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Design, Manufacturing and Testing of Hoop Wound Tubular In-plane Shear Specimens

by

Thomas E. Dickson

Thesis submitted to the School of Graduate Studies
in partial fulfilment of the requirements for the degree of
Master of Applied Science

in

Mechanical Engineering

Ottawa-Carleton Institute for
Mechanical and Aerospace Engineering
University of Ottawa

September 1993



Thomas E. Dickson, Ottawa, Canada, 1993



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Abstract

High performance fibre reinforced polymer composite materials have become popular replacements for many traditional materials. Unfortunately, some of the methods used to determine composite material properties have not developed as quickly. In particular, there are presently more than ten different test methods used to determine in-plane shear properties. Many of these methods are criticized for their inability to produce a pure shear state in the test region. Tubular in-plane shear specimens, however, are generally accepted as the only specimens which do (at least theoretically) produce a pure shear state. In this study, design, manufacturing and testing procedures were developed for tubular in-plane shear specimens. A specimen was designed which included a thin-walled composite tube and an internal tapered bonded end fitting. In the manufacturing study, techniques were developed to fabricate a tube of excellent quality with high geometrical tolerances. Tubular in-plane shear specimens were then tested and the results compared to test results for three of the most popular tests in use today, namely, the 90° Iosipescu, 10° off-axis and $\pm 45^\circ$ tensile in-plane shear test methods. The four tests were compared on the basis of their respective shear states and on their ability to produce shear modulus values which could be used to predict the tensile modulus of shear-sensitive laminates. Results indicate that, though the tubular shear specimens produce the most consistent state of pure shear, they do not accurately predict the tensile moduli of actual laminates. Of the five laminates tested, the $\pm 45^\circ$ tensile shear test best predicted the tensile modulus of four laminates and was next best for the remaining laminate. It was found that as laminate shear sensitivity decreased, the suitability of different test methods changed.

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Chapter 1.0

Introduction

1.1 Composite Materials

In today's competitive recreational, transportation and aerospace markets, there is continued pressure on manufacturers to use state-of-the-art materials. Fibre reinforced polymeric (FRP) materials are at the forefront of this movement. Composite materials are generally described as being comprised of two or more materials which are mechanically combined to utilize the advantageous properties of each material. Fibreglass composites, for example, combine strong brittle glass fibres with a tough but weak epoxy matrix to form a strong and tough material. Designers have begun to take advantage of this combination in using FRP's to supplement and even replace the more traditional materials.

The major advantage in using composite materials is their high strength to weight and stiffness to weight ratios. This makes them particularly attractive for use in the aircraft, automotive and recreational industries. Aircraft wing skins (CF-18), golf clubs, automobile body panels, satellite structural members (Radarsat) and even helicopter tail rotor drive shafts can be made from FRP's of one form or another. The specific modulus and specific strength are a measure of a materials strength to weight and stiffness to weight performance (Specific modulus = E/ρ ; Specific strength = σ_{ULT}/ρ). Figure 1.1 is a graph of the specific tensile strength versus

specific tensile modulus for various materials. It is worth noting that in this diagram steel, aluminum and the other bulk metals are in the bottom left corner of Figure 1.1 suggesting that they have low strength to weight and stiffness to weight abilities as compared to many composite materials. This does not indicate that metals should be replaced out of hand, rather it reinforces the value of FRP's when high strength or high stiffness and low weight are the main design considerations.

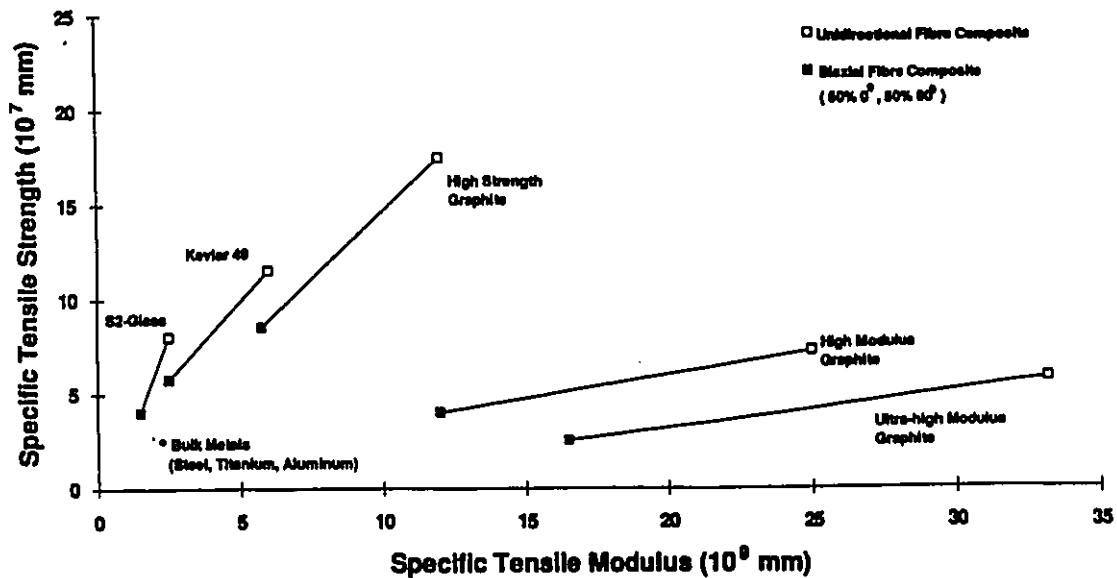


Figure 1.1 Specific strength vs. specific stiffness for various materials [1]

Another major advantage of FRP's is their ability to satisfy many specific load conditions. This tailorability is derived from the nature of how they are formed. FRP's are comprised of long or short high strength and stiffness fibres (graphite, carbon, glass, aramid) which are encased in a matrix or bonding material (thermosets, thermoplastics, ceramics, metals, etc.). The matrix protects the fibres from abrasion and transfers the external loads to the fibres. The fibres in turn provide high strength and stiffness along their length. FRP's are not typically strong and stiff perpendicular to the fibre direction or in in-plane shear. By orienting the fibres properly many loading situations can be accommodated.

The individual layers for the unidirectional and biaxial cases of Figure 1.1 are called laminae (Figure 1.2) and consist of fibres in one direction only. For aerospace FRP's these laminae are extremely thin (approximately 0.13 mm) and are thus of little structural value on their own. It is when the laminae are stacked one on top of the other to form laminates (Figure 1.3) that they reach their structural performance potential. The strength and stiffness characteristics of the laminate are directly related to the characteristics of the individual laminae, the thickness of the laminae, the order in which they appear in the laminate and the individual laminar orientations. By analyzing the strength and modulus of each lamina the strength and modulus of the entire laminate can be determined.

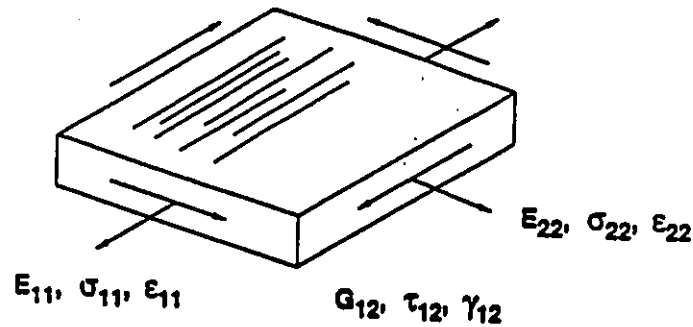


Figure 1.2 Representation of a FRP lamina.

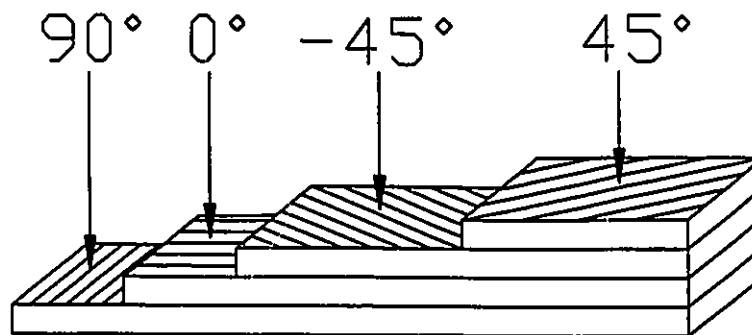


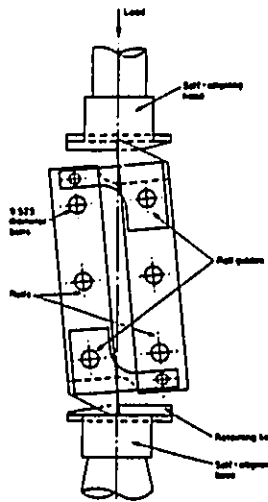
Figure 1.3 Typical laminate configuration (modified from [2]).

Classical Laminate Plate Theory (CLPT), Finite Element Analysis (FEA) and composite failure theories all use the properties of the individual laminae to calculate laminate strength and stiffness characteristics. There is therefore a need to determine the laminar material properties. Composites are considered to be orthotropic by nature, meaning the moduli have different values along each principal axis (Figure 1.2). In other words, there are many different material properties to be determined. These values are usually determined by experiment, however due to the rapid development of composite materials, experimental determination of laminar material properties is still a developing field. This is particularly true for in-plane shear properties.

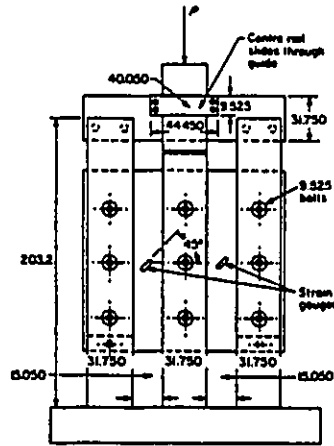
1.2 In-plane Shear Properties

The mechanical properties of an individual lamina which are required for structural analysis are generally determined using standard test methods. For example, there are accepted ASTM (and other) test methods for longitudinal tension, longitudinal compression, transverse tension and transverse compression. There is little controversy as to whether these test methods are appropriate for determination of longitudinal tension or transverse tension properties and some controversy with respect to determination of longitudinal compression and transverse compression properties. There is, however, considerable controversy regarding in-plane shear properties. This is evident, first of all, in the review in 1986 by Lee and Munro [3] in which nine substantially different in-plane shear test methods were evaluated (Figure 1.4). Secondly, since then, an additional test method has been proposed [4]. This controversy and resultant high level of activity is due to the difficulty in obtaining a state of pure shear in the test section. Lee and Munro [3] identified (using decision analysis techniques) three test methods ($\pm 45^\circ$ tensile, 10° off-axis tensile and Iosipescu) as the most promising for future study. These three test methods are particularly attractive because they have simple, easy-to-fabricate specimens and involve simple test fixtures and procedures.

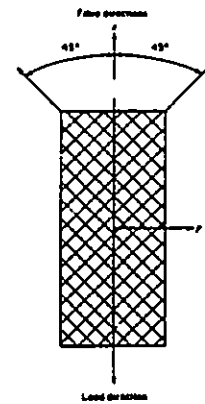
Both the $\pm 45^\circ$ tensile and 10° off-axis tensile tests can be conducted in a load frame much the same as any traditional tensile test. The Iosipescu specimen, however, must be placed in a special fixture prior to being installed in the load frame. Figure 1.5 shows the general



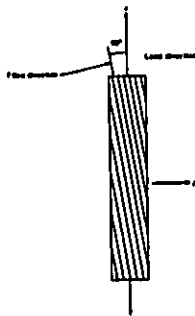
Two-Rail Shear



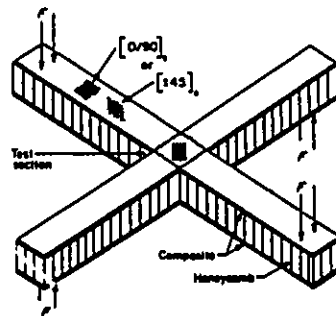
Three-Rail Shear



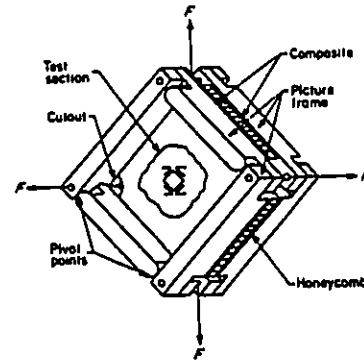
±45° Tensile



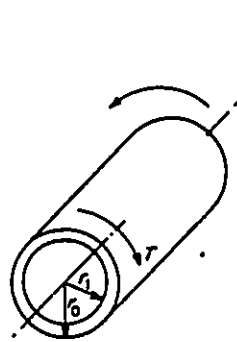
10° Off-Axis



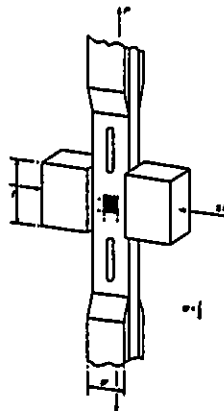
Cross-Beam



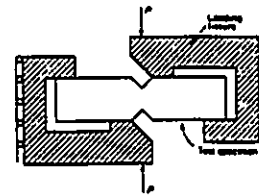
Picture Frame



Thin-Walled Tube



Slotted-Tensile



Iosipescu

Figure 1.4 Nine in-plane shear test methods [3].

configuration of these three test specimens. The $\pm 45^\circ$ specimen, as the name implies, is comprised of alternating laminae with fibres oriented at $\pm 45^\circ$ to the load axis. The 10° off-axis specimen is a unidirectional laminate with all fibres oriented at 10° to the load axis. The Iosipescu specimen has two different configurations, one in which the fibres are oriented perpendicular to the load direction (0° configuration) and one in which the fibres are parallel to the load direction (90° configuration). Lee and Munro [5] have shown that the 90° configuration is the more appropriate of the two for determining in-plane shear properties, thus, in later sections only the 90° Iosipescu specimen will be evaluated.

Despite the overall desirability of these three methods there are still problems with each method. Specifically, it is well known that neither the $\pm 45^\circ$ tensile nor the 10° off-axis specimens have a test region which is in pure shear. Both have significant normal stresses in addition to the in-plane shear stresses. Calculations by others [6,7,8] have shown that the Iosipescu test specimen still experiences normal stresses, although to a much lesser degree than the other two specimens (approximately 5% of the shear stress magnitude). Most of the research has concentrated on evaluating the individual test methods to check for shear uniformity using FEA [6,7,8,9,10], strain gauges [5,9], Moire interferometry [10,11] and in some cases to determine correction factors [6,7].

The basis for this work was the assumption that the most appropriate test method would be the one that best predicts the behaviour of a lamina in actual laminates. Lee and Munro have carried out an initial study on this basis [12] using a common aerospace laminate, $[45,0,-45,90]_{6S}$. Very little difference was found between the test methods even though the G_{12} values found with the three in-plane shear methods were significantly different. It was determined that the laminate selected for the study was not sensitive enough to small changes in the in-plane shear modulus; therefore, it was concluded that more shear sensitive laminate(s) should be used. The shear sensitivity of many different laminates was evaluated in [13] for use in shear test evaluation, though no tests were conducted.

