-PLANE JOINTS II: BULKHEAD-TO-SHELL CONNECTIONS

5.4.1 Load Transmission Features

Typical bulkhead to shell joints are shown in figure 5.2. One feature common to all types, is that the overlaminate is wrapped around the corner linking the two structural elements. This is arranged so that the load is transmitted between the bulkhead and shell through this overlaminate in its in-plane direction. Traditionally this has been the only criterion considered - that is the sum of the in-plane properties of the two overlaminates is equivalent to the weakest member being joined. Since, in most applications, the material used in the overlaminate is the same as the material used in the structure, the thickness of each overlaminate is specified as a minimum of half the thickness of the thinner structural element.

However, it has long been recognised that the overall flexibility of a joint is as important as its strength. This is valid if the stress raising effect of the "hard point", created by the presence of an out-of-plane element such as a bulkhead, is to be avoided. Thus, the designer is faced with apparently contradictory requirements, that of strength (requiring a thick overlaminate) and flexibility (requiring a thin overlaminate).

5.4.2 Failure Modes

This contradiction is ably illustrated by considering the failure modes associated with these joints. The two elements carrying load are the overlaminate and the fillet. For both elements to be exploited efficiently they should both reach their respective failure stresses (both in-plane and through-thickness in the case of the overlaminate) at the same load. If the overlaminate is relatively thick (as in most cases) then it is also stiff. As load is applied to the joint, high through-thickness stresses develop in the corner of the joint and it fails by delamination. As the load is increased, further layers will delaminate until the overlaminate delaminates from the fillet. As soon as this occurs, the unsupported fillet fails. During this process the in-plane stress in the overlaminate is much lower than its failure value. Similarly, prior to failure, the stress in the fillet is also lower than the failure value of the material. This failure process has been observed by Hawkins et al. [22].

Clearly, to fully exploit the materials used, it is necessary to reduce the through-thickness stress, whilst allowing the in-plane stress and the fillet stress to increase. Traditionally, attempts have been made to reduce the delamination failure by increasing the overlaminate thickness. This is done to increase the stiffness of the joint, reduce deflections, and hence reduce the through-thickness stress. Also different laminating resins have been used with improved through-thickness properties. However, this approach reduces still further the in-plane stress and the fillet stress, so is less efficient in its use of materials, and creates a worse "hard point" in the structure. Any design method must be able to analyse the joint to quantify these stresses, and be able to determine the effect of design variables on them.

5.4.3 Design Variables

The design variables that relate to this form of joint are similar to those for top hat stiffeners above:

a) Radius of fillet
b) Fillet material
c) Thickness of overlaminate
d) Length of overlap of overlaminate
e) Lay-up of overlaminate
f) Overlaminate resin
g) Gap between bulkhead and shell

h) Edge preparation of bulkhead
i) Number, position, and make-up of additional plies
j) Number, position, and size of reinforcing bolts

Here, again, it is desirable to avoid using additional plies or reinforcing bolts if possible.

5.4.4 Design Modelling

Numerical analysis using, for example, the finite element method, is the easiest way to assess these relatively complex problems and account for all the design variables. A series of studies have been done [23] to investigate the impact of these variables, and which resulted in the new type of joint design shown in figure 5.2. This design exploits the low modulus and high strain to failure of the fillet material which allows large displacements and rotations to accommodated without failure.

The mechanism of this joint is as follows. As load is applied, the thin overlaminate bends so that the load can be transferred in its in-plane direction, as before. However this time, because the laminate is thin, and therefore flexible, the through-thickness stresses remain low. The out-of-plane deflections are accommodated by the flexible resin. The large radius gives the overlaminate a greater leverage on the joint so the in-plane stresses remain reasonable despite the overlaminate being thin. The actual values of overlaminate thickness and fillet radius can be optimised so that the ideal of ultimate failure occurring in the overlaminate (both in-plane and through-thickness) and in the fillet at the same load can be achieved. This effectively removes the contradiction that develops when using the traditional approach.

These two variables (i.e. overlaminate thickness and fillet radius) are by far the most important and their effects are summarised in table 5.2. Other variables have also been considered but were found to be less significant. Generally, variables should be adjusted to make the joint as flexible as possible whilst keeping stresses low. This can only properly be monitored by using the finite element method or by extensive, and expensive, experimental work.

The method just described can be equally well applied to the joining of sandwich panels but several other aspects must be considered. Principal amongst these is the need to keep the joint as flexible as possible to reduce through-thickness stresses between the skin and the core of the sandwich panels, which would otherwise

Table 5.2. Impact of geometry and material variations.

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

delaminate. The same numerical analysis methods can be used but with the different limitations applied a different optimum solution will develop. The core of the sandwich panels can be reinforced or removed in the region of the joint to overcome this problem - see figure 5.16 - and detailed finite element analysis can ascertain the stress patterns [24,25].

5.5 STIFFENER INTERSECTIONS

5.5.1 Purpose

In large 3-D structures that are subjected to complex, multidirectional loads it is sometimes necessary to stiffen the structure in two orthogonal directions. These stiffeners may be forced to intersect with one another. This should be avoided if at all possible. Stiffener intersections are fiddly to fabricate, expensive, heavy, and produce stress concentrations.

5.5.2 Design Features

The design features depend on the form of stiffeners being joined. In

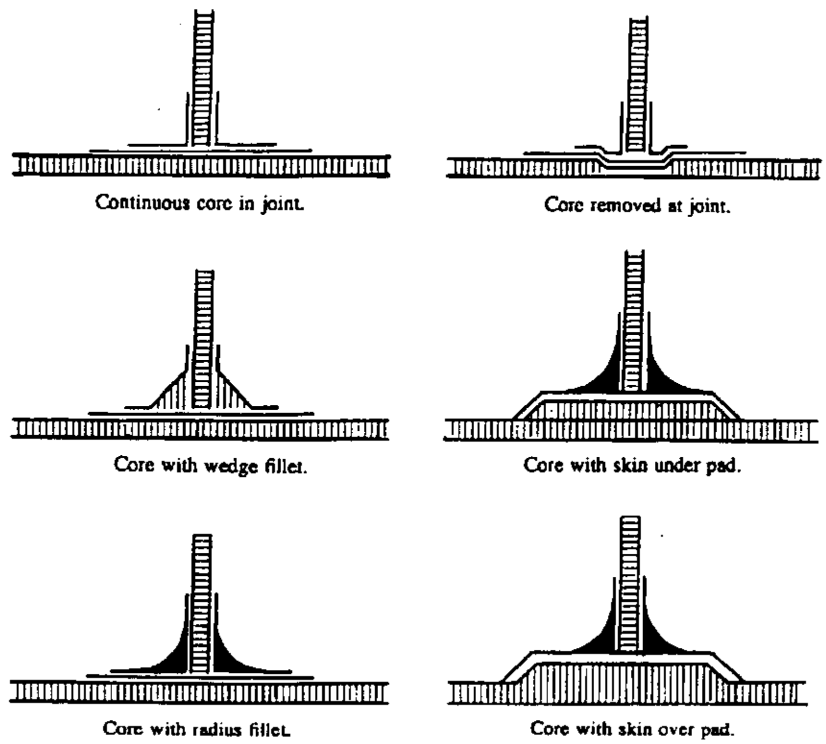


Figure 5.16. Bulkhead-shell joint geometries in sandwich construction.

marine applications the top hat stiffener is prevalent and is considered here. A typical arrangement is shown in figure 5.17. Typical design variables for such an intersection include:

a) Ratio of the intersecting stiffener heights
b) Thickness of external flanges
c) Length of external flanges
d) Radius of fillet
e) Fillet material
f) Number, position, and size of additional reinforcements

The two intersecting stiffeners should be of different sections, with the smaller one being continuous and the larger one being built over it. The greater the difference in height, the stronger the intersection will be. A ratio of at least 2:1 should be aimed for. This ensures that the tops of the stiffeners are not broken and so the global properties