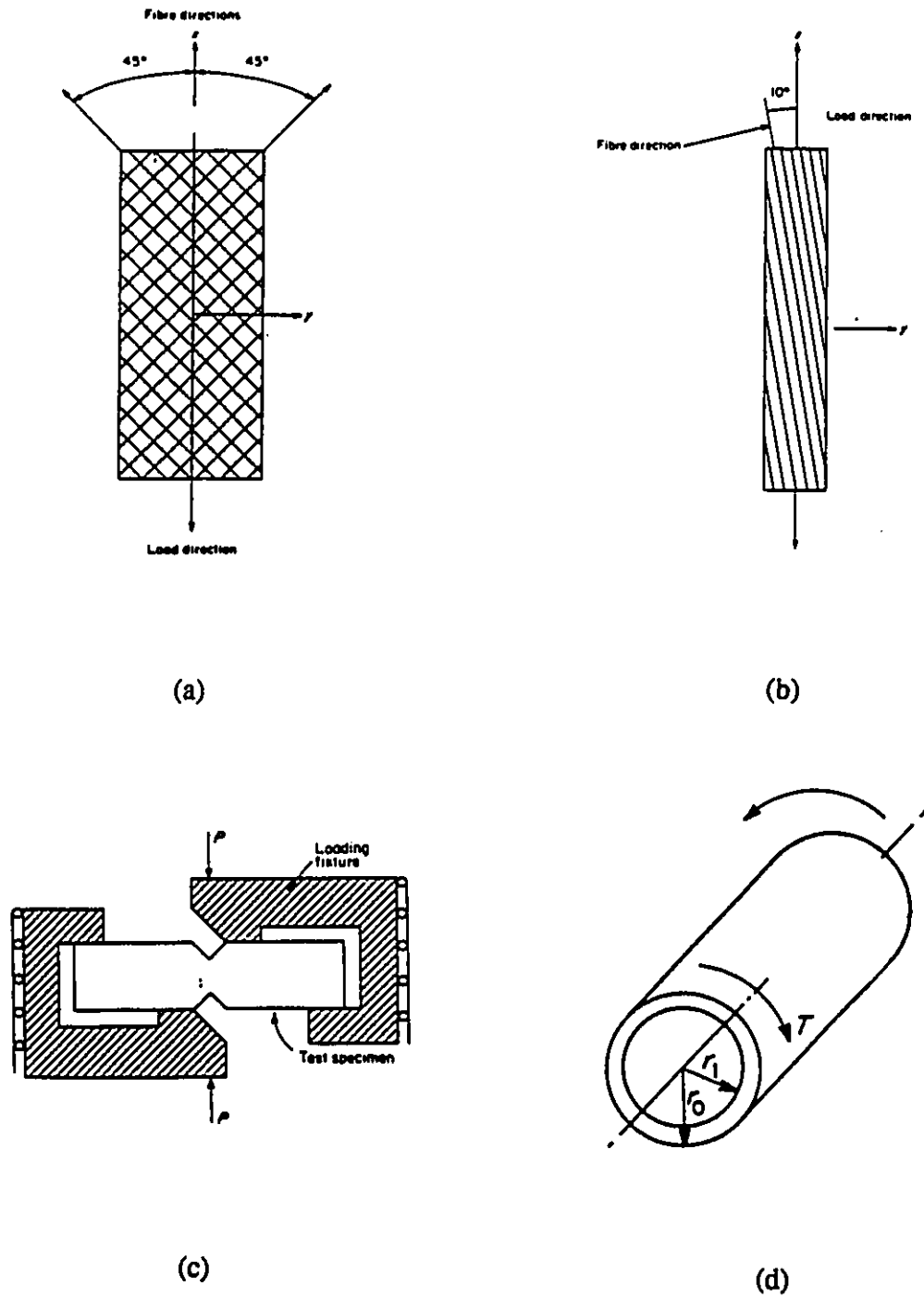


Figure 1.5 Configuration of the $\pm 45^\circ$ tensile (a), 10° off-axis (b), Iosipescu (c) and torsion tube tests (d).

The three test methods are still being actively investigated by other researchers. There is, though, general agreement that the torsion testing of a thin-walled circumferentially wound tube (Figure 1.5 (d)) results in a very uniform and pure shear state. This has been found by others, however, to be an expensive and difficult specimen to fabricate and, in particular, to test. Since it is reported to have the most uniform and pure shear state of all the specimens, it is appropriate to consider it further.

1.3 Problem Statement

The controversy regarding the test methods used to determine in-plane shear properties is addressed in this work. The first objective of this study is to develop the design, manufacturing and testing techniques for a tubular in-plane shear specimen. The second objective is to compare in-plane shear modulus and strength values using all four test methods. The selection of the most appropriate method will be done using the approach proposed by Lee and Munro [12] (the comparison of predicted and experimental laminate tensile moduli) with the exception that the comparison will be carried out for the four shear-sensitive laminates determined in [13].

Since tube design and manufacturing methods are new areas, it is necessary to review the literature on composite tube technology so that design and manufacturing procedures can be developed. Manufacturing details for tubular in-plane shear specimens have not been generally published in the open literature. Therefore, a manufacturing study will be undertaken with the goal to produce high quality in-plane test specimens using circumferentially filament wound tubes. Finally, test procedures will be developed for the preparation and testing of tubular in-plane shear specimens.

Some preliminary evaluations for this project [14,15] were conducted by undergraduate students. The initial results of this work have also been recently published [16].

Chapter 2.0

Review of Thin-Walled Tube Technology

2.1 Overview

The literature on thin-walled tube technology has been reviewed and will be summarized in the following way. The initial discussion will focus on past, present and future applications for composite tubes. Next, a review highlighting the important aspects of thin-walled tube testing will be given. Finally, the critical problem of end fittings will be presented in detail, resulting in the selection of the most appropriate type of fitting.

2.2 Thin-walled Composite Tube Applications

The use of composite torsion tubes as load bearing members has become more popular in the last ten years. In particular, composite tubes are used for some helicopter tail rotor drive shafts [17,18]. In addition, the LEAR FAN 2100 business aircraft used two such drive shafts to transmit power from two Pratt and Whitney Canada PT6 engines to a gearbox which in turn drove an aft propeller [17]. Work has also been done to gauge the effectiveness of composite tubes as drive shafts for ground vehicles [19]. The main problem with composite tubes in this application lies in the inability of the drive shafts to withstand impact loads [19]. For now at least, the aircraft industry is the largest user of composite torsion tubes.

There are several design benefits that can be gained when using composite tubes as drive shafts in place of steel or aluminum. First and most obvious of these is the weight reduction which can be achieved with composites. In addition to lessening the overall weight of the vehicle, long composite drive can have longer unsupported lengths than steel or aluminum shafts which can deflect under their own weight [17]. This benefit is enhanced by the ease with which composites can be tailored to avoid this deflection should it become a problem. Another advantage which composites bring to the design of drive shafts has to do with the ability to tailor them for specific torque and speed applications. Furthermore, lateral stability of the shaft, which is dependent upon rotational speed of the shaft, can be increased depending on the tube laminae stacking order. Typically, the rotational speed of the shaft must be lower than the first natural bending frequency of the shaft [17,19]. By increasing the number of laminae with fibres running along the tube length (0° plies), the first natural bending frequency can be increased and lateral stability can thus be increased [19]. Therefore, with greater lateral stability longer unsupported lengths can be used. There is of course a balance to be maintained between the number of 0° plies which can be used and the torsional properties.

The prevailing attitude with respect to designing these tubular drive shafts is to use the available theoretical design approaches with appropriate safety factors to yield shaft dimensions. Typically, the maximum speed of the shaft is known so that the longest unsupported length of the shaft can be determined using

$$f_{CRIT} = \frac{\pi}{4\sqrt{2}} \frac{D_M}{L^2} \sqrt{\frac{E_L}{\rho}} \quad (2.1)$$

where f_{CRIT} is the critical rotational frequency, D_M is the mean diameter of the tube, ρ is the material density, E_L is the modulus along the tube length and L is the unsupported length of the tube. The thickness of the tube and necessary laminar orientations can then be determined from stress considerations [17,19]. The tube can buckle under torsional load and thus this must be considered as well. It is clear, though, that in order to effectively design a drive shaft or any

torsional component, a more detailed knowledge of simpler tubular components is appropriate. A review of the design and testing of simple tubular torsion specimens will provide insight for the more complicated design situations.

2.3 Overview Of Thin-Walled Tube Testing

Thin-walled composite torsion tubes have become more attractive to designers in recent years because of the promise of a lightweight alternative to traditional structural and power transmission components. In addition, thin-walled torsion tubes are ideally suited for determining the in-plane shear properties of composite laminae. It is not surprising then to find that this type of composite structure has been studied both analytically and experimentally for tensile, compressive, bending and torsional loading situations for almost as long as fibre reinforced composites have been used. The use of these tubes in torsion is of special interest since a main objective of this project is the determination of composite in-plane shear performance.

Proper assessment of tubular torsion components, or assessment of in-plane shear properties, requires the development of reliable manufacturing and test techniques. Torsional cylindrical specimens experience different types of failure depending on the ratios of both tube length and thickness to tube diameter. Buckling, for example, occurs if the specimens are too long or too thin in relation to the tube diameter. This must be avoided if true material failure is to occur during torsional testing. Two other important points involve the end fittings which are used to attach the specimen to the torsional loading equipment. They must be designed so that stress levels near the fittings do not cause premature tube failure yet are strong enough to cause tube failure rather than joint failure. An understanding of tube behaviour in torsion is thus necessary so that appropriate design and test parameters are used.

Researchers have been developing tube technology, however, no one set of design and test techniques has been compiled which will assure proper material failure. Combinations of selected aspects of these design and test techniques though, may provide the best possible approach to tube design and testing. The following is a review of the state of composite

specimen design and tube testing including proper determination of tube dimensions, design of composite tubular joints and the behaviour of both in-plane shear specimens and general laminates.

2.4 Design Of Tubular Specimens

There are two major problems which might adversely affect the validity of in-plane shear test results. First, stress gradients which promote premature failure can exist within the gauge section. Tube dimensions, fibre orientation, degree of tube anisotropy and loading and clamping conditions all affect these gradients [20]. The second major problem which can occur during testing is torsional buckling. The presence or absence of this during testing is dependent entirely upon the specimen dimensions, material stiffnesses and the angle between the fibres and the longitudinal axis of the tube. These two effects will be dealt with individually.

2.4.1 Effect Of Stress Gradients On Tube Dimensions

The goal when designing a tube to be used for in-plane shear testing (or any kind of tube testing) is to select those dimensions and end constraints which would produce uniform stresses through the wall thickness and uniform stress in the gauge section simultaneously. This is driven by the need to ensure that premature failure does not occur. In addition, the measuring devices (strain gauges), which are mounted on the outer surface of the tube, will measure the surface strain up to failure which can be used to determine the elastic modulus. The outer surface might fail and destroy the gauge when the remainder of the tube has not yet failed. Furthermore, since flat specimens under shear may not experience the stress gradients found in tubes, using tubes to find material properties becomes a questionable decision. It is also true, however, that flat specimens are subject to edge effects which tube specimens are not. In order that tubes can be used, stress gradients in tubes must be eliminated.

Based on an analysis of anisotropic tubes pioneered by Sherrer [21], Whitney and Halpin [22] developed a technique which uses a combination of laminate theory and shell theory to

predict stresses in anisotropic tubes under various loading conditions. Shell theory is an approximate approach, however, a comparison with an exact elasticity solution [23] showed that shell theory can in fact be used to successfully predict stresses in thin tubes. This analysis provides the basis for the linking of tube dimensions to the variation of stresses through the wall thickness of the tube. Specifically, as t/D_i becomes small, the stresses become nearly uniform through the thickness [20,23,24,25]; t is the tube thickness and D_i is the inner diameter of the tube. In addition, the selection of t/D_i can be made for orthotropic symmetric tubes under torsional loading based on the analysis of an isotropic tube [20,23,26]. The only constraint for this assumption is that G_{12} must have the same value for each ply [23]. This type of analysis is only valid for $t/D_i < 0.05$ and $l/D_i > 3$ [20,25,27], where l is the tube gauge length.

Pagano and Whitney [20] using the shell theory approach conducted a detailed study into the effect of various parameters on the distribution of stresses in both the through thickness and axial directions. The purpose of their study was to define a tubular specimen geometry, using t/D_i and l/D_i as the main parameters, such that the stress gradients in the gauge section are small. Both t/D_i and l/D_i are functions of known material constants. The procedure consists of first reducing the stresses in a typical tube cross-section to equivalent moments and forces (Figure 2.1). A length, ϵ , is then defined over which the stresses are non-uniform (Figure 2.2). The ϵ value is the length which is necessary to reduce the equivalent forces and moments above to specified small values. One selects the value of t/D_i (using the existing closed form shell theory solution [22,24]) for a low through thickness stress gradient. This value is then used in the calculation of ϵ . The tube dimensions can thus be determined for a given set of material properties and state of applied loading.

Pagano and Whitney, after using this approach to finding desirable tube dimensions, went on to study factors which affect the stress gradients [20]. In general, the higher the anisotropy ratio, E_L/E_T (E_L is the tensile modulus along the tube length and E_T is the tensile modulus transverse to the tube length), within the tube the higher will be the stress gradients during loading. As the magnitude of t/D_i becomes smaller the stress distribution through the thickness becomes more uniform. For a unidirectional tube in torsion the through-the-thickness stress

gradients vary with fibre angle, where a 60° fibre orientation yields the highest stress gradients. Figure 2.3 shows that the normalized stress on the outside of the tube (maximum stress) approaches unity near the 90° fibre orientation for $t/D_i=0.025$. This is an indication that through-the-thickness stress gradients should be acceptably low for hoop wound in-plane shear specimens. Stresses due to end constraints were also found to dissipate after a length equal to the tube diameter. Rizzo and Vicario [27] substantiated this last finding. Finally, during torsional loading angular displacement is not uniform with respect to the tube length until a distance equal to the tube diameter away from the specimen end has been reached [20].

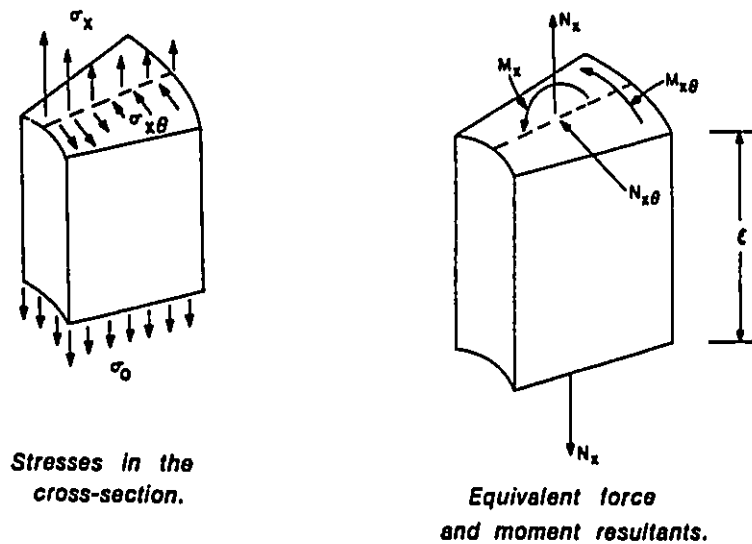


Figure 2.1 Schematic view of elemental stresses in the form of equivalent forces and moments [20].

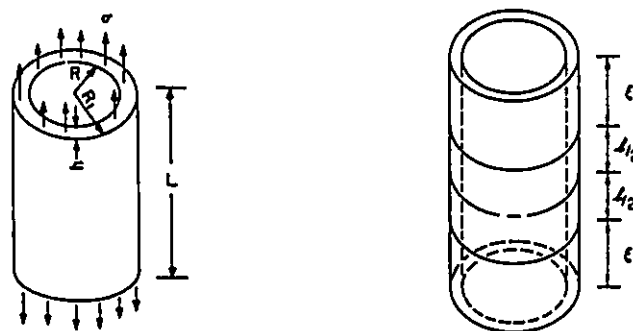


Figure 2.2 Nomenclature used for critical length determination [20].

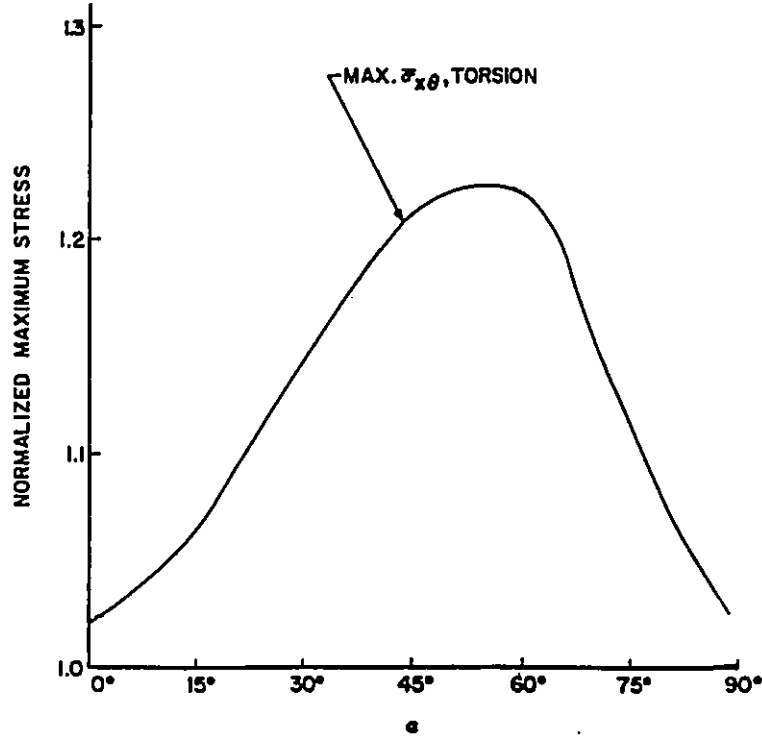


Figure 2.3 Variation of shear stress level vs. fibre angle (modified from [20]).