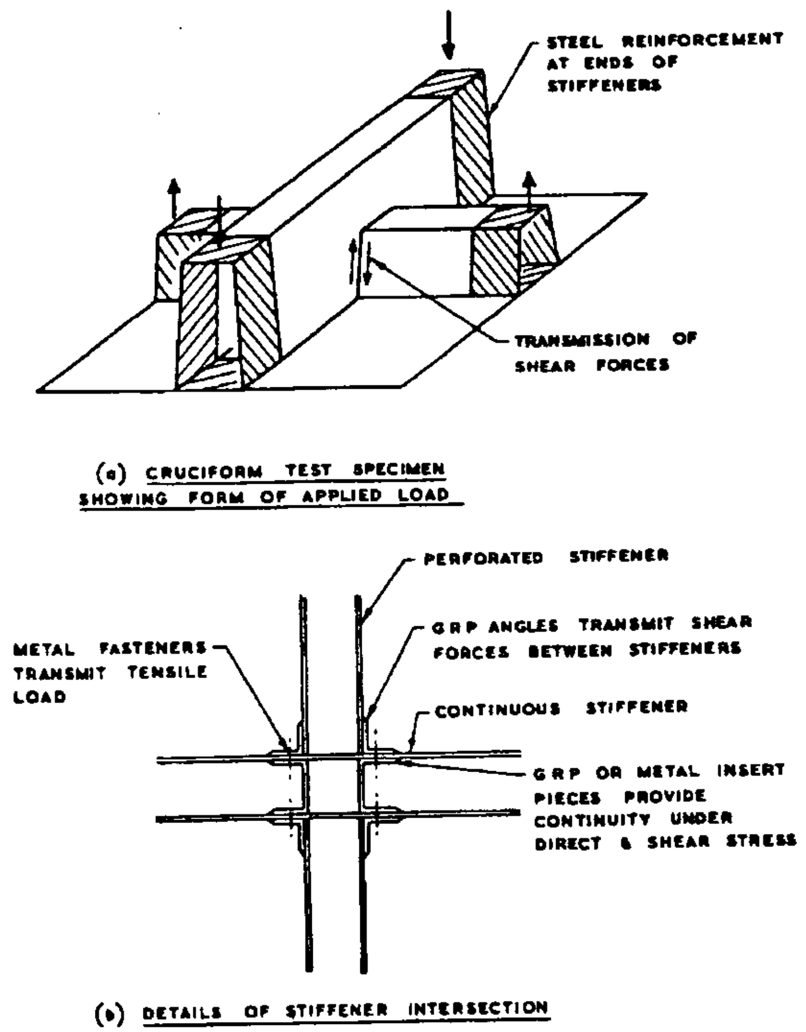


Figure 5.17. Typical arrangement of intersection between top hat stiffeners. (From [26]).

remain virtually unchanged. If this can not be arranged then insert pieces may be required as shown in figure 5.21. All that is then required is to attach the web of the large stiffener around the small one such that the direct, bending and shear loads are transmitted from one to the other. Since there is no access inside the larger stiffener this can only be done by external flanges.

5.5.3 Modelling

This can only be done using a numerical method or by experimentation. Some experimental work has been completed [26,27] on several different forms of intersections used on mine warfare vessels to confirm the designs used. The approach adopted was to use a boundary angle of at least the thickness of the stiffener web. There is a further requirement for parametric studies to be done in order to assess the impact of different variables.

5.5.4 Production Features

As mentioned earlier joints between top hat stiffeners of traditional construction are very awkward to fabricate. Any inserts required within the smaller stiffener must be accurately positioned prior to its fabrication. Any misalignment can affect the performance of the stiffener, particularly in compression; hence their use should be avoided where possible. Once the smaller stiffener has been formed, the surface where the large frame will bond to it is prepared. The former for this frame is then shaped to go over the other frame and is bonded in place. For simplicity the fillet radius and material should be the same as that used for the stiffener-to-shell joint. The larger frame is then formed, with the cloth having to be cut to go around the smaller frame. Further reinforcement is then added, with each piece of cloth having to be cut to accommodate the curvature. Clearly a lot of trouble is saved if the construction is, for example, sprayed-up chop strand mat, but nevertheless the work necessary to produce a quality joint of this form is considerable.

If the stiffeners are of the preformed variety described in section 5.3.1 above, then the construction of intersections between these stiffeners is potentially greatly simplified. Inserts can easily be added to the smaller stiffener, if required, before it is bonded in place and the larger preform simply needs to be cut to fit around the smaller frame with a suitable gap. The radius can be applied as before and the thin overlay requires much less work to form around the contours of the smaller stiffener. The suitability of this, as with any other, design needs to be confirmed, by either numerical analysis or experiments, for a given application.

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A PARAMETRIC STUDY TO DETERMINE THE INFLUENCE OF GEOMETRIC VARIATIONS ON THE PERFORMANCE OF A TOP-HAT STIFFENER TO SHELL PLATING JOINT

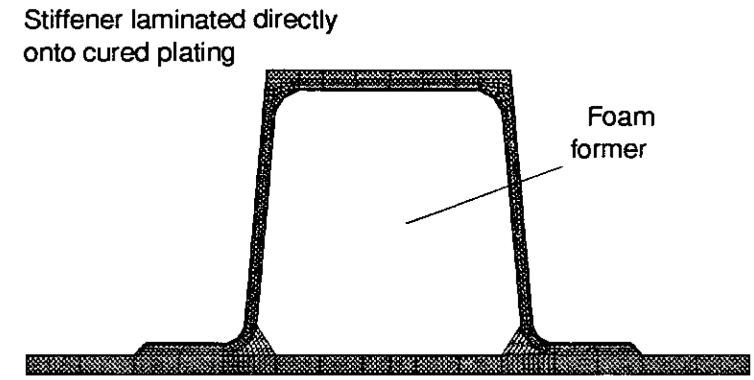
1. INTRODUCTION

A major constraint to overcome in the design and construction of large fibre reinforced plastic (FRP) structures is the relatively low modulus of the materials. Specific stiffness of FRP is a third to quarter of the value in metals. This feature leads to the need for stiffening large, unsupported spans of plating panels by appropriate mechanisms. The most prevalent form is the use of top–hat stiffeners.

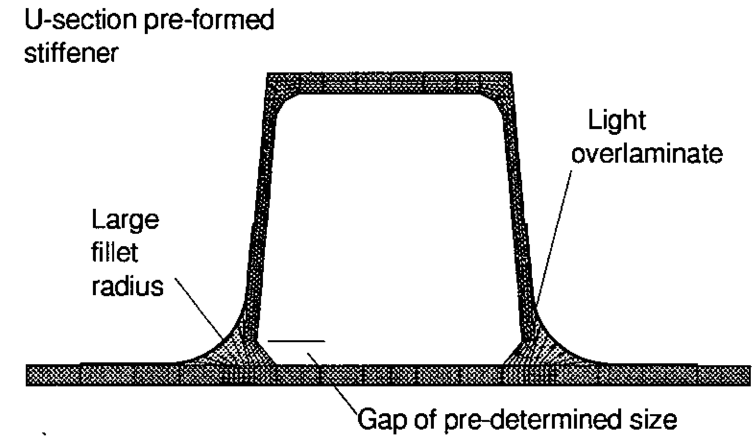
The present method to fabricate such stiffened structures starts initially with the laying–up of the unstiffened panel. Rigid foam cores are then laid on these panels in the appropriate locations, where stiffness is desired. An adequate amount of filleting resin is then squirted in the recess between the foam and the plating panel. Next, the required radius is scraped out by a curved spatula. The process of overlaminating now takes place; the resin–impregnated cloth is layed across the table, down the web and around the fillet on to the base plate panel. A similar cloth run is carried out for the opposite side. The process of overlaminating is repeated a number of times as necessary, frequently as many as 14 plies, to obtain the required stiffness. Occasionally, a limited number of unidirectional plies may be applied at the top of the table (or crown) to obtain extra stiffness. This is a time–consuming and expensive process. Figure 1A illustrates this type of stiffener.

An alternative form of building up the stiffness which requires study is owed to the increasing availability of pre–formed sections, as shown in Figure 1B. The proposed method would involve placing the pre–formed section at the right location and being bonded to the base panel by the squirting of filleting resin (as in the case of the foam former). This would then be capped by two or three layers of woven roving overlaminate.

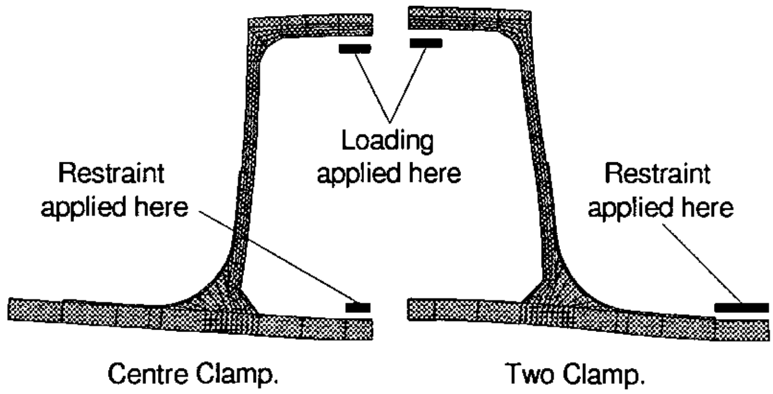
In either case, a key characteristic of such joints is that, because of a lack of continuity of reinforcing fibres across the joint, they are susceptible to fail by peel or delamination well before the ultimate in–plane material stress is



A: Current Design of Stiffener.