There are two very useful findings from all of the above work which are helpful in determining in-plane shear specimen geometry. First, there is a conservative estimate of the length required to dissipate the increased stress at the tube ends due to end constraints. With a gauge length of the tube equal to or greater than twice the tube diameter end effects should not reach the centre of the gauge section where measurements are made [20,24,27,28]. In essence, the length of the tube is defined by

$$l = 2D_i + l_g \quad (2.2)$$

where l is the unsupported tube length and l_g is the desired gauge section of the tube. Secondly, the stress distribution through the thickness can be assumed to be uniform for a tube with a t/D_i between 0.025 and 0.05 [24,26,27]. In addition, for $t/D_i < 0.05$ the through-the-thickness stress distribution can be assumed equal to the stress distribution for an isotropic tube with the same

dimensions [24,26]. The expression for shear stress through the tube thickness is a simple one and is given below [23,26]:

$$\frac{\tau_{MAX}}{\tau_r} = (1 + \frac{t}{D_m}) \quad (2.3)$$

where τ_{MAX} is the stress at the outside surface of the tube, D_m is the mean diameter (nearly equal to D_i for tubes of concern to this study) and τ_r is the stress at the mean thickness. By plotting the stress ratio as a function of t/D_m a design curve is generated. An example of such a curve appears in Figure 2.4. The stress ratio gives an indication of how much the stress varies linearly as the relative thickness of the tube increases. By combining the length requirement from equation 2.1 with the thickness selected from Figure 2.4 a general tube design can be developed.

Foley et. al. [26] have since developed an expression which can be used for thick as well as thin laminates. A thin-walled tube is defined by assuming the ratio t/D_m is small compared to unity. This enables the elimination, during the stress analysis, of terms containing t/D_m [22,23,26]. In thicker laminates these terms become increasingly important and the stress ratio begins to differ from the isotropic or thin tube predictions, although the distribution is still taken to be linear in nature [26]. As is apparent below, there is an additional term in the more general expression of equation 2.3.

$$\frac{\tau_{MAX}}{\tau_r} = (1 + \frac{t}{D_m}) [1 + \frac{1}{6} (\frac{t}{D_m})^2] \quad (2.4)$$

Figure 2.5 shows typical results for the equations 2.3 and 2.4 for $\pm 45^\circ$ laminates plotted together. It is clear that there is only a significant difference at high values of t/D_m , indicating that the isotropic expression is accurate enough for most in-plane shear testing applications.

The boundary conditions of the tube analysis above considers the end constraints to be applied at the very end of the tube. In reality, the end fittings which transmit torque to the tube

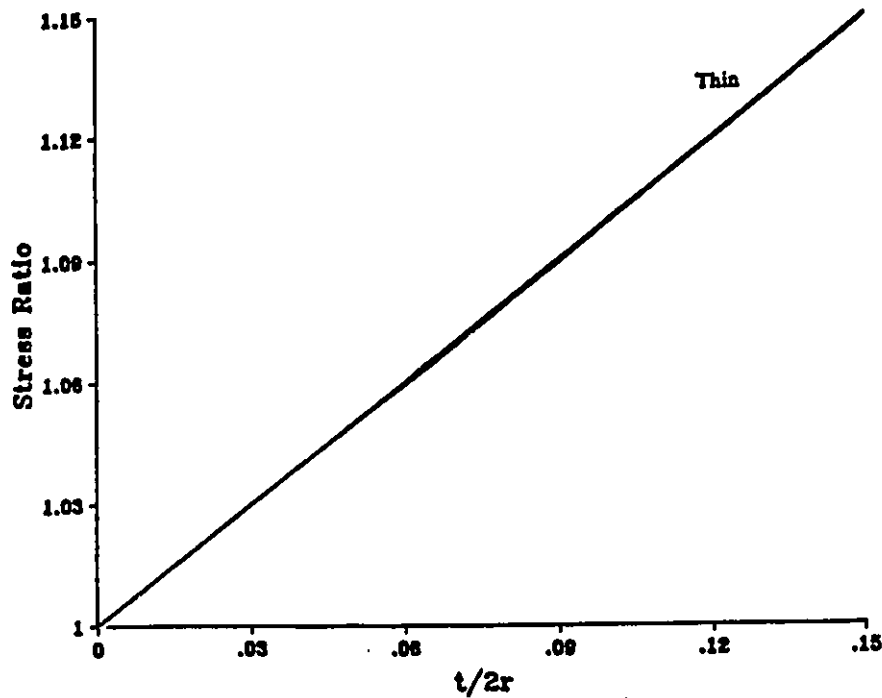


Figure 2.4 Design curve for an isotropic tube found using equation (2.3) (modified from [26]).

act over a certain length. For example, the end fittings might be 50 mm long. The transmission of load to the tube acts over the entire 50 mm. In this situation the end constraints might stress the tube to a greater extent than was considered under the above analysis [27]. In addition, the analysis does not consider the contribution of the interlaminar shear stresses, τ_{θ} and τ_r . Rizzo and Vicario [27], using a finite element technique studied these areas to see if the effect on tube modulus determination or failure determination is significant. The geometry of the tube and the relevant parameters appear in Figure 2.6.

The advantage of using the finite element analysis of Rizzo and Vicario is that it has the ability to make stress determinations for the case of general anisotropy with all shear coupling effects. These were neglected in the shell analysis [22,23,24,27]. This finite element approach can deal with complex geometrical situations and boundary conditions allowing it to accurately determine gripping effects along the tube surface. Since this approach does not lend itself to the

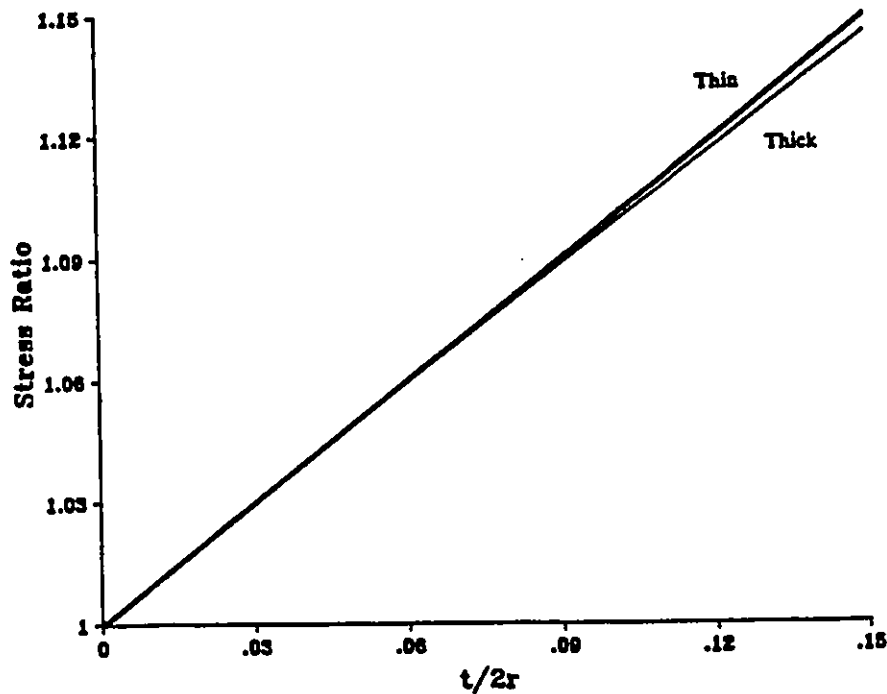


Figure 2.5 Variation of through thickness stresses for isotropic and anisotropic tube [26].

determination of shear properties from experimental data or conveniently lend itself to design, the main purpose of Rizzo and Vicario's work was to determine the conditions under which the shell theory analysis is applicable.

There are several interesting results of the analysis of gripped tubes. They reported a definite coupling between stress components during torsional loading [27]. In particular, there were significant average axial stress components induced by torsional loading as is shown in Figure 2.7. During in-plane shear testing it is important that a state of pure shear exist in the test area, thus, this additional stress must be considered when using torsion tubes as in-plane specimens.

Also evident in Figure 2.7 (a) and (c) is the effect of varying t/D_i . As t/D_i decreases from 0.05 to 0.015 there is a substantial decrease in the variation of shear stress from the inside of the tube to the outside of the tube. Even at a thickness ratio of 0.015 the shear stress varies by approximately 10% which indicates that the variation in shear stress predicted by Foley et. al. in Figure 2.5 is too low.

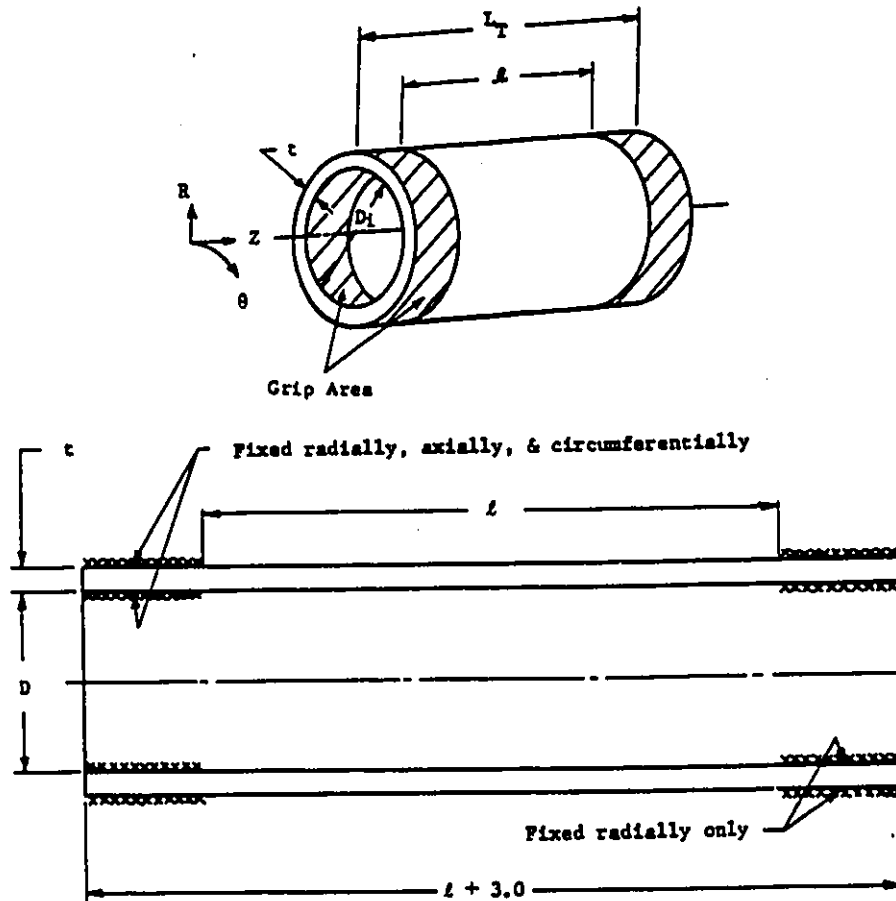


Figure 2.6 Rizzo and Vicario's tube geometry [27].

Despite the disparity in shear stress between the inside and outside of the tube, shear modulus values found using the finite element simulation compared favourably with those found analytically [27]. For a value of $t/D_i = 0.05$, a difference of 5% was found between the model of Pagano and Whitney [20] and the finite element analysis of Rizzo and Vicario. Thinner tubes had lower differences; the $t/D_i = 0.015$ tube had only 1% difference between techniques.

One area which was not well modelled by analysis was the stress distribution under the grips. There is clearly a stress peak caused by the application of torque through the grips to the tube. This peak increases in magnitude with increasing thickness. This sudden increase is most apparent in the axial stress results of Figure 2.7. The stress peak could cause premature failure. A solution to this problem would be to thicken the portion of the tube under the grips and then

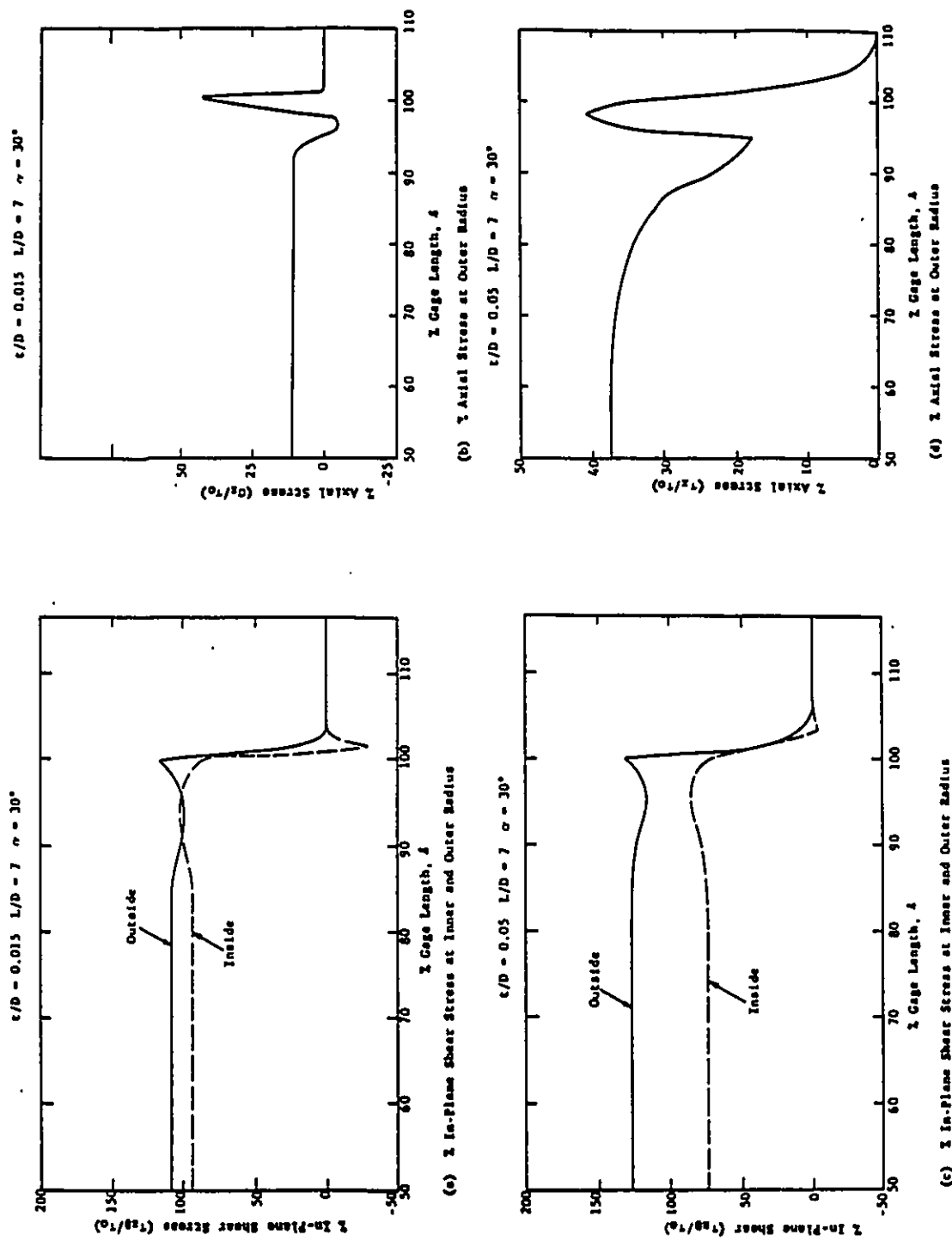


Figure 2.7 Variation of shear stress across tube thickness and axial stress induced during torsional loading [27].

taper down to the thickness of the gauge section. This is similar to the end tabs used for many flat bar specimens.

Based on these results the optimum ratio, t/D_1 , is in the range of 0.015 to 0.05 while the restriction on the length remains, as was mentioned earlier, as $l/D_1 > 3$.

2.4.2 Buckling Behaviour Related to Tube Design

The second major problem to consider in tube design is the factors which affect buckling behaviour. As discussed in the previous section, as the wall thickness of the in-plane shear tube specimen decreases, the more indicative the test results will be of pure shear loading. In practice, however, for a given geometry, there will be a threshold thickness below which torsional buckling will occur. It is extremely important that the possibility of buckling be eliminated since failure levels found in the presence of buckling do not constitute a material failure. Buckling is directly affected by the dimensional parameters t/D_1 and l/D_1 and the fibre orientations [26,28,29-32].

Buckling is an elastic phenomenon accompanied by an immediate drop in load carrying capacity of the tubular specimen [19,26,31]. If the torque is reduced immediately after the onset of buckling, the tube will return to its original state and will buckle again at the same load levels each time [31,32]. This phenomenon is visible in compliant composites [26], however, in more brittle materials, buckling and tube failure occur almost simultaneously. The detection of buckling is thus an important concern for the in-plane experimental procedure. Although buckling is not always visible, strain gauges can be used to detect its occurrence [26,31,32].