B: New Design of Stiffener.



C: Boundary Conditions Applied to Models.

Figure 1

reached. Furthermore, their dependence on interlaminar properties make them somewhat sensitive to material imperfections such as voids and to minute changes in geometry in the laminate. Because of such sensitivity in structural performance and the weight/cost implcations involved, it is important to ensure that the design and production of such joints is carefully carried out.

The purpose of this paper is to study the problem of top–hat joints and to identify key variables that control and govern the transfer of load from the panel to the stiffener and vice–versa.

2. CURRENT PRACTICE

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 with simple design examples. Section moduli and moments of inertia are tabulated for different geometries of top–hat stiffeners. The dimensions of the boundary angles and overlaminates, it is stated, "should be minimum compatible with strength requirements". However, no specific procedures concerning joint design are elaborated. Early work in the UK centred around the design of GRP minehunters (2) and this formed the impetus to drawing–up of naval engineering standards (3). Early designs specified the use of bolts through both the base plate and the overlaminate to double–up on the bonded connection, thus ensuring that total peeling of the top–hat from the base plate did not occur. This practice was superceded by the use of bonded connections alone, with the bonding being achieved by the use of flexible, urethane acylate resin for the fillets (4). The rules for naval ships are based primarily on extensive experimental work. The standards prescribe minimum limits to various scantlings. Overlaminate thickness, for example, is specified to be at least half (and preferable two–thirds) the thickness of the thinnest member at the joint. Flange overlap dimensions, the lay–up and stacking sequence are also specified.

With regard to non–naval craft, design guidance is sought primarily from classification society rules. Lloyd's rules (5) state that, for top–hat stiffeners, the "width of the flange connection to the plate laminate are to be 25mm + 12mm per 600 g. per sq.m. of reinforcement in the stiffener webs, or 50mm whichever is the greater". ABS rules (6) state that "the minimum overlap on the plating should be 20% of the web depth or 50mm, whichever is greater. There is also a minimum thickness requirement for the overlaminate. DnV rules (7) specify that first–principles based calculation procedures should be adopted to determine section moduli of top–hat stiffeners and scantlings of plate laminates. However, no explicit guidance is given with regard to the requirements, standards or design allowables. This lack of specificity is further evident from a recent survey of current practice (8) where the design procedures for top–hat are merely updates on the Gibbs and Cox practice.

The main thrust of the current practice seems to be one of aiming to make the top–hat stiffener to base plate connection as stiff as possible. Furthermore, there is a mistaken belief that increased stiffness also corresponds to increased strength of the joint. This fallacy has been exposed for other similar bonded connections, i.e. tee joints (9). The procedure adopted for the work outlined in this paper follows the tee joint approach which focused on an explicit identification of design variables and production processes influencing joint behaviour.

With regard to the choice of production method, bearing in mind advances in technology, it was felt that the potential use of pre–formed sections merits study. Figure 1B illustrates the key design variables thought to influence the behaviour of the joint. Because of the lack of analytical approaches, giving exact solutions, recourse was made to numerical techniques for modelling the joint behaviour.

3. MODELLING

All the modelling was done using the ANSYS suite of programs. This offers three elements with composite capabilities, two shell, and one 3D solid. The geometry of the top–hat section means that some stacking of elements is required so the 3D solid element is used throughout. This element is defined by eight nodes, each having three degrees of freedom (translations in the nodal x, y, and z directions), and has large deflection and stress stiffening capabilities.

Several preliminary analyses of a similar problem were conducted to assess the effects of incorporating these capabilities, and also the effect of varying the mesh density. Memory and wavefront considerations apply an upper limit to this but nevertheless a reasonably high level of definition was achieved. Models were developed with a consistant mesh density so that any mesh density effects are eliminated from the parametric results. These were run using the large deflection option.

The material properties used were derived from manufacturers data (where this was available), from experimental results, and by interpolation from properties of similar materials. This last method was necessary since for some materials there was no data available, and the nature of the material made the manufacture of test coupons very difficult. Nevertheless the derived properties have been shown to provide good correlation with experimental results (9). The properties used are listed in Table 1. The properties of the fillet material are nonlinear and incorporate very large plastic deformation before failure, making it necessary to apply the load incrementally to follow this plasticity.

Figure 1C shows the boundary conditions applied to the models, which represents centre–clamp and two–clamp loading. These two loading methods apply opposite bending moments to the region of the joint so that both the fillet, and the overlaminate, are subjected to tensile loading. In previous experimental and theoretical work (4, 10) different loading methods were applied and it was found that centre clamp loading was the most severe in terms of having the lowest failure load. This loading method was then adopted as a standard for U.K naval practice. However, this loading method does not precipitate the premature delaminations in the root of the joint seen during explosion trials so two clamp loading, which does result in these delaminations,

Material	E (GPa)	υ	Gxy (GPa)	UTS (MPa)	Failure Strain (%)
Polyester	3.2	-	-	58	6
* CR1152	0.5	-	-	26	100
+ CR1200	0.7	-	-	32	27
Polyester/WR					
Warp	14.68	0.123	3.09	207	1.4
Weft	13.06	0.139	3.09	207	1.4
Interlaminar	-	-	-	12.2	
Polyester/CSM					
Inplane	6.89	0.13	3.45	-	
Interlaminar	-	-	-	11.2	
CR1200/WR					
Warp	6.375	-	-	183.3	2.8
Weft	3.926	-	-	188.9	4.8
CR1200/CSM					
Inplane	3.023	-	-	110.4	3.6

* CR1152 = Urethane Acrylate Resin.
\+ CR1200 = Polyester/Urethane Acrylate mix.

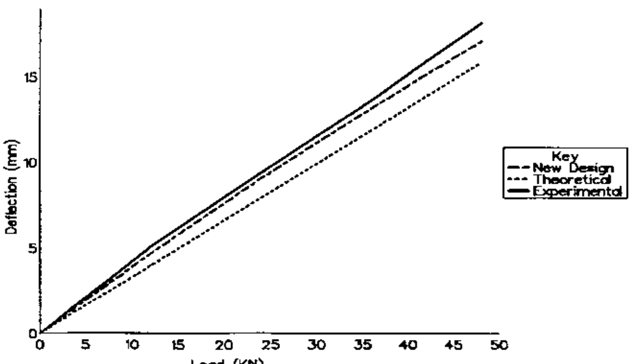
Table 1: Material Properties used in F.E. Analysis.

is also considered here.

4. DISCUSSION OF RESULTS

Initial results are taken from a study to assess the relative performance of the new design of top hat when compared to the current method of construction. The load/deflection characteristics of the model of the current design were compared to existing experimental data. Unfortunately there is no experimental data available for the new joint design, but since a consistent model definition has been maintained from the current to the new design, it has been assumed to be similarly representative. Figure 2 shows load/deflection curves for the current design, both experimental and theoretical, and for a nominal new design with a fillet radius of 75mm, a gap of 20mm, a backfill angle of 45 degrees, and an overlaminate thickness of 2 layers of woven rovings, all for centre clamp loading. This shows close correlation between experimental and theoretical results, and also shows that the overall stiffness of the top hat section is little affected by the design of the joint.

This last result is explained by Figures 3A and 3B. These show the deflection distribution through the model of the new design, for centre clamp and two clamp loading respectively. For centre clamp loading the displacement of the base panel is very similar in magnitude to the overall displacement at the flange of the top hat, and for two clamp loading it is a significant proportion of the overall displacement. This indicates that the deflection of the base panel has the most influence on the overall deflection. Since the construction of the base panel is similar for all models, their deflections are also similar. This important result shows that the function of the stiffener, that is to stiffen the base panel, is a function of the stiffener scantlings, and not the design of the joint between the stiffener and the base panel. This means the design of this joint can be refined to improve its performance without impairing the overall performance of the stiffener itself.



Comparison of Load/Deflection Curves (Centre Clamp)

Figure 2.