Typically, for the detection of buckling a stacked rosette ($0^\circ/45^\circ/90^\circ$) strain gauge is attached to the tube with the 45° arm of the rosette parallel to the tube axis [26]. In this way one of the remaining two arms is in tension while the other is in compression. Prior to buckling both compression and tension gauges show a linear increase in stress with respect to strain. Upon buckling, however, the tensile arm shows a sharp decrease in strain while the compressive

strain rapidly increases [26]. Figure 2.8, taken from [26], illustrates this behaviour for S-glass-epoxy composites. The strain gauges may not react exactly this way for all cases, but sharp changes in gauge response of one sort or another will almost certainly accompany buckling.

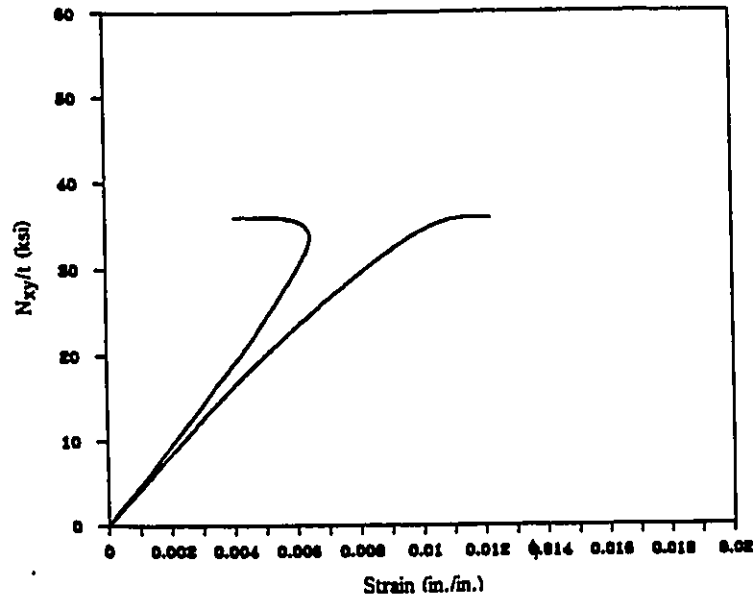


Figure 2.8 Shear stress versus strain for tensile and compressive arms of strain gauge during buckling [26].

Buckling is predominantly sensitive to the relationship between tube gauge length, tube thickness and tube diameter. There is no empirical relationship that can be followed since the orientation of the fibres make each different laminate a completely new problem. Instead, there are several analytical approaches which if used with caution can be beneficial in determining the limiting length and thickness to diameter ratios.

Most classical approaches to the buckling of thin fibre composite tubes assume a linear stress-strain relationship [30]. This is not true in many cases for composites, especially for shear loading. In addition none of the existing approaches has been used to develop a definitive buckling model. The following review highlights a number of buckling theories but does not cover all the work done in this field; an extensive review has been done by Tennyson [30].

Several researchers have developed buckling theories and yet there are conflicting reports as to the ability of these models to accurately predict critical buckling loads. Cheng and Ho, for example, developed a general model for the buckling of anisotropic tubes based on Flugge's shell theory [33,34]. Tennyson [30] reported that for compression buckling this theory provided good results but for torsional buckling it did not. Tennyson also compared his own torsional buckling experimental data to a buckling model based on Donnell's shell theory and found that they agreed to within $\pm 18\%$ [30]. Bauchau et. al. [19] developed a model based on a general shell theory (no simplifying assumptions) to predict buckling loads and they found good agreement between theory and experiment. Chao [35] employed Timoshenko's buckling equilibrium equations to develop a buckling criterion. The predictions were substantiated by Marlowe et. al. [32] while considerable differences between theory and experiment were reported by Wall and Card [29]. The selection of an appropriate model from the above methods would require a large scale experimental evaluation which falls outside the scope of this work.

As mentioned above, most of the predictive buckling theories do not always accurately predict the critical buckling loads. It should be noted that these techniques rely heavily on the assumption that the shear stress-strain relationship is linear and this is not often correct when dealing with composite structures [5,30]. This may result in discrepancies between theory and experiment and could also explain the varying degrees of success these analytical models have met. The purpose of this study is to find a simple relation or set of relations, if possible, which can easily be used to optimize tube design from known material constants. The above mentioned models and others discussed in [30] do not fill this requirement.

There are, however, several relationships reported in the literature which do lend themselves to specimen design. If it is possible to treat composite tubes (with $l/D_i > 3$ and $t/D_i < 0.05$) as isotropic tubes, it is possible to use isotropic simplifications of buckling theory. Patterson [36] reported two expressions developed by Timoshenko and Gere:

$$\tau_{CR} = \frac{Et^2}{l^2(1-\nu^2)} \left(7.8 + 1.67 \left(\sqrt{1-\nu^2} \frac{l^2}{2tr} \right)^{\frac{3}{2}} \right) \quad (2.5)$$

$$\tau_{CR} = \frac{4.39Et^2}{l^2(1-\nu^2)} \sqrt{1.27 + 0.0257(1-\nu^2)^{\frac{3}{4}} \left(\frac{l}{\sqrt{tr}} \right)} \quad (2.6)$$

Patterson [36] also reported two other expressions; one by Roark and Young and another by Simitses.

$$\tau_{CR} = \frac{Et^2}{l^2(1-\nu^2)} \left(1.27 + \sqrt{9.64 + 0.466 \left[\sqrt{(1-\nu^2) \frac{l^2}{tr}} \right]^{\frac{3}{2}}} \right) \quad (2.7)$$

$$\tau_{CR} = \frac{0.761Et^{\frac{5}{4}}}{r^{\frac{3}{4}}l^{\frac{1}{2}}(1-\nu^2)^{\frac{5}{8}}} \quad (2.8)$$

In all cases E is the tensile modulus in the axial direction of the tube, ν is Poisson's ratio, l is the unsupported tube length, t is the tube thickness and τ_{CR} is the shear stress required to buckle the tube.

Ellis and Hoskin [31] used expressions developed by Hayashi [37] for torsional buckling of an homogeneous orthotropic material where one of the axis of orthotropy must be parallel to the tube axis. They reported buckling loads in excess of experimental values for the $[\pm 30^\circ, 90^\circ]$ laminates studied while good results were found for the $[\pm 45^\circ]$ laminate tubes. These expressions, as those above, are in closed form which make them desirable for design

calculations. Based on the assumption that the ends are clamped, a basis for comparison, N , is calculated:

$$N = \left(\frac{1}{7.8} \right) \left(\frac{l}{D_i} \right)^2 \sqrt{\frac{E_L}{(1 - \nu_L \nu_T) E_T}} \quad (2.9)$$

where l and D_i are defined as before, E_L is the tensile modulus along the length of the tube, E_T is the tensile modulus transverse to the tube length, and ν_L and ν_T are Poisson's ratios along the tube length and transverse to the tube length. One of two expressions for τ_{CR} is used depending on whether D/t is less than or greater than N .

a) $D/t > N$

$$\tau_{CR} = \left(\frac{E_T}{(1 - \nu_L \nu_T)} \right) \left(\frac{D_i}{l} \right)^2 \left(\frac{t}{D_i} \right) \left(4.6 + \sqrt{7.8 + 1.67 H^{\frac{3}{2}}} \right) \quad (2.10)$$

where

$$H = \left(\frac{l}{D_i} \right)^2 \left(\frac{D_i}{t} \right) \sqrt{\frac{(1 - \nu_L \nu_T) E_L}{E_T}} \quad (2.11)$$

b) $D/t < N$

$$\tau_{CR} = 0.77 \left(\frac{t}{D_i} \right) \left(\frac{E_L^{0.25} E_T^{0.75}}{(1 - \nu_L \nu_T)^{0.75}} \right) \quad (2.12)$$

By selecting values of D_i , t and l , a suitable tube which does not buckle can be designed for in-plane shear testing.

Aside from the tube dimensions, there is another factor which influences the buckling strength of the tubular specimen, namely, the angle between the fibres and the tube axis [19,26,29,31,32]. The $\pm 45^\circ$ laminate for example, is particularly strong in shear while the unidirectional 90° laminate is not [13]. It is not surprising then that the $\pm 45^\circ$ laminate has a greater tendency toward buckling than does the 90° tube which has a comparatively low failure stress. The Hayashi [37] formulation takes this into account whereas the isotropic equations are less sensitive in this respect. Of particular note is the fibre orientation in the outside layer of the tube [19,30,32]. When the tubes are loaded such that the outer layer fibres are in compression, buckling occurs at much higher loads than in the case of the outer fibres being in tension [30,32]. Marlowe et. al. [32] reported that this effect can be predicted by Chao's analysis, however, the Hayashi model as well as many others do not.

2.4.3 Summary and Conclusions for Design of Tubular Specimens

It is clear that design of in-plane shear specimens must include the consideration of both stress gradients and torsional buckling. By using both the results of the linear elastic shell theory analysis [20,22-24] and one of the buckling models one can design a specimen which has acceptable stress variations and which does not buckle before static failure occurs. A desirable approach would be one which includes a closed form buckling model since this would reduce the necessity for a numerical solution indicative of the more complex approaches. The Hayashi formulation as used by Ellis and Hoskin [31] yielded fair results for thin carbon fibre reinforced composite tubes which are of simple laminate configuration. For this study a simple a 90° unidirectional lay-up was to be used, making the Hayashi model an acceptable choice for buckling predictions. As an added precaution, where possible, tube dimensions similar to those used by Ellis and Hoskin were selected (eg. tube diameter). In addition carbon fibre reinforced epoxy was chosen as the material to be used. One tube design approach would be to first select a tube diameter. Since the laminae have a fixed thickness, the thickness can only be a multiple of the individual laminae thicknesses. A practical thickness which conforms to the t/D_i restriction can be determined once the laminate configuration is known. It is then a simple matter to select a gauge length that satisfies $l/D_i > 3$ and does not cause buckling.

For the design of tubular composite specimens to determine the in-plane shear modulus, G_{12} , and the in-plane shear strength, τ_{12} , the consideration of the effect of fibre orientation may not be critical. The tubular specimen will have fibres oriented at approximately 90° to the tube length for all layers. In the case of stress gradients in the through thickness direction and in the case of stress gradients in the gauge section due to end constraints, the 90° fibre orientation is the weakest with respect to shear [13] and is thus the most likely to fail prior to buckling for a given set of tube dimensions.

2.5 Tubular Joints For Specimen End Fittings

The joint which connects composite tubes to metallic end fittings must be designed with careful consideration of the load and environmental conditions. This can often lead to the necessity for very strict design specifications [17]. The end fittings for transmitting torsional loads to the in-plane shear specimen must be designed much like any other tubular joint which transmits torque. With this in mind, a review of the state of the art in joint and end fitting technology was conducted.

The requirements of the end fitting to be used for in-plane shear specimens are few but stringent. Foremost among these are that the end fitting cannot cause premature failure through stresses developed by the joint and that it must not affect the stress state in the designated gauge region. Secondly, the joint must not itself, during testing, fail prior to the specimen in the gauge length. The joint should be re-usable in the sense that after the tube has failed the removal and re-attachment to another tube should not be difficult. Finally, the joint design must be a simple one to manufacture either by filament winding or hand lay-up and one which facilitates easy attachment to the specimen.

Fuchs et. al. [17] has presented some typical joints which could be used to support torsional loads. These included bonded joints, pinned joints, compression/friction joints and polyhedron joints (Figure 2.9). Another possible fitting design includes the combination of bolting and bonding techniques [31,38]. Each of these constitutes a possible end fitting design

and each will be discussed in turn.

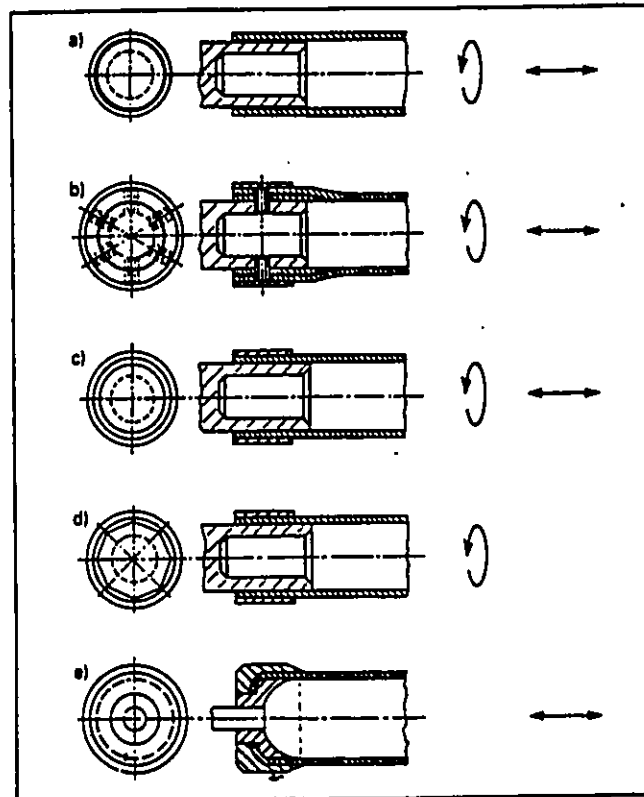


Fig. 16. Fitting design

- | | |
|-------------------------------|-----------------------|
| a) bonded joint | d) polyhedron |
| b) pinned joint | e) form-locking joint |
| c) compression/friction joint | |

Figure 2.9 Approaches to tubular joint design [17].

2.5.1 Bonded Composite/Metallic Tubular Joints For Torsional Applications

Whitney et. al. [39] determined that a joint which deformed somewhat during loading could help reduce the problem of high stress concentrations caused by the end fitting. Adhesively bonded joints can be compliant while at the same time maintaining a strong bond. This is possibly one reason for the popularity of such joints in the area of tube testing. Several types of bonded joint can be used, namely, bonded scarf joints, single tubular lap joints and simple bonded end plates.

2.5.1.1 Bonded Tubular Lap Joints

Bonded tubular lap joints (Figure 2.10) are the most studied and best understood of the bonded joints mentioned. The simplicity of this joint is readily apparent. Lap joints with the metallic component outside the composite shaft also exist and there is no reported difference between the two. The tubular lap joint has been studied [38,41] to determine the effects of various factors on joint efficiency and to determine design guidelines. The results of these analyses are summarized below.

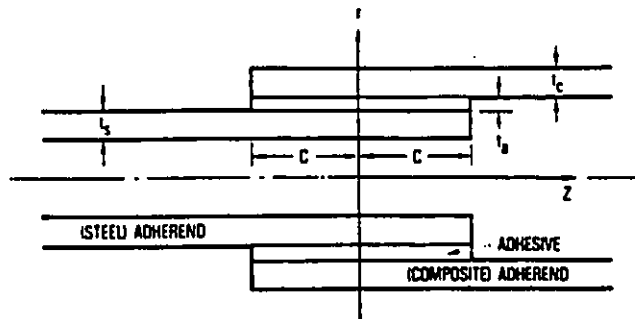


Figure 2.10 Schematic view of a typical bonded lap joint [37].

Through the use of parametric studies, Hipol [41] was able to identify several general trends with respect to shear stress behaviour in the adhesive layer of tubular bonded lap joints. The first point of note is that the stress distribution within the adhesive layer reaches a maximum in the end of the joint from which the least stiff of the two adherends protrude. Furthermore, the distribution is asymmetric with a second stress peak at the other end of the specimen joint. This finding was substantiated by the work of Heguera and Murat [42] and Habib et. al. [40]. The latter work also showed that stresses vary through the adhesive thickness (Figure 2.11). Hipol [41] also found stresses to vary through the joint thickness with the maximum occurring in the adhesive layer.

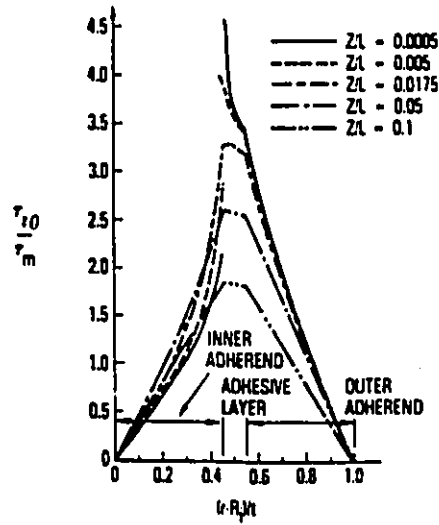


Figure 2.11 Stress distribution through the thickness of the adhesive layer of a tubular bonded lap joint in torsion (L is the total bond length, Z is the axial position) [41].