Once the functionality of the new design had been ascertained a parametric analysis was conducted to study the effects of changing the four main geometric variables shown in Figure 1(b). The results of this analysis are shown in Figures 4, 5, and 6. These show the overall displacement, maximum fillet stress,

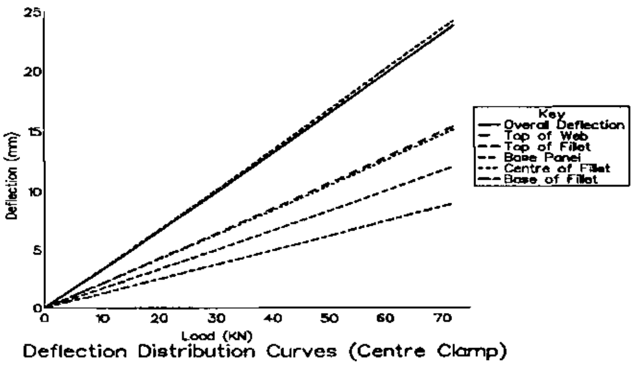


Figure 3A

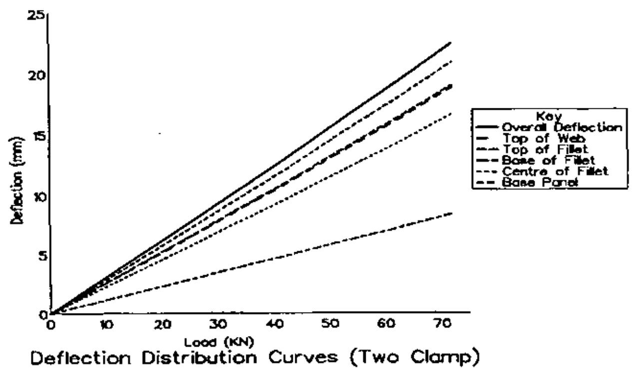


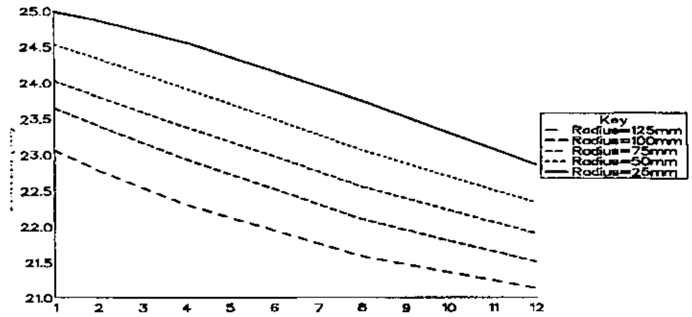
Figure3B

maximum overlaminate in–plane stress, and maximum overlaminate through–thickness stress, at a fixed load, for overlaminate thickness, gap, and backfill angle respectively. Both centre– and two–clamp loading are considered, and each graph includes a series of curves for different fillet radii. Previous work on a similar problem (9) had shown fillet radius to be the most significant geometric variable, and since it is the best representative of the scale of the joint, it is used as the "base" for the results presented here.

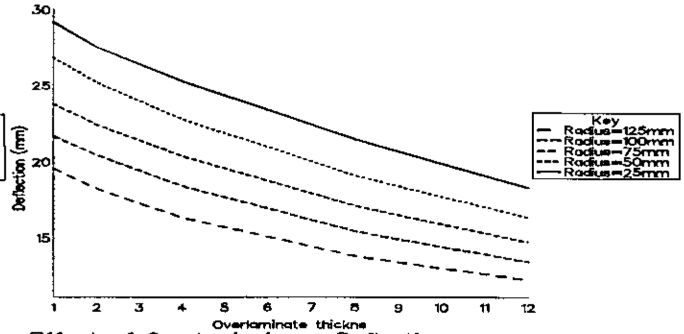
Firstly considering Figure 4, the results for the variation in overlaminate thickness. Backfill angle and gap are constant at 45 degrees and 20mm, respectively, and fillet radius varies from 25mm to 125mm. Several features can be seen:

1. Overall deflections decrease with increasing overlaminate thickness and fillet radius. The range of variation is small for centre clamp and larger for two clamp, as expected from Figure 3 above, but this variation occurs in a relatively small region (the joint) of the top hat section and thus can have a significant effect on the stress distribution within this region.

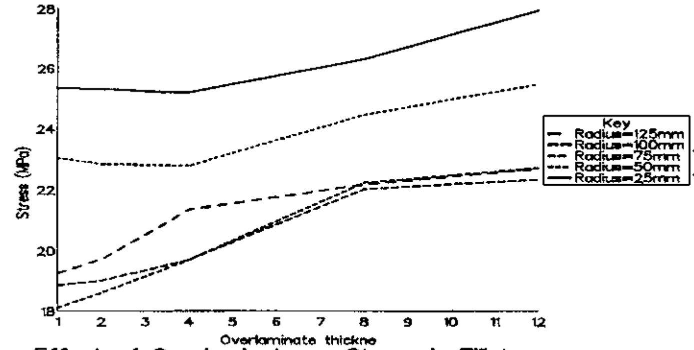
2. The maximum principal stress in the fillet increases with increasing overlaminate thickness for centre clamp loading, but the range of variation is small. This is associated with an increase in the overlaminate in–plane tensile stress (indicating a reduction in compressive stress) and an increase in the overlaminate through–thickness stress. Previous work on a similar problem (11) indicated that failure in joints of this type is initially precipitated by delamination in the overlaminate (caused by high through–thickness stresses) followed by fillet failure. Failure never occured in the plane of the overlaminate. In light of this it would seem that increasing overlaminate thickness, whilst being beneficial in reducing in–plane stresses, will reduce overall joint performance by increasing through thickness and fillet stresses.



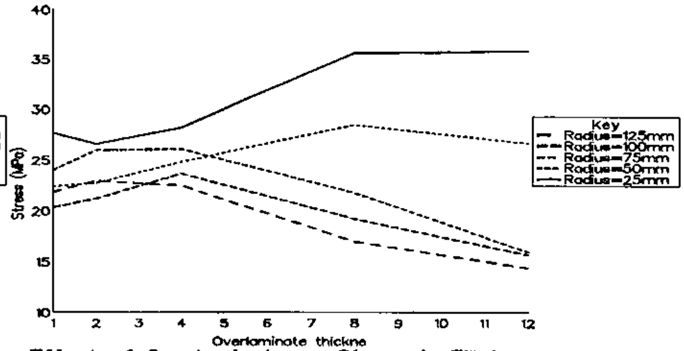
Effect of Overlaminate on Deflection



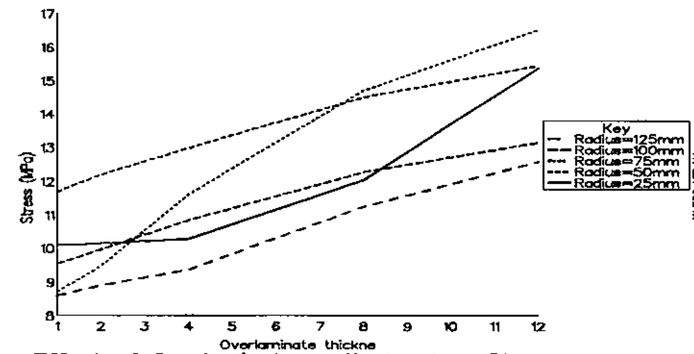
Effect of Overlaminate on Deflection



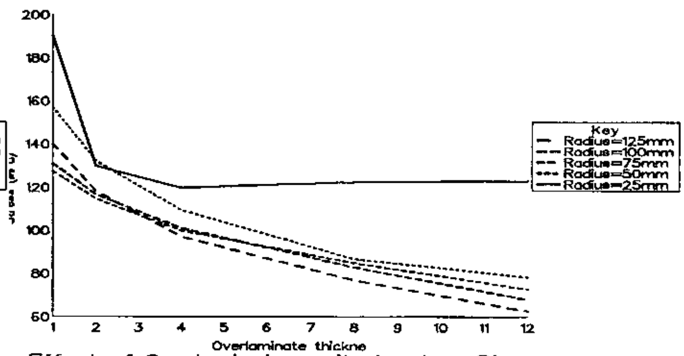
Effect of Overlaminate on Stress in Fillet



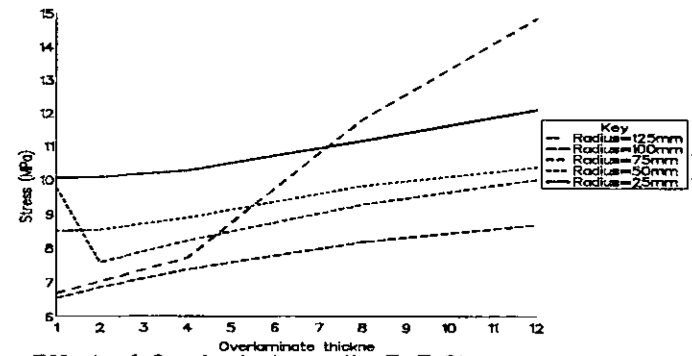
Effect of Overlaminate on Stress in Fillet



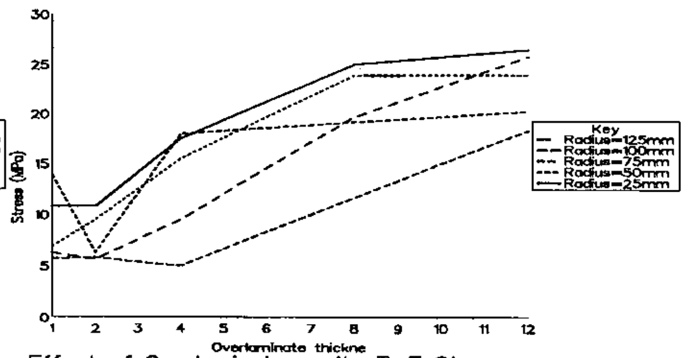
Effect of Overlaminate on its In-plane Stress



Effect of Overlaminate on its In-plane Stress



Effect of Overlaminate on its T-T Stress



Effect of Overlaminate on its T-T Stress

Centre Clamp **Two Clamp**

Figure 4: Results for Different Overlaminate Thicknesses.

3. A similar trend is seen for two clamp loading when the radius is small. However, for larger radii the fillet stress reduces as the overlaminate thickness increases. The range of variation is much larger. The in–plane stress in the overlaminate also reduces and the through thickness stress increase, as with centre clamp loading, and the same conclusion can be drawn.

4. For both centre and two clamp loading the fillet stress decreases with increasing radius, whilst the radius is small. Once the radius exceeds approximately 75–100mm, however, the fillet stress level remains reasonably constant. This pattern is even more pronounced when considering the overlaminate through thickness stress. Once the radius exceeds 100mm stress levels increase, especially at higher overlaminate thicknesses. This indicates that excessively large radii can reduce the joint performance.

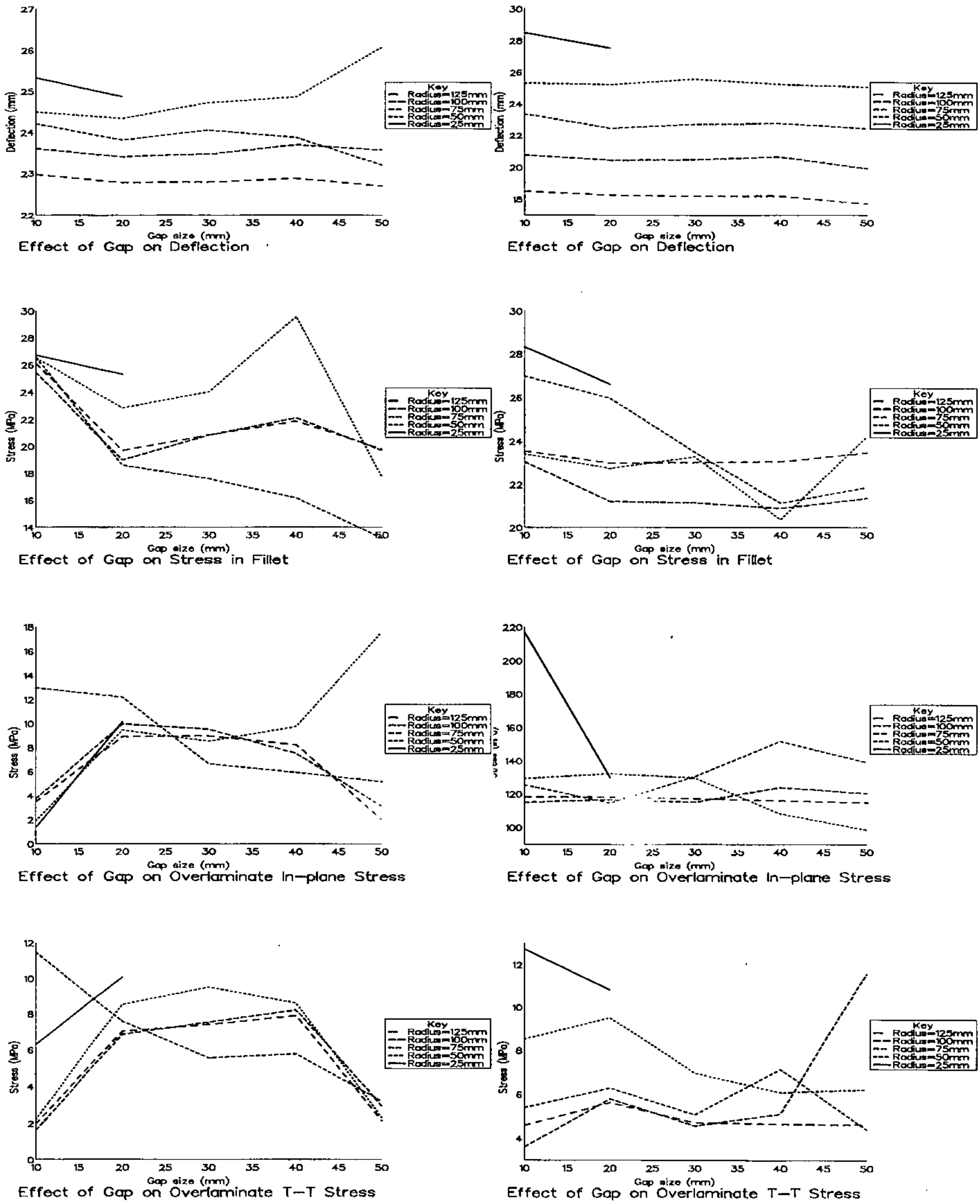
Next considering Figure 5, the results for the variation in the gap between the stiffener web and the base plate. Overlaminate thickness and backfill angle are fixed at 2 layers of woven roving, and 45 degrees, respectively. In a production context the gap size can be controlled within certain limits, but these will be variable depending on the complexity of the curvature of the base plate. The range of gap sizes considered in this study (10–50mm) was considered to sufficiently cover production variations. Significant results are:

1. Gap size has little effect on overall deflection for both centre and two clamp loading, indicating that overall stiffenner performance will not be affected by production variations.

2. For two clamp loading the fillet stress reduces with increasing gap size until a minimum is reached, and as gap size increases further the fillet stress begins to rise. The optimum gap size is dependent on the fillet radius and generally reduces with increasing radius. For centre clamp loading the same general trend is seen, except that as gap size increases to the higher values the fillet stress reduces again.

3. The overlaminate in–plane stress is little affected by gap size except at very small radii, where increasing gap size reduces the in–plane stress.

4. For centre clamp loading the overlaminate through–thickness stress generally rises with increasing gap size to a maximum value, and then decreases again. For two clamp loading this effect is less marked with little variation.



Centre Clamp **Two Clamp**

Figure 5: Results for Different Gap Sizes.

Finally, considering Figure 6, the results for the variation in fillet backfill angle. The overlaminate thickness and gap size are constant at 2 layers of woven rovings and 20mm, respectively. The backfill angle is considered as a variable here because in production it is a parameter that is very difficult to control, depending very much on the skill and consistency of the injector operator. Careful use with optimised nozzles should produce consistent 45 degree backfill angles with some attachment to the back of the web, but it was considered prudent to determine the effect of less than ideal production. Several salient points are shown:

1. Increasing backfill angle decreases the overall deflections slightly for the smaller radii with centre clamp loading, but has no effect for any other geometry or loading condition considered. As with gap size this shows that production variance should not affect overall stiffenner performance.