Factors which directly affect the design of tubular lap joints have also been identified. Hipol [41] determined that a ratio of shear stresses was proportional to a ratio of adhesive thicknesses:

$$\frac{\tau_{r0}}{\tau_b} \propto \sqrt{\frac{t_b}{t_a}} \quad (2.13)$$

where τ_{r0} is the maximum shear stress in the adhesive layer, τ_b is the maximum shear stress in the adhesive layer for a baseline condition, t_a is the thickness of the adhesive layer and t_b is the adhesive thickness for the baseline test. The advantage of this expression is that the maximum shear stress can be found for a given adhesive thickness and hence a shear stress capacity can be determined. Unfortunately, Hipol's work has not been verified experimentally.

Figure 2.12 shows the effect of changes in stress in the adhesive layer as a function of adhesive thickness; the thicker the adhesive layer, the stronger the joint [41]. Hipol's expression (equation 2.13) implies this as well.

A second variable of interest for the design of lap joints is the amount of overlap in the joint. Figure 2.13 shows the variation of stress as the overlap length is increased [41]. The stress maximum is not affected significantly as the overlap length is increased [41]. There is evidence, however, which suggests that the stress in the middle of the joint reaches zero at some critical value of overlap length. For the purposes of in-plane shear specimen testing, the maximum stress is the limiting design criterion and therefore the amount of overlap is of lesser importance.

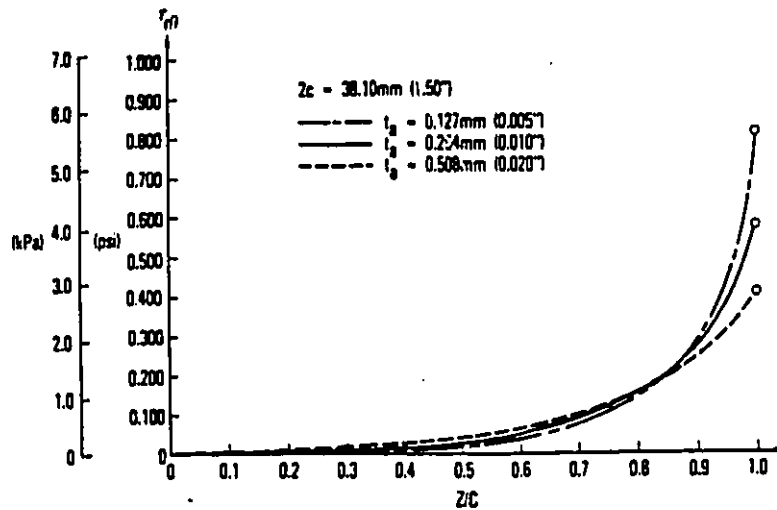


Figure 2.12 Variation of stress distribution with changing adhesive thickness [41].

The relative stiffness of the adherends plays an important role in joint stresses. As the difference between the stiffnesses of the adherends decreases the stress maximum in the adhesive layer decreases [41-43]. Figure 2.14 shows this by comparing several composite materials with different end fitting materials and a steel-steel joint. Clearly the best combination is the steel-steel joint (ie. the smallest stress maximum is reached when identical materials act as the adherends [42,43]). Furthermore, the more similar the material stiffnesses, the lower the stress level [41].

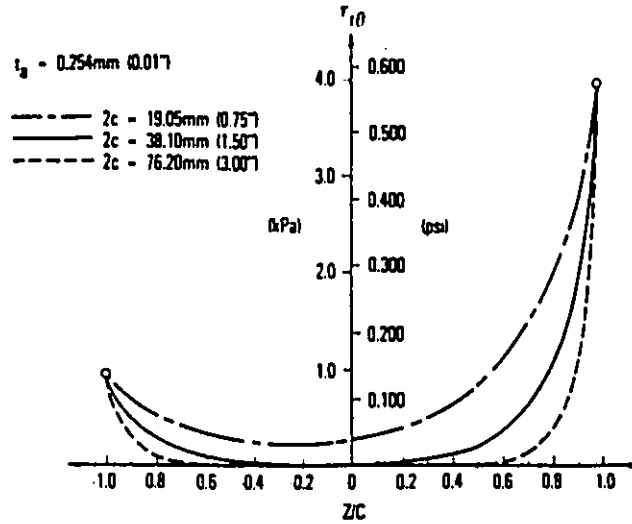


Figure 2.13 Variation of adhesive stress distribution with changing overlap length [41].

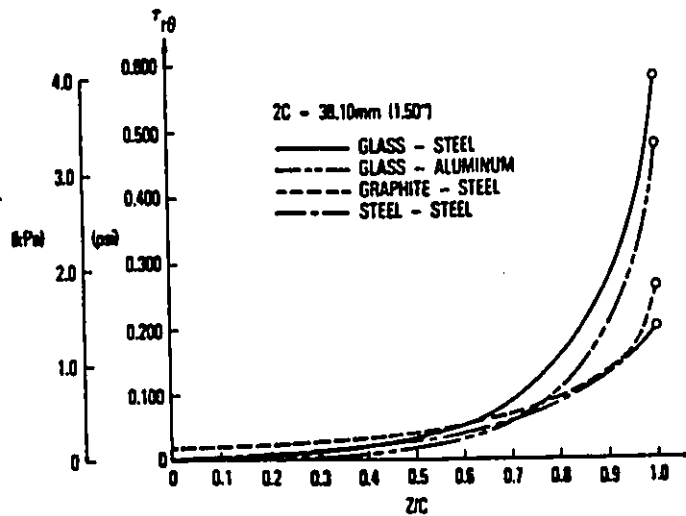


Figure 2.14 Effect of relative adherend stiffness on stress maximum in the adhesive layer [41].

The geometry of the metallic adherend was thought to be a possible area which might be used to reduce stresses in the adhesive layer. Hipol [41] first used various metal thicknesses and found that only in the case where the composite tube is considerably thicker than the metal would reducing the metal thickness be of benefit (Figure 2.15). It has already been established that a thin tube must be used for in-plane testing [20,22,23,24,25,27]; thus, this finding is of limited

value for this study. In addition, Hipol [41] studied the effect of tapering the end of the metal adherend closest to the tube gauge section. This proved that large angles were necessary to obtain only a marginal decrease in stress. Large angles would result in manufacturing difficulties and perhaps reduce the overall advantage in using the taper for stress reduction.

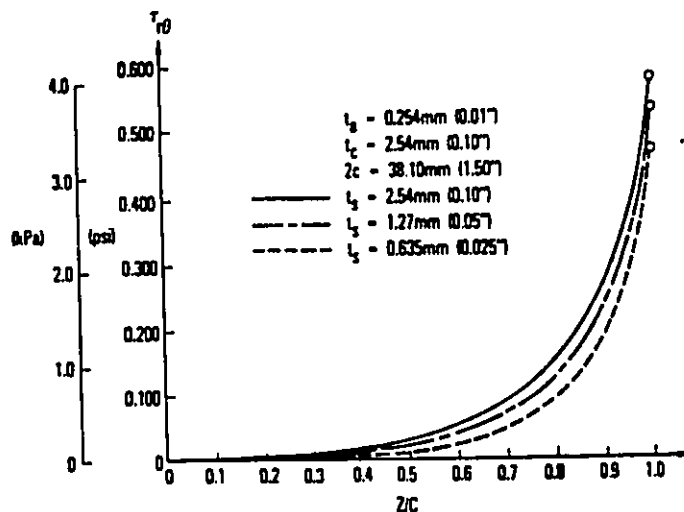


Figure 2.15 Effect of changing adherend thickness on stress level in the adhesive [41].

Mathews et. al. [43] compiled a short list of general design guidelines which he recommended to prevent premature joint failure and several are pertinent to in-plane shear testing of tubes. Ideally, the adhesive shear strength should be 50% greater than that of the adherends. Secondly, the design should consider a failure load 20% higher than that which is expected in order to account for incomplete wetting of the adherends.

Bonded tubular lap joints have been used to test torsion tube specimens with marginal success. Ellis and Hoskin developed a split cylindrical lap joint which used an internal plug [31]. A schematic of this design appears in Figure 2.16. Though this design was able to sustain high failure loads, many adhesive failures in the joint were also observed. Ellis and Hoskin [31] suggested that non-uniformity of the adhesive layer may have been responsible for causing this failure as there was poor control of both the inner and outer adhesive thickness. In addition to this, many of the effects already cited were not considered. It is possible that a more judicious

use of these design guidelines may have yielded consistent joint performance. Another similar design was used by Hahn and Erikson [28] which is shown in Figure 2.17. An internal plug is used for alignment while the tube and plug slide into a metal adherend sleeve. The holes shown are to transmit torque from the torsion device to the upper half of the end fitting and to help join the fitting and plug together. Adhesive is placed between the end fitting and the tube as well as between the plug and the fitting. It is not certain whether this fitting will sustain maximum failure loads since it was only used for measurements in the elastic range. The reasons for selecting this design were not given.

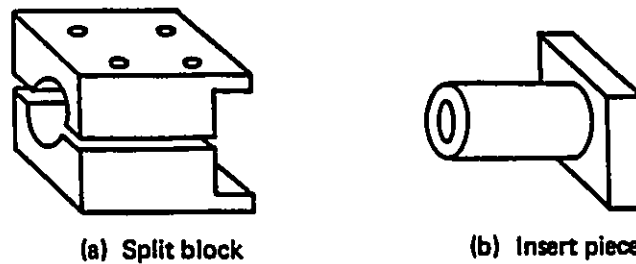


Figure 2.16 Split block/plug insert variation of the tubular lap joint [31].

Marlowe et. al. [32] used the internal lap joint shown in Figure 2.18. The fitting is in the form of an internal plug. Adhesive is applied to the inside of the tube and the close fitting plug is inserted. Design details and failure modes were not mentioned in the article. It is worth noting that a uniform glue thickness may be difficult to achieve with this approach.

Two additional lap joint examples have been found which have not been used for torsion specifically. Mazenko et. al. [38] developed a lap joint to be used on structural trusses for space applications (see Figure 2.19). Of particular interest is the use of a fibreglass collar which sits between the composite tube and the metallic fitting. This was used as a "buffer" between the fitting and the tube to take material thermal mismatches into consideration [38]. This concept could also be used to bridge a stiffness mismatch between the fitting and the composite tube which in turn would allow greater flexibility in the fitting material chosen for a given application.

Another interesting design developed by Mazenko et. al. [38] was the use of glass beads in the adhesive to insure a uniform bond thickness. This is not recommended due to the possibility that stresses might generally increase in the presence of these stress concentrations.

Higuera and Murat [42] used a simple external lap joint to verify the results of a finite element analysis which considered tensile loading and thermal cycling. The tube failed at 50% of its actual strength due to the high stress concentration at the free edge of the joint. This concentration was predicted by the finite element analysis. Thus it may be prudent to build up the tube thickness at the ends and then taper down to the thickness of the gauge section so that stress concentrations can be reduced near the ends. This is a precaution which should be observed for all fittings.

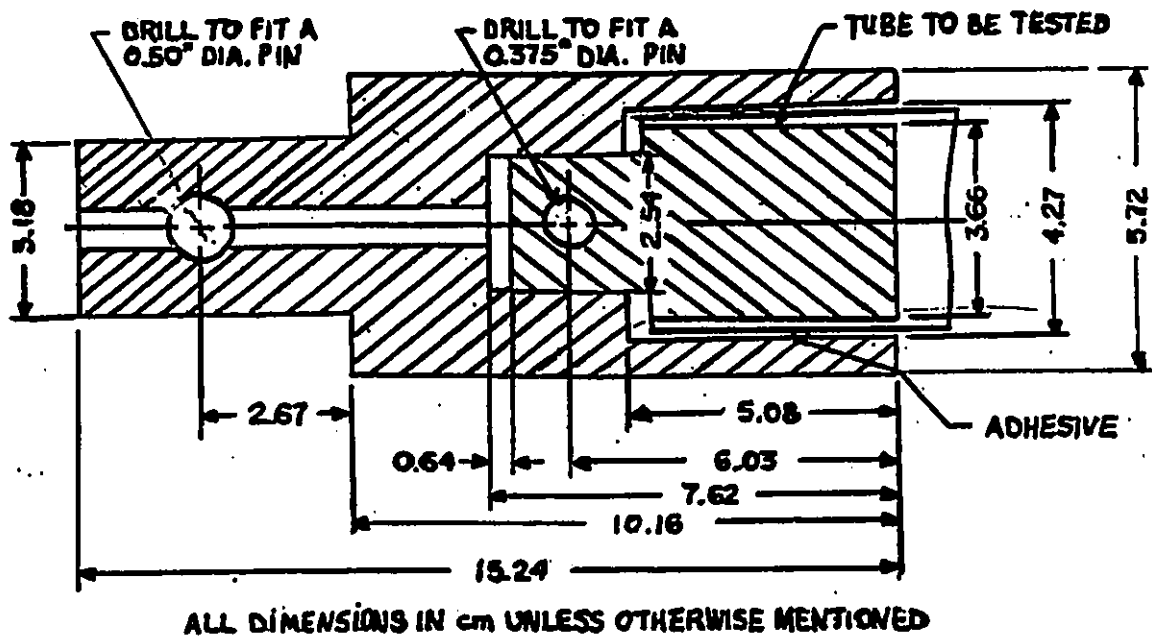


Figure 2.17 End fitting used by Hahn and Erikson [28].

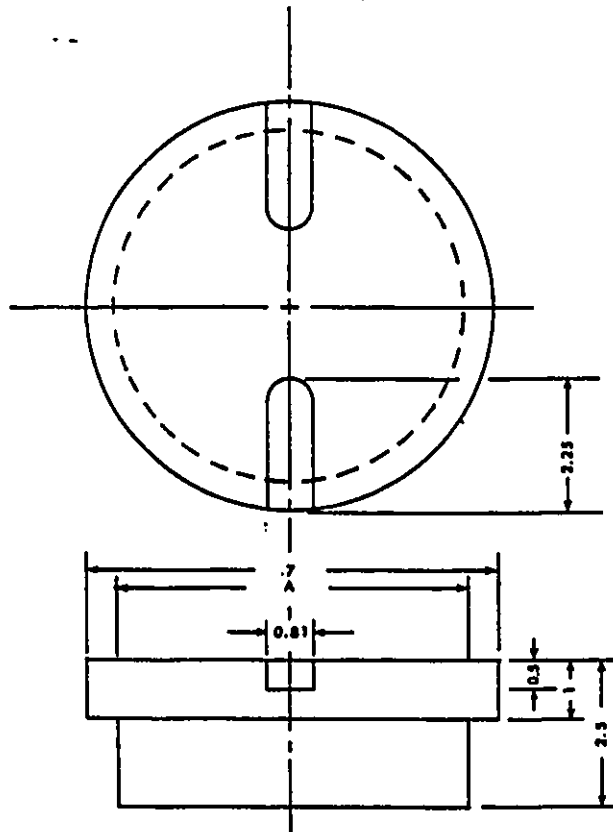


Figure 2.18 Internal lap joint end fitting [32].

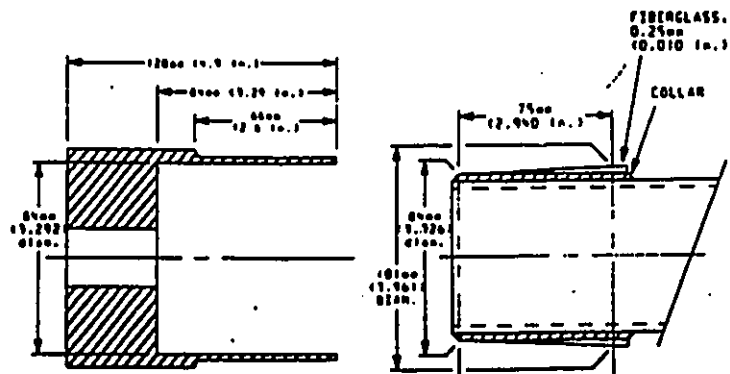


Figure 2.19 Tubular lap joint with fibre-glass collar [38].

2.5.1.2 Other Bonded Joint Designs

It is generally agreed that the bonded scarf joint is amongst the most efficient bonded joints that are available [43,44]. Small et. al. [44] stated that by using the scarf joint instead of the lap joint, a 150% increase in joint strength could be realized. A general schematic of a bonded scarf joint is presented in Figure 2.20. There are, however, two aspects which might limit the use of this type of joint for use as a specimen end fitting. The most obvious is the complexity of the joint cross-section. The manufacturing of this joint would require a more complex mandrel for filament winding of the composite tube or, alternatively, a difficult machining operation. Machining the tube end might produce residual stresses which may in turn contribute to failure in the tube at the joint tube interface. Small et. al. [44] have proposed that there is a correlation between flat composite scarf joints and tubular composite scarf joints. This approach was to determine a correction factor which would allow one to determine tubular joint strength from the testing of a flat specimen.