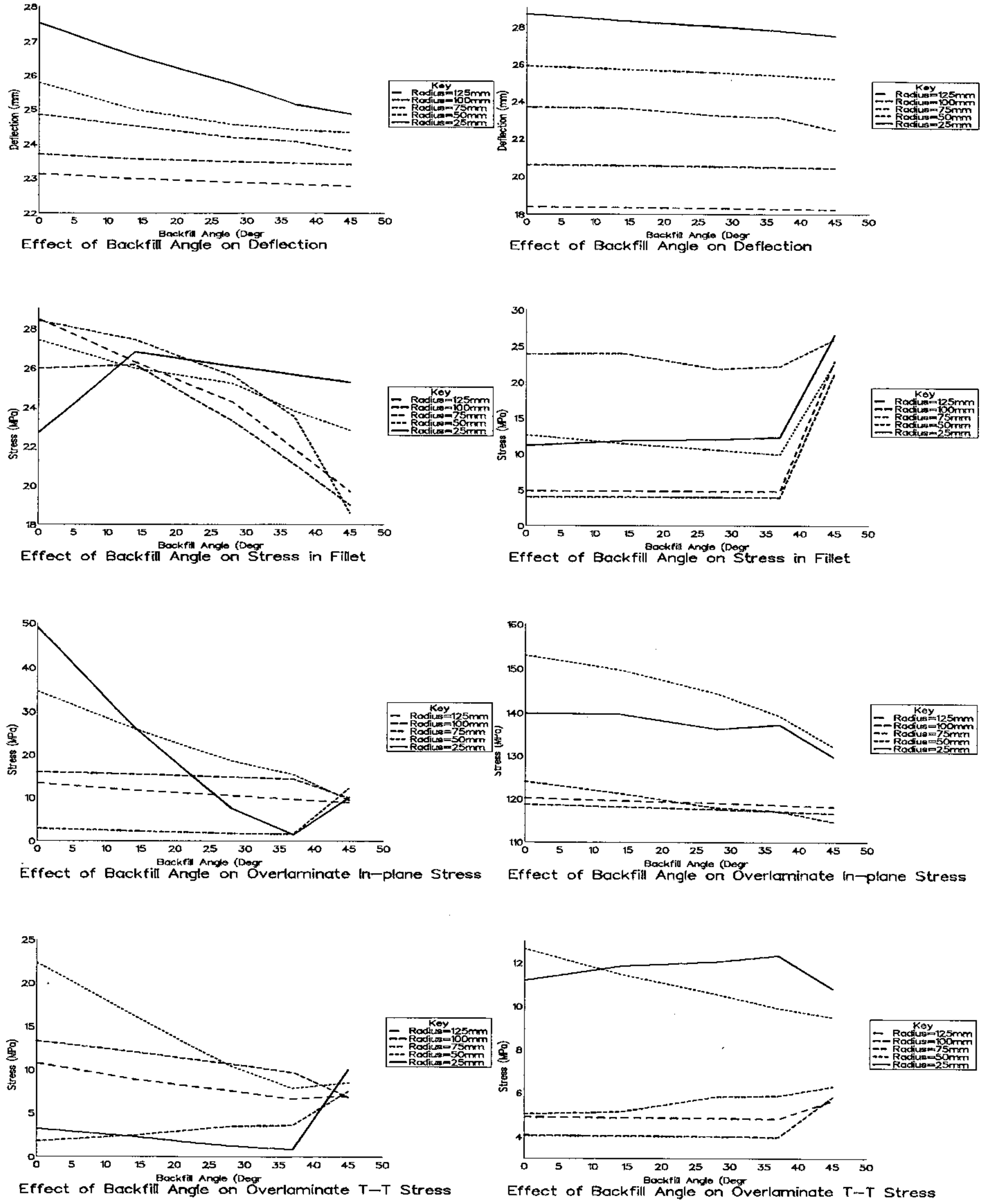
2. For centre clamp loading the stress in the fillet decreases with increasing backfill angle. For two clamp loading the fillet stress is unaffected by changes in the backfill angle except at high values when the fillet stress increases.

3. In–plane and through–thickness stresses are little affected by backfill angle except when the fillet radius is small. In these cases the stresses decrease with increasing backfill angle.

5. CONCLUSIONS

Generally the results of this work have shown that the new design of top hat stiffener is a feasable alternative to the current practice. Moreover, the results indicate that a suitably refined joint should be able to atleast match the overall performance of the current design without being prone to premature delaminations in the overlaminate. This, along with the potentially considerable savings in production costs, should ensure its further consideration.

With the exceptions noted in the discussion of results above, the "production" variables (gap size and backfill angle) have a limited effect on the performance of the joint. Fillet radius and overlaminate thickness have significant effects on joint behaviour and can be optimised for a given application. Increasing fillet radius and reducing overlaminate thickness can overcome the problems of delamination in the overlaminate by reducing through–thickness stresses. This is in direct contrast to current practice.



Centre Clamp **Two Clamp**

Figure 6: Results for Different Backfill Angles.

The graphs included in this paper can be used for a design optimisation procedure providing that failure information exists for the materials used. These are not available at present.

6. ACKNOWLEDGEMENTS

The work outlined in this paper was funded by the SERC (administered through the MTD Ltd.) and Vosper Thornycroft Ltd. Southampton.

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A Parametric Study to Determine the Influence of Geometeric Variations on the Performance of a Bulkhead to Shell Plating Joint.

1. Introduction

FRP structures are especially prone to failure at bonded connections. The weakness in this context is caused by the absence of load bearing fibres across bonded surfaces, by the relatively thin adhesive layer forming the bond, and through the occurence of stress concentrations associated with joint geometry and production considerations. 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.

The connections could be entirely bonded or may be reinforced by the use of metal fasteners such as bolts, screws or rivets. Use of fasteners in ships is not very cost efficient and hence this practice is relatively uncommon. Because of this lack of redundancy in attachments, special care needs to be excercised when dealing with major structural connections.

In ships, the connections could refer to:

butt joints which may be required between prefabricated deck panels;

intersections between longitudinal and transverse stiffeners;

linking deck edge with shell; or

joining bulkheads to deck or shell.

In a design context, much progress has been made with regard to the overall hull structure. However, apart from some testing and analysis in the early days of GRP ship building, relatively little attention has been paid to the design of joint details. Whilst the resulting joints have fully met stringent design requirements which, in warships, may require underwater shock loadings, there is much scope for better understanding of joint behaviour, leading to more cost and performance–efficient designs.

In a design context, some testing and analyses of these joints were carried out in the early days of GRP shipbuilding, and this formed the basis of current design practices. More recently work has centred on determining the key design characteristics that affect joint performance, using both experimental, and detailed analytical studies. This work resulted in a "new" design of joint being developed whose characteristics directly opposed the current design principles.

The object of this paper is to study the behaviour of this new design of Tee connection by parametrically varying these key geometric features to determine their influence on the joints performance.

2. Current Practice.

The primary function of a Tee joint is to transmit flexural, tensile and shear loads between two sets of panels meeting at the joint. The joint itself is formed by laminating strips of reinforcing cloth (or overlaminates) either side of the joint to form a double (or boundary) angle connection, as illustrated in Figure 1. The load transmission is done entirely through the cloth–resin plies and any filleting resin within the boundary angle. The details of the cloth plies and resin therefore need careful attention in order to ensure the joint is strong enough to resist the applied loads.

Such careful and explicit modelling of joint geometry and material makeup is not allowed for by currently available design synthesis methods. One of the earliest approaches to GRP structure design is outlined in the Gibbs and Cox manual (1) where it is stated that the dimensions of the boundary angle "should be consistent with strength requirements"; specific procedures concerning joint design, however, are not elaborated. Current naval standards (2) prescribe the thickness of the boundary angle to be at least half (and preferably two–thirds) the thickness of the thinnest member meeting at the joint. Flange layup dimensions, the layup and stacking sequence are also specified. A typical arrangement is shown in Figure 1A. Again no design requirements or modelling procedures are laid down.

With regard to current merchant marine and small craft practice, design guidance is sought primarily from classification society rules. Lloyds rules (3) state that "the scantlings are to be determined by direct calculation". Guidance is provided with regard to layup and stacking sequence. In the rules of the American Bureau of Shipping (4), boundary angle thickness is again given as a function of members meeting at the joint. Det Norske Veritas (5) has no formulae for direct derivation of scantlings but offers tables of loadings, allowable material properties and maximum/limiting stresses from which the

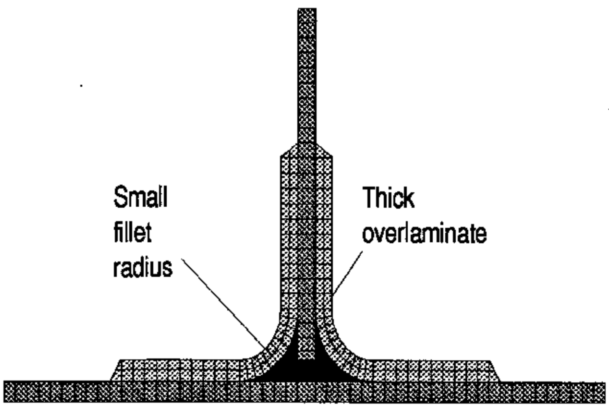
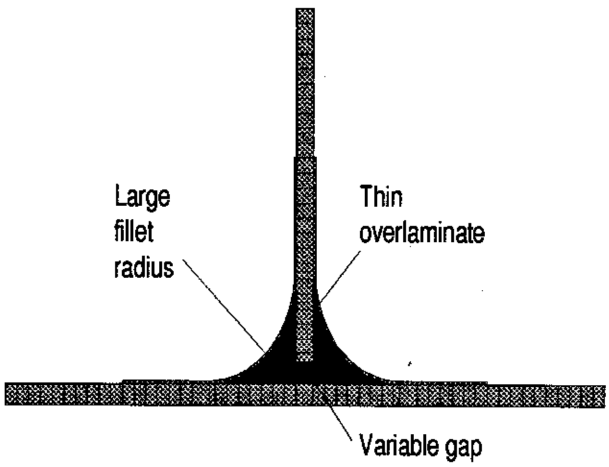


Figure 1: Geometries and Features of Tee-joints.