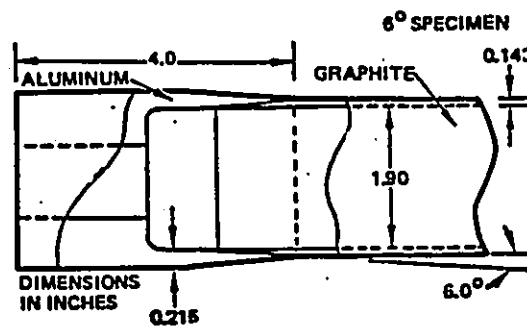


Figure 2.20 General schematic of a bonded tubular scarf joint [44].

The simplest joint found in the literature was that used by Swanson et. al. [45]. It consists of a flat plate bonded to the ends of the tubular specimen. A schematic of the plates bonded to the specimen is presented in Figure 2.21. This configuration conforms exactly to the conditions assumed in the shell analysis already discussed [20,22,23,24]. Furthermore, Swanson et. al. [45] reported that this is adequate for shear testing. It is indicated, however, that not all

specimen failures occurred in the gauge section suggesting perhaps that there may be some problems with the plate end fittings. Details of the failures were not discussed.

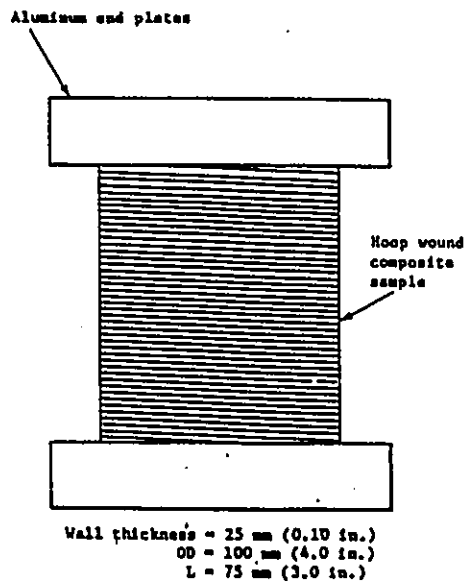


Figure 2.21 Flat plate joint used for end fitting [45].

2.5.2 Mechanical Tubular Joints

Fuchs et. al. present three examples of possible mechanical fittings - pinned or bolted joints, polyhedron joints and compression/friction joints (Figure 2.9) [17]. The pinned or bolted joint and the polyhedron joints have been used somewhat as tubular fittings for torsional testing but the compression/friction joint has received little, if any, use. One disadvantage of all the mechanical joints is that the effect they have on the stress distribution in the tube is not clearly defined. The polyhedron in particular could induce a greater through thickness stress gradient or an extended stress gradient with respect to the gauge length. This issue has not been studied for these types of complex geometries.

Though the compression/friction joint could be used for torsional loading [17], it is not in all likelihood desirable for tests reaching failure loads. This joint is produced by shrink fitting

a metallic ring component onto the tube [17]. Proper choice of the ring material, tube wall thickness and operating temperature is purported to provide a stable joint with a wide usable temperature range. A notable advantage of this technique is the possibility of simply reheating the ring to remove it from a tested specimen. One disadvantage of this approach is that joint strength tests have not been conducted so that applicability of compression/friction joints to failure testing can not be determined.

Bolted and pinned connections have received considerably more attention than either the polyhedron or compression/friction joints. Fuchs et. al. [17] reported that the pin joint of Figure 2.9 was used to test carbon fibre reinforced polymer tubes to failure. The tube wall within the joint was increased to limit the stresses in this region of the tube during testing. Ellis and Hoskin [31] also used pins with a bonded lap joint. Failure of the joint was observed in the region of the pins. Mazenko et. al. [38] found that by combining adhesive and pins (Figure 2.22) a joint superior to either bonded or bolted methods could be obtained; however, these tests were conducted in tension. Bauchau [46] used a bolted joint which included washers which conformed to the tube radius. The washers were used to distribute the bolt loads in an attempt to reduce stress in the region of the joint [46]. This joint was not tested to failure.

It is clear that bolted and pinned end fittings can be used to test in-plane shear specimens. Joint failure occurred when the joint was not properly designed for the stresses resulting from using pins as in the case of Ellis and Hoskin [31]. Additional thickness in the ends of the tube would increase the strength of the joint. The disadvantage of this type of joint is the additional care with which the specimens must be manufactured. Holes, if drilled in the tube, would require special fixtures and drilling techniques. Moreover, to attempt to manufacture the tubes with small diameter holes using the filament winding technique would be very difficult. This style of fitting does lend itself to re-use, however, which is an advantage.

A polyhedron joint has been developed by Spencer and Rumberger [47] and a similar joint is discussed by Fuchs et. al. [17]. An example of a polyhedron fitting appears in Figure 2.23. The polyhedron shape is incorporated into the tube by using a special mandrel during the

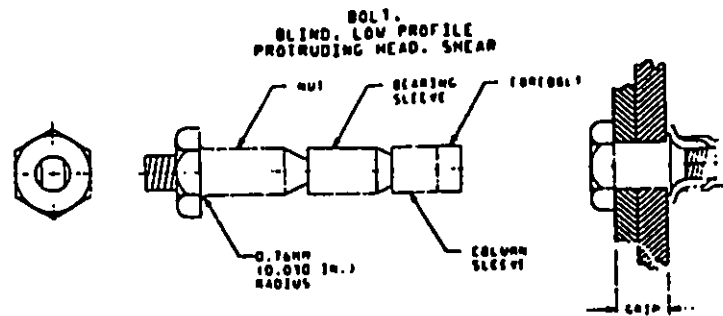


Figure 2.22 Fasteners used by Mazenko et. al. [38].

filament winding process [17,47]. The metallic fitting consists of either an internal or external polyhedron component which meshes with the tube [17,47]. An outer or inner collar is used to prevent bulging of the tube during loading [47]. The collar and polyhedron component are held in place by a lock nut and thrust washer arrangement [47]. A shaft is integral with the polyhedron component for load introduction [47].

The polyhedron design is very effective for torsion loading of tubular components [17] and is capable of achieving very high loads without failing [47]. Furthermore, the joint is removable [47] and the fitting can be used again. The major disadvantage of the fitting is the complexity of the tubular specimen and the number of parts required for the fitting. If a filament winding process is not used to manufacture the tube then this joint design is not very practical. Hand lay-up of tubes with complex geometries is difficult and does not lend itself to the accurate manufacture of the polyhedron ends.

2.5.3 Summary and Conclusions for Tubular Joints and End Fittings

The requirements of the end fitting to be used for in-plane shear testing have been stated but they are repeated here. Foremost is that the end fitting cannot cause premature failure in the tube through stresses developed by the joint and must not affect the stress state in the designated

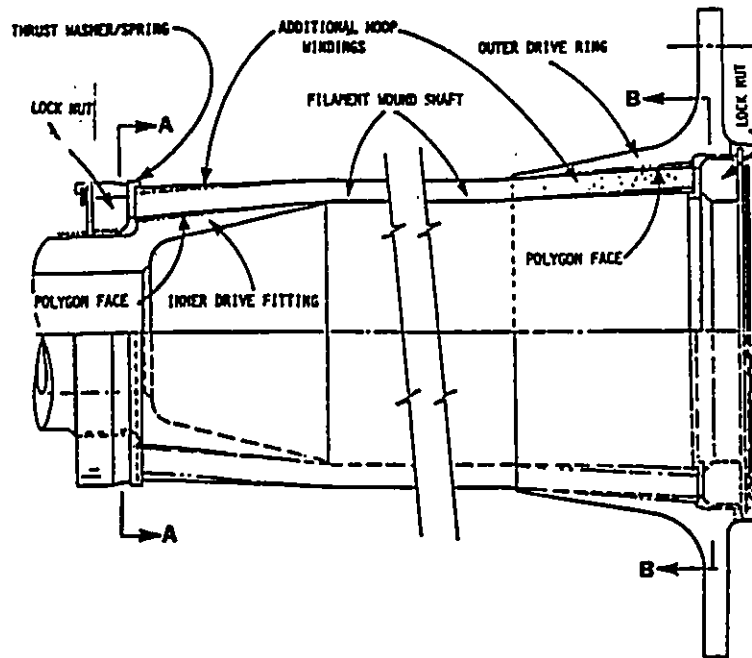


Figure 2.23 Side view of the Polyhedron torsional end fitting [47].

gauge region. Secondly, the joint must not itself, during testing, fail prior to the failure of the material. The end fitting should be reusable in the sense that after the tube has failed the removal and re-attachment to another tube should not be difficult. Finally, the design must be a simple one which facilitates easy attachment to the specimen and is easy to manufacture.

The only joint which satisfies to some degree each of the above stated requirements is the bonded tubular lap joint. The joint can be properly designed to withstand high torsional loads and at the same time not initiate failure due to end constraints. The tube can be removed from the fitting by incinerating the composite in a properly ventilated oven. In addition, the lap joint is very simple to produce and is easily attached to the tube.

Chapter 3.0

Selection of Material and Detailed Tube and End Fitting Designs

3.1 Introduction

The purpose of this section is to provide the basic information regarding the design of the tubular in-plane shear specimens. The composite material for the final in-plane shear testing is selected. A tube design is formulated based upon the information outlined in the review of Section 2.4. In addition, two end fitting designs are developed for use in transmitting torque load to the tubes based upon the information in Section 2.5.

3.2 Selection of Tube Material

Carbon fibre epoxy composites are receiving considerable attention in Canada for aerospace applications. The choice of materials was thus essentially between a well characterized material (such as AS4/3501-6) or a newer material (such as a toughened resin with IM6 carbon fibres). Since one of the major goals of the study was to develop an in-plane shear test specimen and test method, it was decided to select the AS4/3501-6 material for which a large amount of in-plane shear data already exists. In addition, the results would support existing IAR programs for the CF-18.

3.3 Specimen Design

3.3.1 In-plane Shear Tube Design

The selected in-plane shear specimen is a circumferentially wound thin-walled cylindrical tube. The dimensions of the tube (diameter, thickness and gauge length), as indicated in Section 2.3, can not be selected arbitrarily. The tube must be thin-walled in order to avoid stress gradients through the thickness yet sufficiently thick to prevent buckling of the tube prior to tube failure. The gauge portion of the tube must be long enough so that end effects due to the end fittings can be neglected yet short enough to again prevent buckling.

The approach used in designing the test tube followed directly from the review of the available tube technology (Section 2.4). Firstly, a desirable tube diameter was selected which satisfied manufacturing, material and handling concerns. It was felt that a tube diameter of approximately 50.8 mm (2") was reasonable for ease of fabrication and appropriate for easy handling during testing. This size was also very similar to many existing in-plane shear specimens as well as tubes found in structural torque applications. As detailed in Section 2.4, a wall thickness to diameter ratio of 0.05 is the maximum allowed so that stress gradients through the wall of the tube are sufficiently low. A wall thickness of 1.27 mm (0.050") was chosen to provide a more conservative ratio of 0.025. As also mentioned in Section 2.4, a tube length which is twice that of the tube diameter is required to assure that stress gradients due to end fittings do not affect deformations at the tube centre. In order to have a central region approximately 101.6 mm (4") long that was not affected by the end fittings, a final length between the end fittings of 203.2 mm (8") was selected.

It was necessary to then verify that this selection of dimensions would not result in buckling prior to material failure. Hayashi's [37] closed form equations (equations 2.11 to 2.14) were used to plot the buckling shear stress as a function of the length between the end fittings and tube thickness for a tube fabricated from AS4/3501-6 composite material (Figure 3.1). The figure clearly indicates the major influence of the gauge length. In order to determine an

accurate value of the buckling shear stress, a plot of buckling shear stress versus gauge length for the selected tube thickness of 1.27 mm was prepared (Figure 3.2). A typical value for ultimate in-plane shear strength of this material is 60-80 MPa. The critical buckling stress for a thickness of 1.27 mm and a gauge length of 203.2 mm is approximately 350 MPa (Figure 3.2). Since the stress required to initiate buckling is greater than the material shear strength, the material will fail before the buckling shear stress is reached.

In order to ensure failure in the gauge section, the thickness of the tube under the end fitting was doubled and then tapered over an axial length of 12.7 mm (0.5") to the required 1.27 mm in the gauge section of the tube. This corresponds to the ASTM recommended taper of 5° for end tabs on flat bar specimens. The taper serves to reduce the severity of the stress gradients in the gauge section due to the end fittings during loading. A cross-sectional view of the final composite tube design appears in Figure 3.3.

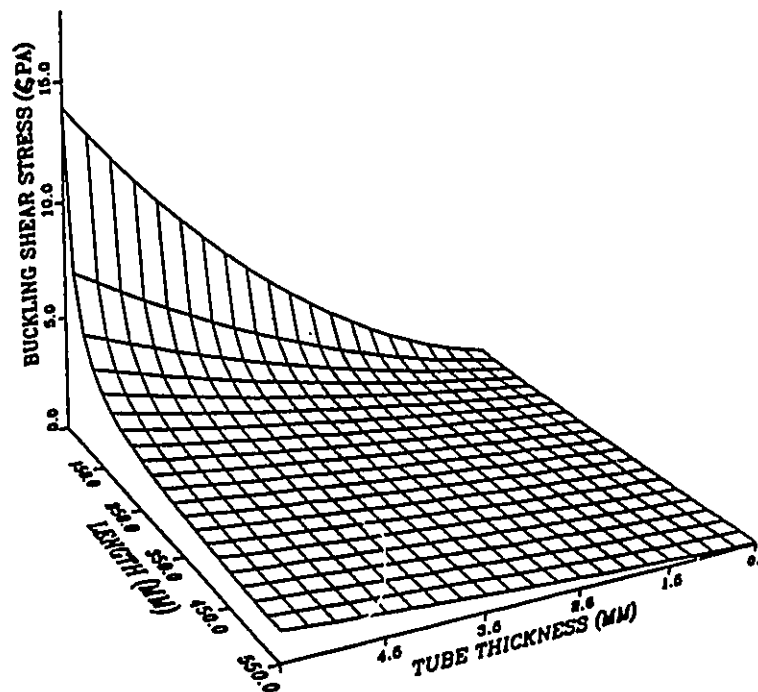


Figure 3.1 Buckling shear stress as function of length and tube thickness (according to Hayashi).