A: Typical Tee-joint Designed to Current Practice



B: "New" Design of Tee-joint

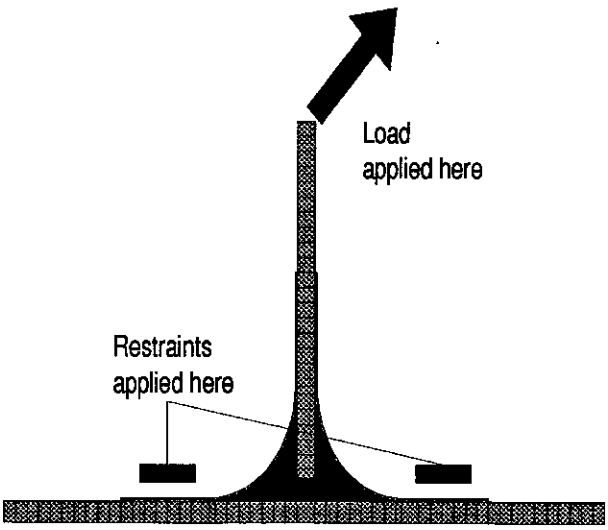


Figure 2.

Boundary Conditions Applied to Tee-joint Models boundary angle details are to be determined. As before, no modelling preference is specified.

Arguably the most comprehensive efforts towards understanding joint behaviour have been in the field of naval minehunter design. Early efforts (6–7) were targetted towards mechanically fastened joints. However, primarily because of cost considerations, later developmental research (8–9) concentrated upon replacing bolted connections with bonded ones. The thrust of these studies was also of a practical and developmental nature. The studies were targetted towards specific applications and hence broader conclusions towards general usage are difficult to extract.

On a theoretical front, limited attempts at studying joint behaviour have been on the basis of numerical finite element studies (10–11). The scope of these studies was restricted to some extent by the then available software. "Traditional", analytical methods are currently incapable of modelling joint behaviour due primarily to complexities in geometry and material make up. Furthermore, there are practical considerations standing in the way of exact analysis. These include the discontinuous nature of the layup within boundary angles, presence of voids and imperfections due to production processes and the extensive step variations in material properties.

Currently, therefore, there is an absence of a rigorous design procedure that allows optimum layups within Tee joints. The first step towards developing this procedure (12) has been to obtain an understanding of boundary angle or Tee joint behaviour by explicitly identify the variables which define Tee joint design – see Figure 1B. Then these parameters have been systematically varied using numerical modelling.

3. Modelling Considerations.

The numerical analysis was performed using the ANSYS finite element analysis package (13). This has a composite capability in that it provides options from three layered elements, comprising two eight–noded shell and one eight–noded solid. The geometry of the boundary angle problem precludes the use of shell elements as some stacking is required, so the 3D solid element is used throughout. This element is defined by eight nodes, each having three degrees of freedom (translations in the nodal x, y, and z directions), and has large deflection and stress stiffening capabilities. The flexible resin fillet was represented using an isoparametric 3D solid element with large deflection and plasticitycapabilities.

Several preliminary analyses were conducted to assess the effects of incorporating these capabilities, and also the effect of varying the mesh density. Memory and wavefront considerations apply an upper limit to this but nevertheless a reasonably high level of definition was achieved. Models were developed with a consistant mesh density so that any mesh density effects are eliminated from the parametric results. These were run using the large deflection option.

The material properties used were derived from manufacturers data (where this was available), from experimental results, and by interpolation from properties of similar materials. This last method was necessary since for some materials there was no data available, and the nature of the material made the manufacture of test coupons very difficult. Nevertheless the derived properties have been shown to provide good correlation with experimental results (12). The properties used are listed in Table 1. The properties of the fillet material are nonlinear and incorporate very large plastic deformation before failure, making it necessary to apply the load incrementally to follow this plasticity. Iterative solutions were completed using 10 iterations per load step and 15 load steps. A convergence criterion was applied to each iteration to prevent excessive processing. Convergence was usually found to occur after between 4 and 8 iterations depending on the model and the loadstep.

The models were restrained as shown in Figure 2, to represent the clamping arrangement used during experimentation. This loading condition was developed to represent a boundary angle subjected to both a tensile pulloff and sideloading, as would occur within a tank structure. This has been used as a "standard" test for many years and has been shown to give good comparative results (8).

4. Discussion of Results

The basis models from which all the following results have been derived have been shown previously (12) to give reasonable correlation with experimental results. Experimental results are not available for most of the wide range of geometries considered here, but it has been assumed that the consistency maintained across the models will ensure the validity of the comparative results.

Results are given for the three most significant variables, that is fillet radius, overlaminate thickness, and gap size. Fillet radius has been used as the "base"

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Polyester/WR					
Warp	14.68	0.123	3.09	207	1.4
Weft	13.06	0.139	3.09	207	1.4
Interlaminar	-	-	-	12.2	
Polyester/CSM					
Inplane	6.89	0.13	3.45	-	
Interlaminar	-	-	-	11.2	
CR1200/WR					
Warp	6.375	-	-	183.3	2.8
Weft	3.926	-	-	188.9	4.8
CR1200/CSM					
Inplane	3.023	-	-	110.4	3.6

* CR1152 = Urethane Acrylate Resin.
+ CR1200 = Polyester/Urethane Acrylate mix.

Table 1: Material Properties used in F.E. Analysis.

variable since it has the greatest effect on joint performance, and is the best representation of scale, allowing the results to be interperated for other applications. Results are given in terms of overall deflection, maximum stress in the fillet, maximum overlaminate in-plane stress, and maximum overlaminate through-thickness stress, for a constant load (in this case 12.5 KN). The aim of efficient joint design is to ensure that all the above stresses reach their maximum values at the same applied load. Traditionally this has not been possible as overlaminate through-thickness stresses have been excessive in thick overlaminates long before in-plane and fillet stresses have developed, resulting in premature delaminations occurring in the root of the boundary angle.

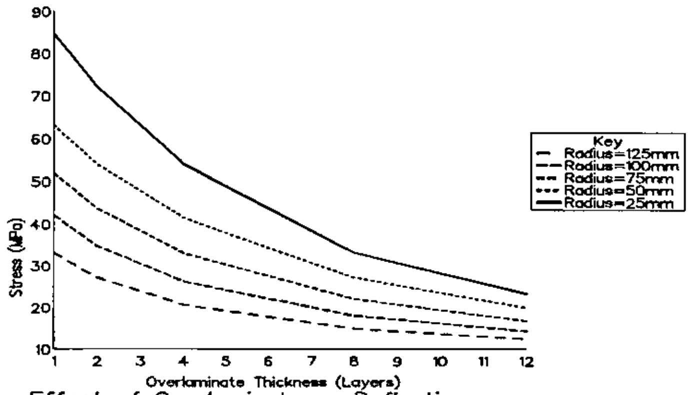
Firstly, consider Figure 3, the results for the variation in overlaminate thickness and fillet radius. The gap size is constant at 20mm. Several points can be seen:

1. Overall deflections decrease considerably with increasing overlaminate thickness and fillet radius. Since in the Tee-joint almost all of the overall deflection occurs in the joint region this would be expected to have a significant effect on the internal stress distributions within the joint

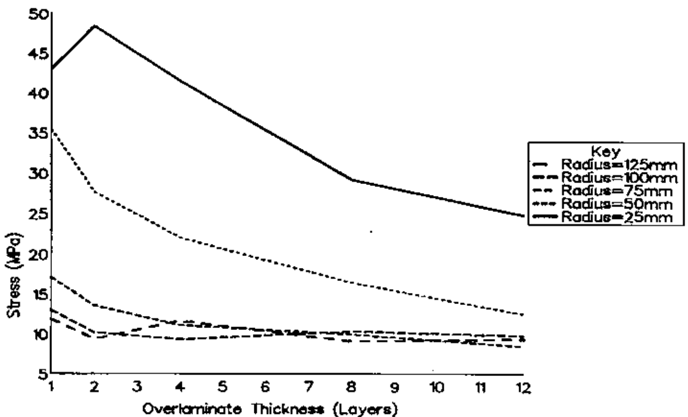
2. When the radius is small the stress in the fillet reduces as overlaminate thickness increases. As fillet radius increases, however, the fillet stress becomes almost independent of overlaminate thickness and fillet radius, indicating that overlaminations of more than four layers, and fillet radii greater than 75mm, have little effect on fillet stress. However, fillet radii smaller than this should be avoided as the fillet stress rises rapidly as the radius is reduced.