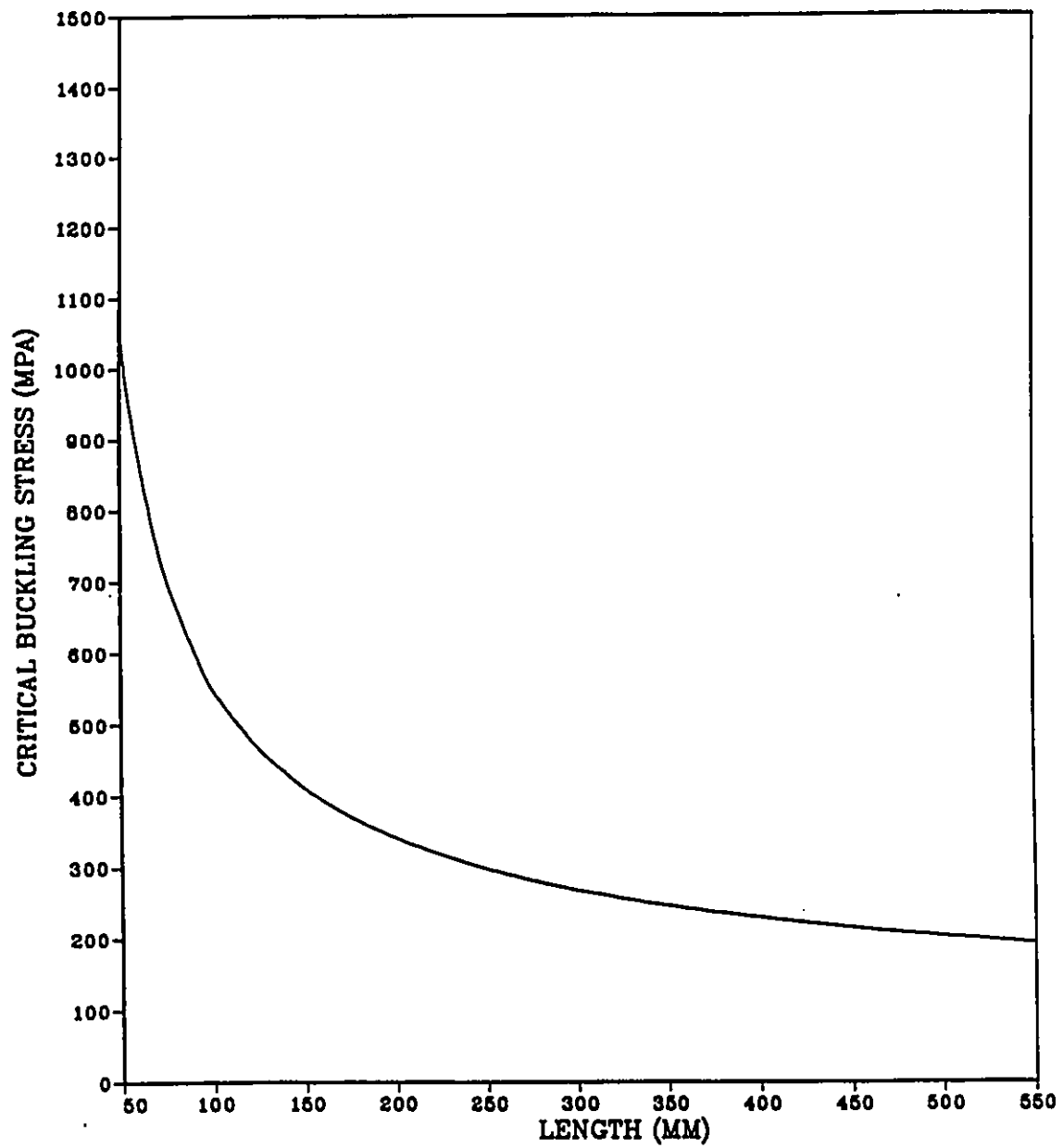
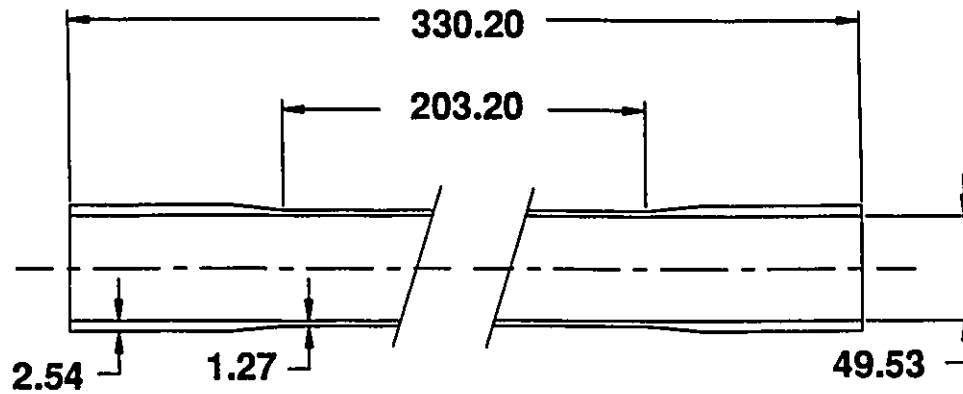


Figure 3.2 Buckling shear stress as a function of length for a thickness of 1.27 mm (according to Hayashi).



NOTE: ALL DIMENSIONS IN MILLIMETERS

Figure 3.3 Cross-section of final composite tube design.

3.3.2 Adhesive Bond Length

As concluded in Section 2.5, bonded end fittings will be used. A minimum bond length of 5 mm (0.2") was determined by equating the force necessary to cause shear failure of the bond area to the shear force necessary to cause material failure, i.e.

$$2\pi r l \tau_a = 2\pi r t \tau_{12} \quad (3.1)$$

where l is the bond length, τ_a is the adhesive shear strength (20 MPa), τ_{12} is the approximate in-plane shear strength of the composite (80 MPa) and t is the adhesive thickness. In order to account for non-uniform adhesive strength and include a safety factor for other parameters, a minimum bond length of 50.8 mm (2") was selected. Thus, the overall length of the composite tube was 325 mm (13").

3.3.3 End Fitting Design

In order to transmit torque to the specimen two bonded end fitting designs were developed. The first of these consisted of an externally bonded lap joint while the second was an internally tapered lap joint.

3.3.3.1 External Shell End Fitting

The externally bonded end fitting which is shown in Figure 3.4 is modelled after the design of Ellis and Hoskin [31]. The various requirements for the end fitting were discussed in Section 2.4. One of the major problems that seemed to cause premature failure of the Ellis and Hoskin design, and other bonded designs, was that of a non-uniform glue line thickness. In the first design an inner plug that had a transition fit with the inner diameter of the composite tube was used. The outer diameter of the composite tube at the built up ends are ground to the final dimension. The split outer shell, when fitted over both the inner plug and the specimen tube, resulted in a clearance of 1.27 mm (.05") between the inner diameter of the split outer shell and the outer diameter of the tube. The resulting adhesive thickness of 1.27 mm (0.05") was selected as the thinnest practical layer. The selected material for the end fittings is mild steel. This is a preferred choice to aluminum considering the stress level in the adhesive (Figure 2.16). Furthermore, the selection of steel permits the end fitting to be re-used after burn off of the composite.

The main advantages of the external shell end fitting is that it is potentially reusable which offers substantial cost savings as fewer end fittings would be needed to conduct the testing. There are, however, several disadvantages to it as well. In particular, it is heavy because it is made of steel. Since it is an external lap joint, an internal alignment plug is required. The many components required makes bonding the end fitting/composite joint cumbersome. Design drawings of the external shell end fitting appear in Appendix A.

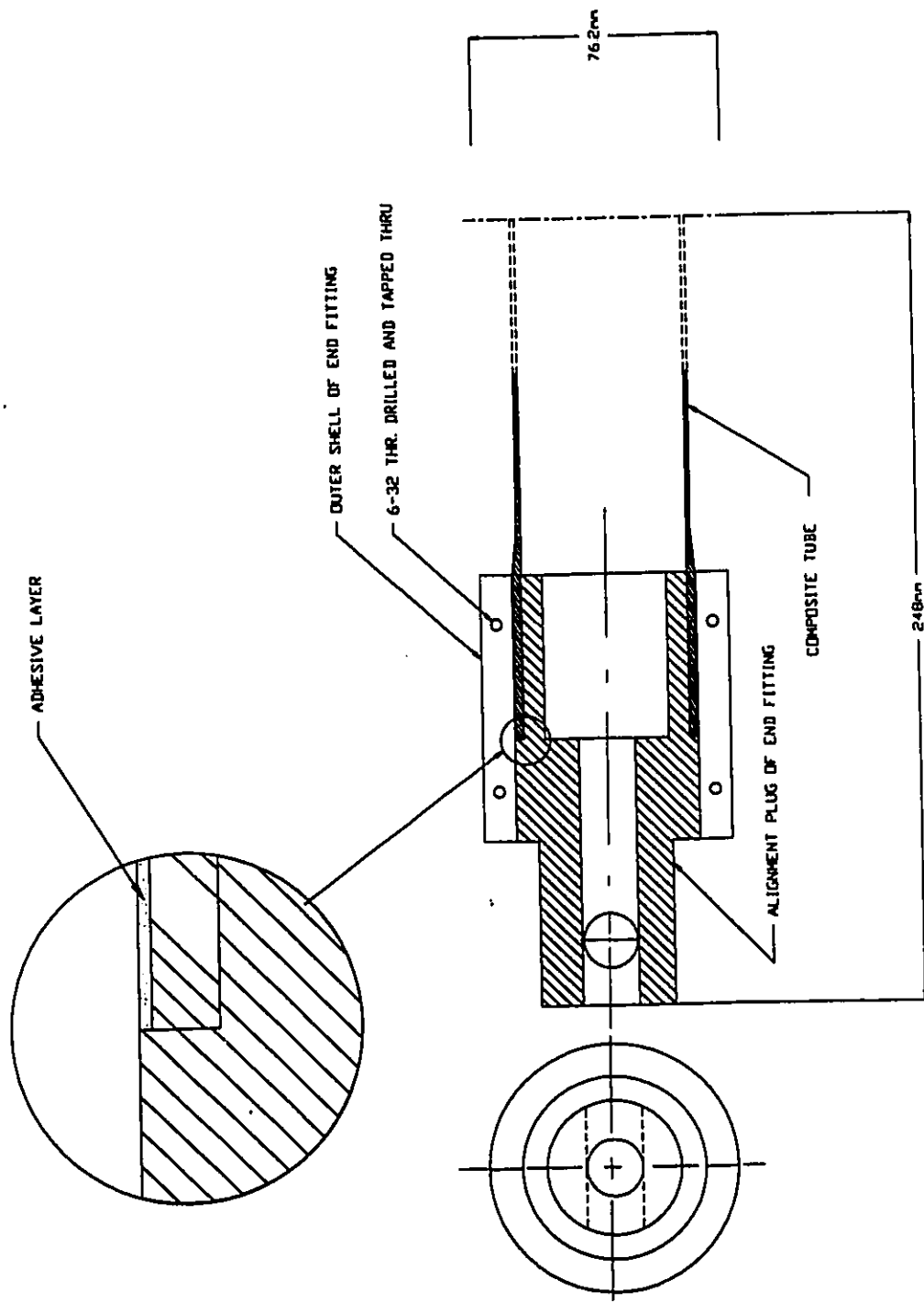


Figure 3.4 Schematic of external end fitting composite joint.

3.3.3.2 Tapered Internal End Fitting

The tapered internal end fitting (Figure 3.5) addresses the two biggest disadvantages of the external shell end fitting. Firstly, it is an internal design and made from aluminum; both changes will result in a substantially lighter component. Secondly, the overall design is much simpler as it is only made from one component. The constant adhesive thickness around the perimeter which is believed to be critical for high load carrying capability is assured by an internal shoulder which is mated to the inside of the composite tube with a light transition fit. The use of a shoulder, however, requires machining of the internal diameter of the tube for an axial distance of 12.5 mm at each end. In addition, it is proposed to remove the tube from the end fitting by machining in an attempt to make the end fitting reusable.

A second feature was inspired by Hipol's work [41] (Figure 2.12) which suggests that stress levels can be reduced if the adhesive layer is increased. The highest stresses in the end fitting/composite joint are at the end of the adhesive near the gauge section where there are a geometric and material discontinuities. By tapering the internal end fitting (Figure 3.5), a greater adhesive thickness would exist at the point where stresses are highest. In order to further reduce the stress levels the end fitting bonding surface was lengthened by 50 mm over that of the outer shell end fitting. The taper was selected such that the adhesive thickness varied from 0 mm at the tube end to 1.3 mm at the edge of the gauge section. This ensured that the same amount of adhesive was used for both end fittings. The length of this fitting is 38.1 mm longer than the external shell end fitting, therefore the specimen length was extended for use with the tapered end fitting to 406.4 mm (16"). A design drawing of the tapered end fitting appears in Appendix A.

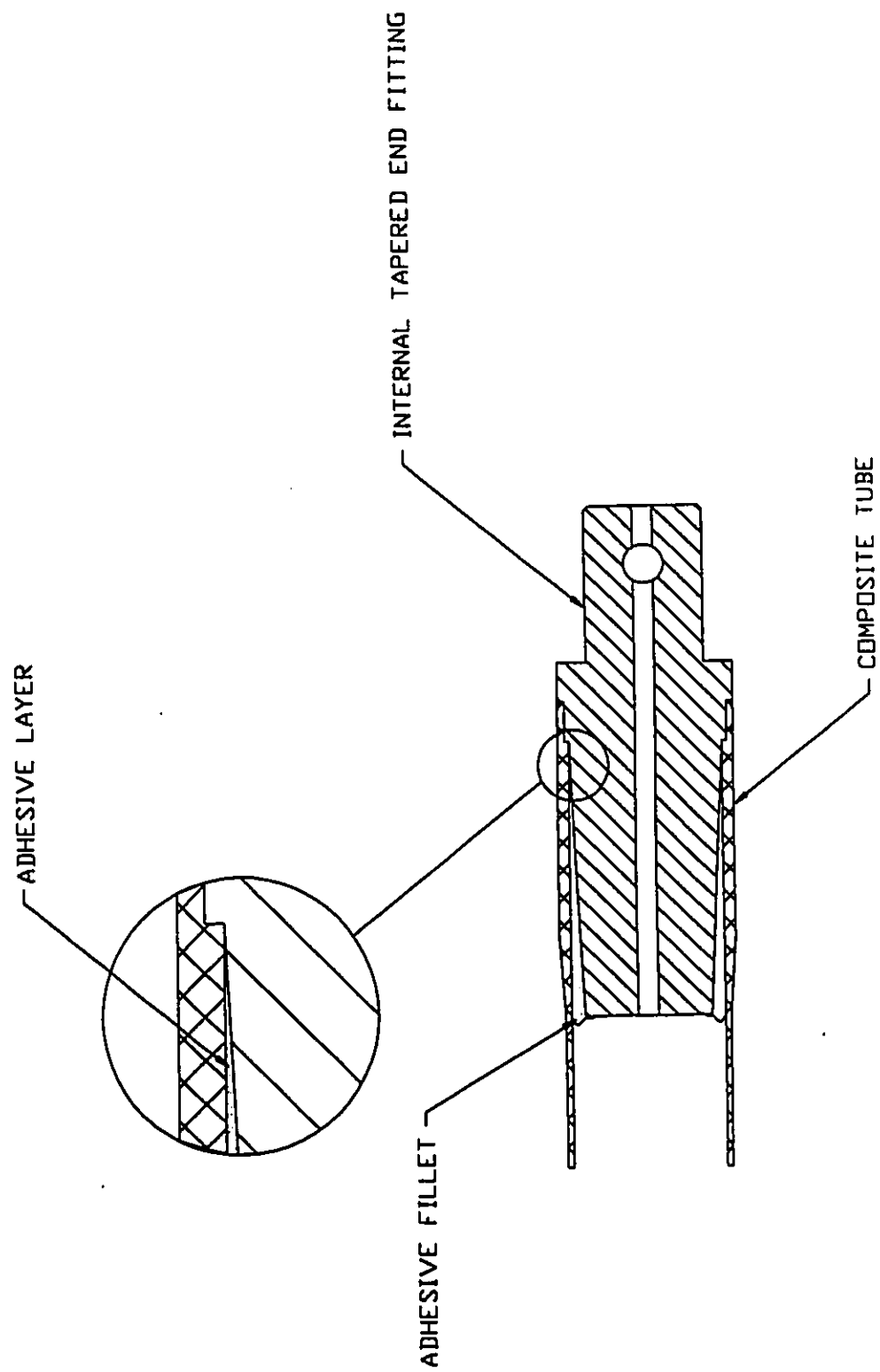


Figure 3.5 Schematic of tapered internal end fitting/composite joint.

Chapter 4.0

Preliminary Testing and Results

4.1 Introduction

Before fabricating the tubular in-plane shear specimens to be used in comparison of the test methods, several design and manufacturing assumptions had to be confirmed experimentally. In particular, the specimen was designed so that the central 100 mm of the gauge section would be free of effects due to the end fittings. The external shell end fitting design was evaluated on the basis of its ability to sustain loading until the specimen fails and its reusability. An effort was made to determine whether there were significant residual manufacturing stresses within the composite tube. Finally, the manufacturing of some preliminary composite tubes also provided an opportunity to optimize the basic filament winding procedure to be used for the in-plane shear study.

4.2 Material Selection

Epoxy matrix FRP's, which are the focus of this study, are manufactured in a two step process. The laminate is produced by stacking the "raw" laminae on top of one another one at a time into a non-structural pre-form (usually called the lay-up process). The pre-form is then cured at high temperature, and sometimes at high pressure. There are two forms in which the

"raw" material is available. In the first of these, resin is added to the dry fibre as the laminae are being laid-up into the pre-cured laminate. The component is then generally cured in an oven in a standard air atmosphere. The second form, called pre-preg, is purchased with the resin and fibre already combined. Moreover, the resin is in a partially cured state called B-staged. The advantages of this second material form are that the resin/fibre content are strictly controlled, better material properties result and the process is generally easier and cleaner. The notable disadvantages are that the laminate must be surrounded by a vacuum (in the form of a vacuum bag) so that air and volatiles can be drawn from the laminate and that an autoclave is required to cure the laminate under precise pressure and temperature control (Figure 4.1).