3. The in-plane stress in the overlaminate reduces rapidly as overlaminate thickness and fillet radius are increased from small values. However, as both these variables increase further their effect is reduced, indicating a diminishing return for the increased joint size.

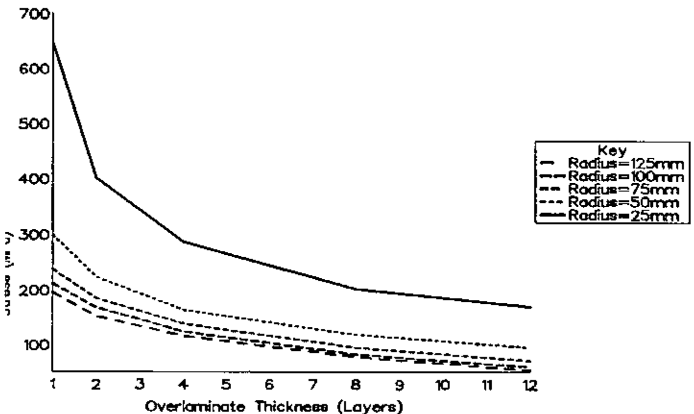
4. For all but the smallest fillet radius, the overlaminate through-thickness stress increases with increasing overlaminate thickness. This increase, whilst appearing small in the figure due to the inordinately high values for the 25mm fillet radius, is very significant when failure, for a typical polyester resin/woven roving laminate, will occur at 8–11MPa (14). The very high stress levels in the 25mm fillet radius models, both in the fillet and the overlaminate, indicate that the joint would have failed long before this level of applied load was reached.



Effect of Overlaminate on Deflection.

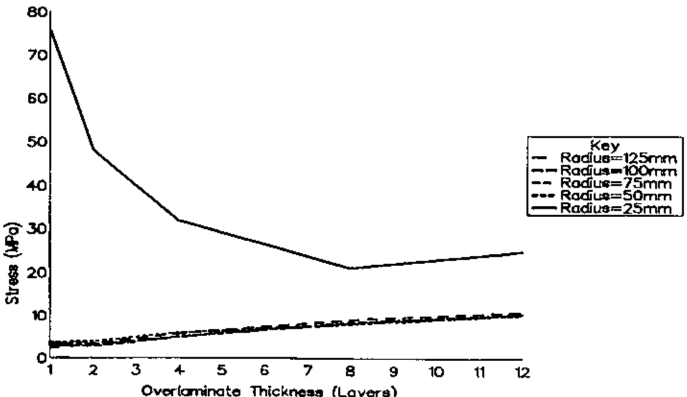


Effect of Overlaminate on Stress in Fillet.



Effect of Overlaminate on its In-plane Stress

Figure 3: Results for Variations in Overlaminate Thickness



Effect of Overlaminate on its T-T Stress

Next consider Figure 4, the results for varying gap size and fillet radius. Overlaminate thickness is constant at 2 layers of woven rovings. Gap size is a significant variable in as much as in the production of large structures it is difficult to control, thus its effect on the joint performance need to be determined. The results are given in the same format as above, for the same applied load. Significant points are:

1. Gap size has no effect on overall deflections or stresses in the overlaminate, and the effect of radius remains as before.

2. The stress in the fillet rises as gap size increases, but this effect is reduced as fillet radius increases, such that for radii greater than 75mm gap size has little effect up to 20mm. Large gap sizes result in high fillet stresses in all but the largest fillet radii.

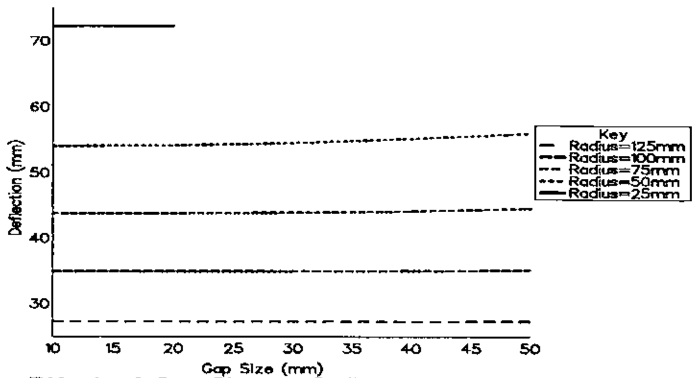
3. The very high stresses associated with the smallest radius above are also evident here.

5. Conclusions

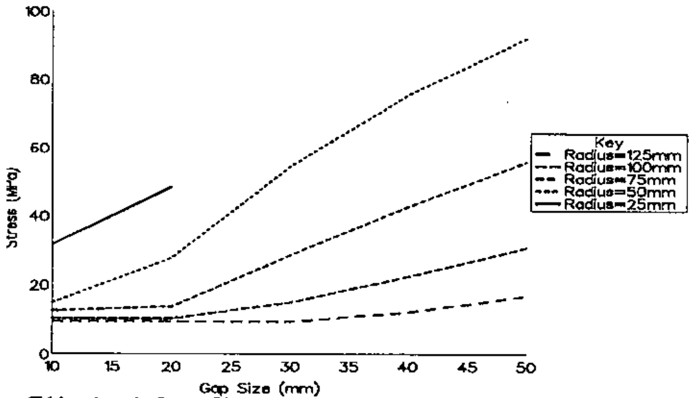
The above results show clearly the effects of the different design variables considered. The benefits of large radii are to reduce stresses in all areas, albeit with diminishing returns in some areas, whilst reducing overlaminate thickness reduces its through–thickness stresses, thus preventing premature delamination. This is in direct contrast to current practice. This decrease in overlaminate thickness also leads to increased flexibility, which reduces the effects of the "hard spot" the joint can create in the overall structure, but also leads to increased overlaminate in–plane stresses and fillet stresses, so clearly a compromise needs to be achieved.

Generally the results of this work have shown that the new design of Tee–joint is a favourable alternative to the current practice. Past results have indicated that a suitably refined joint can exceed the ultimate load carrying capacity of the current design by 150%, without being prone to premature delaminations in the overlaminate. This is achieved with a weight reduction of 60%, and, along with the potentially considerable savings in production costs, should ensure its further consideration.

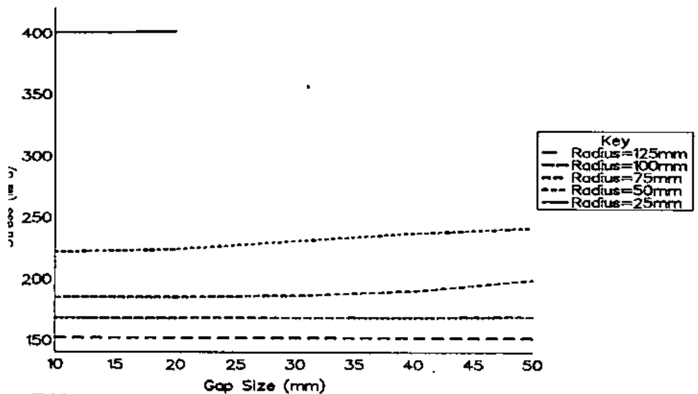
With the exceptions noted in the discussion of results above, the "production" variable (gap size) has a limited effect on the performance of the joint.



Effect of Gap Size on Deflection

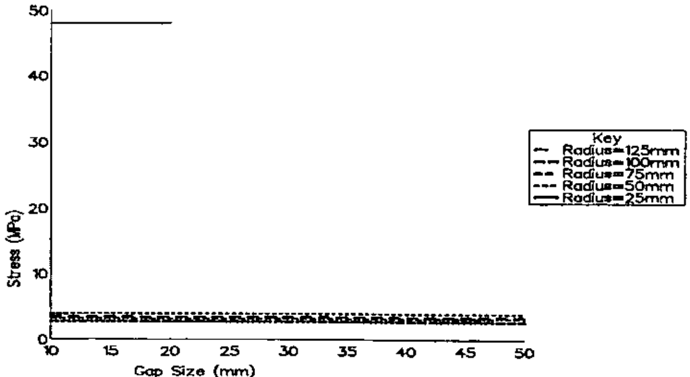


Effect of Gap Size on Stress in Fillet



Effect of Gap Size on Overlaminate In–plane Stress

Figure 4: Results for Variations in Gap Size



Effect of Gap Size on Overlaminate T–T Stress

However, Large gaps, greater than a quarter of the fillet radius, should be avoided to prevent excessive fillet stresses. Fillet radius and overlaminate thickness have significant effects on joint behaviour and can be optimised for a given application.

The graphs included in this paper can be used for a design optimisation procedure providing that failure information exists for the materials used. These are not available at present.

6. Acknowledgements

The work outlined in this paper was funded by the SERC (administered through the MTD Ltd.) and Vosper Thornycroft Ltd. Southampton.

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