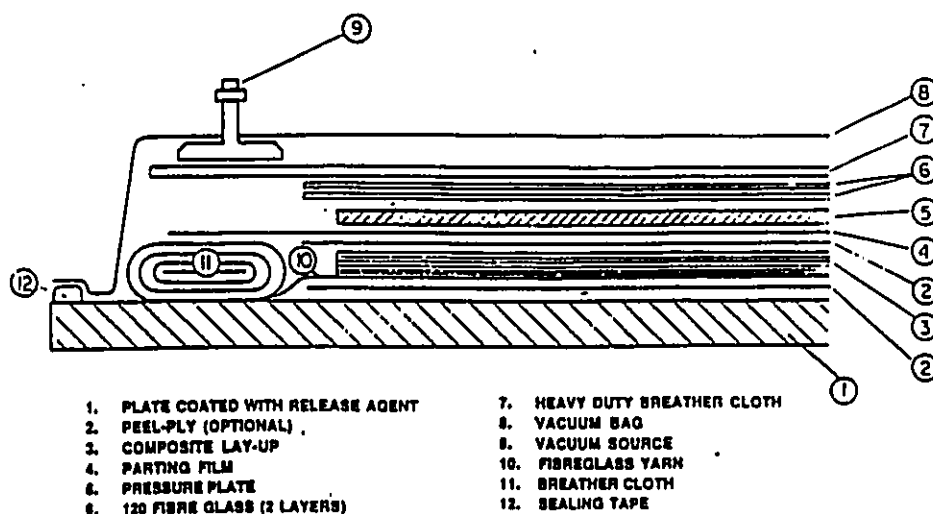


Figure 4.1 Configuration of vacuum bagged composite panel in the autoclave.

At the time of preliminary testing, the AS4/3501-6 pre-preg material was not available. It was therefore decided to develop the basic fabrication procedure for the composite tubes using both alternate pre-preg tow and wet winding of dry fibre and epoxy resin that was available at the University of Ottawa Composite Materials Laboratory. Even though the useful life of the pre-preg (donated by Boeing Canada to the University of Ottawa) had been exceeded, it was felt that the initial procedures could be developed using this material. The material used was Fiberite 6K FX30E86FP (T300 carbon fibres in a modified bismaleimide resin - Lot M88-0011). The second material (for wet winding) was comprised of E/XA-S Grafil carbon fibres and an epoxy

resin matrix consisting of Shell Epon 825 epoxy resin with Pacific Anchor Ancamine 1482 hardener at 19 phr.

4.3 Fabrication of Tubes

While very little detailed information describing tube manufacturing processes is available, the two techniques to produce rotationally symmetric pre-cured composite components are well known, namely, hand lay-up and filament winding. Previous experience at I.A.R. for hand lay-up of small diameter composite tubes had shown that it was difficult to produce a defect free component. Thus, filament winding was selected as the manufacturing method. The basic technique of filament winding consists of passing bundles of filaments (called a tow) of dry fibre or pre-preg from a spool through a number of rollers, around a tensioning device and through a pay-out eye that deposits the fibres on a rotating mandrel (Figure 4.2). In the case of dry fibre the tow is passed through a resin bath. Prior to this study the computer controlled filament winder at the University of Ottawa could only be used for wet winding (Figure 4.3). As part of this study it has been extensively modified to operate with pre-preg tow (Figure 4.4).

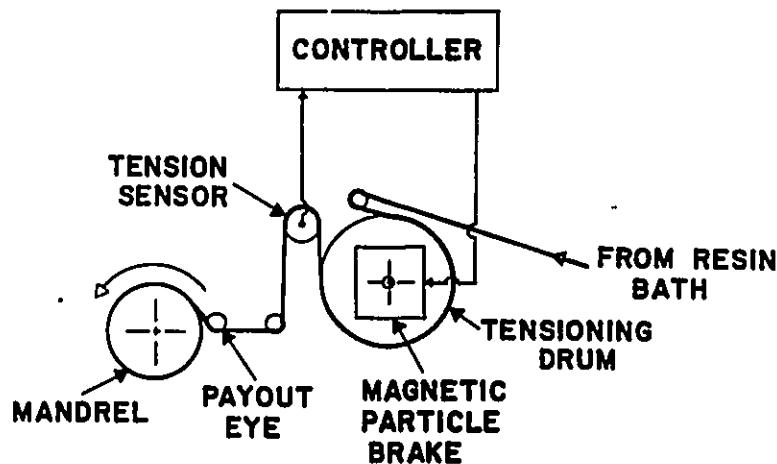


Figure 4.2 Schematic of the filament winding process.

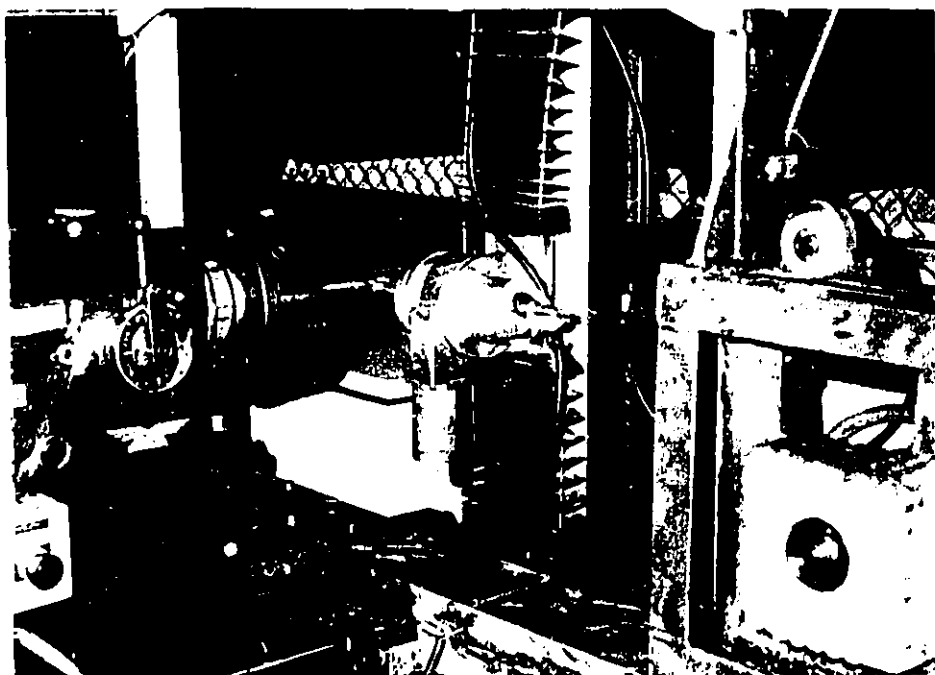


Figure 4.3 Filament winder set up for wet winding.

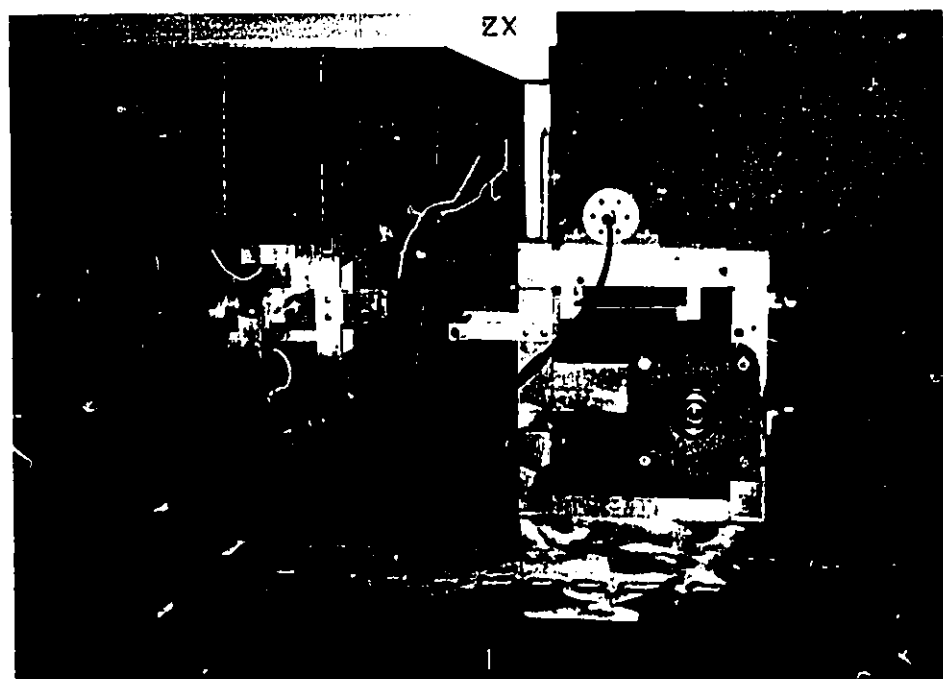


Figure 4.4 Filament winder set up to wind pre-preg tow.

Two general types of test specimens were fabricated; one was used as an in-plane shear specimen to verify the buckling calculations and strain gradient predictions, the other to test the ultimate capacity of the joints.

Two pre-preg composite tubes were fabricated; the first was for evaluation of the manufacturing technique and to confirm that the tube was of adequate quality for testing (low void content, uniform thickness) and the second was used as one of the general types of test specimens i.e., an in-plane shear test specimen. Since the material system was not AS4/3501-6, new buckling shear stress diagrams (Figures 4.5 and 4.6) were prepared. As before there is no concern of premature buckling failure with this material.

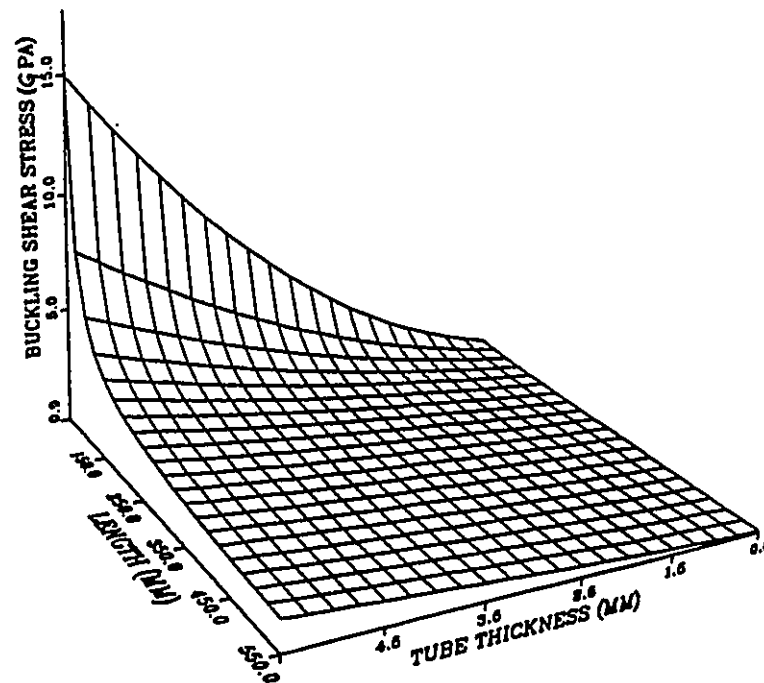


Figure 4.5 Buckling shear stress as a function of length and tube thickness for Fiberite FX30E86FP.

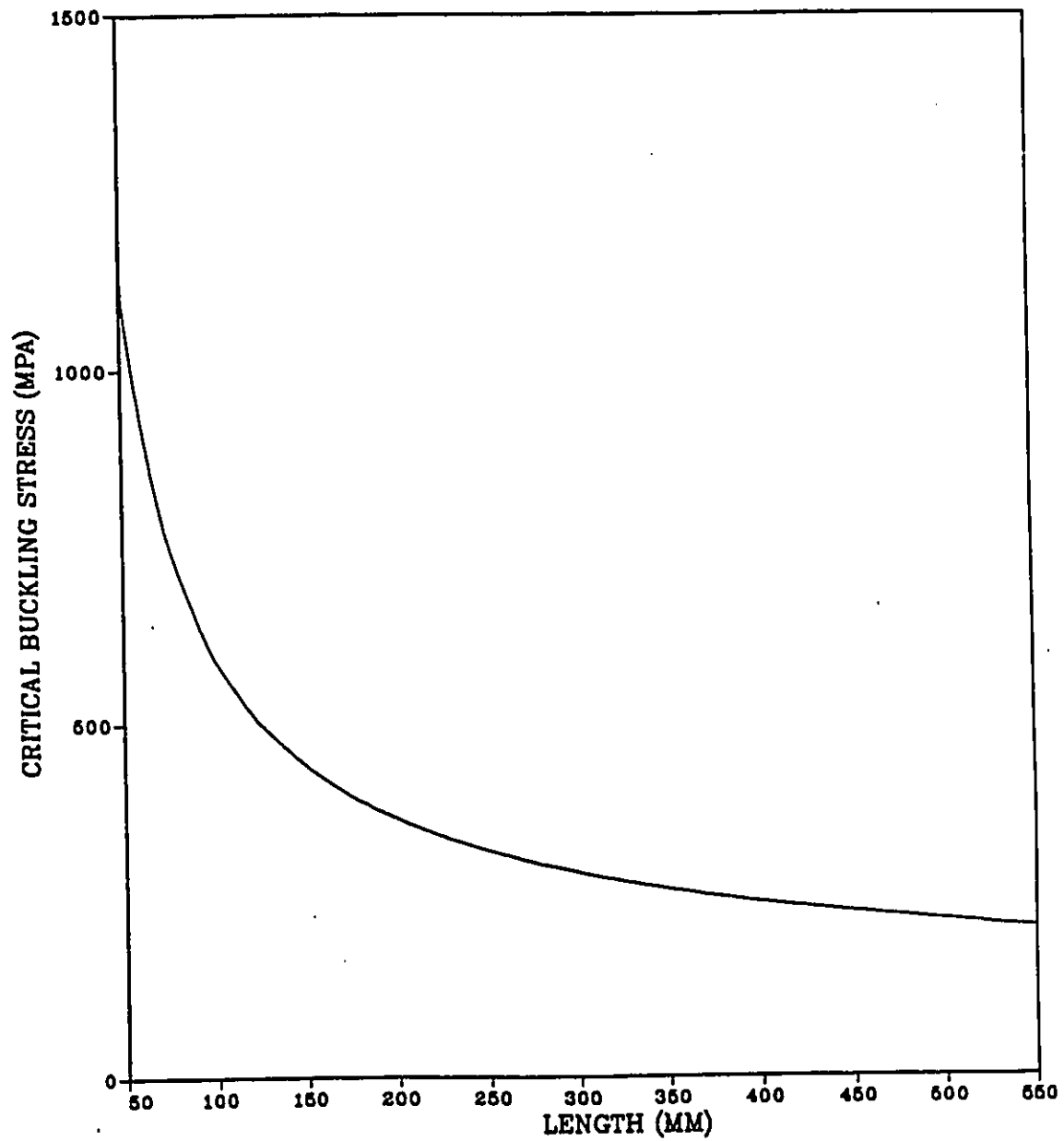


Figure 4.6 Buckling shear stress as a function of length for Fiberite FX30E86FP and a thickness of 1.27 mm.

Two tubes were made to test the ultimate strength of the two end fitting designs. For this reason thicker tubes were fabricated. Due to the nature of the external shell design the maximum possible thickness between the inner steel plug and the inner surface of the two halves of the outer shell was 2.0 mm, an increase of 60% over the in-plane shear specimen thickness. For this end fitting the tube gauge length remained at 200 mm resulting in an over all length of 300 mm. The 2.0 mm thickness was constant along the tube length. A second 2.0 mm constant thickness composite tube was fabricated. For the second end fitting design the length of the tube was increased from 300 to 400 mm (16") to accommodate the longer bonded length of the tapered internal end fitting. Since, there was not sufficient Fiberite pre-preg for the manufacture of the joint strength tubes, they were fabricated using the wet winding materials described in Section 4.2.

4.3.1 Filament Winding

The material selected for the mandrel was mild steel (Figure 4.7). It was checked that the differential thermal deformation between the steel and a circumferentially wound composite tube due to cool down to room temperature after curing was sufficient to permit extraction of the tube from the mandrel.

All the tubes were manufactured using the computer controlled filament winder located in the Composite Materials Laboratory at the University of Ottawa. For the pre-preg winding, individual layers of 0.0125 mm thick tow were deposited one layer at a time in the forward direction only at a wind angle of approximately 89.5°. The tapered section of the in-plane shear specimen between the gauge section and the built up ends was stepped in eight equal increments. Since the wet wound tubes were not used as in-plane shear specimens, the tubes were wound bi-directionally in order to speed up the filament winding process. The wider 12K E/XA-S tows resulted in winding angles of $\pm 88^\circ$.

4.3.2 Curing of Tubes

After filament winding of the pre-preg tube was complete, it was then covered with non-stick tape which was in turn covered with a gum rubber layer. The tube was next bagged for curing in the autoclave. The rubber layer shrank around the tube during the cure cycle. The non-stick tape ensured that the rubber could be removed from the composite tube after the cure process. The autoclave at IAR/NRC was used to cure the composite tube according to the following schedule: room temperature to 188°C at 2.5°C/minute, then four hours at 188°C. The wet wound tubes do not require vacuum bagging or high pressure for curing and so these tubes were cured at 110°C for two hours in the oven in the Composite Materials Laboratory at the University of Ottawa.

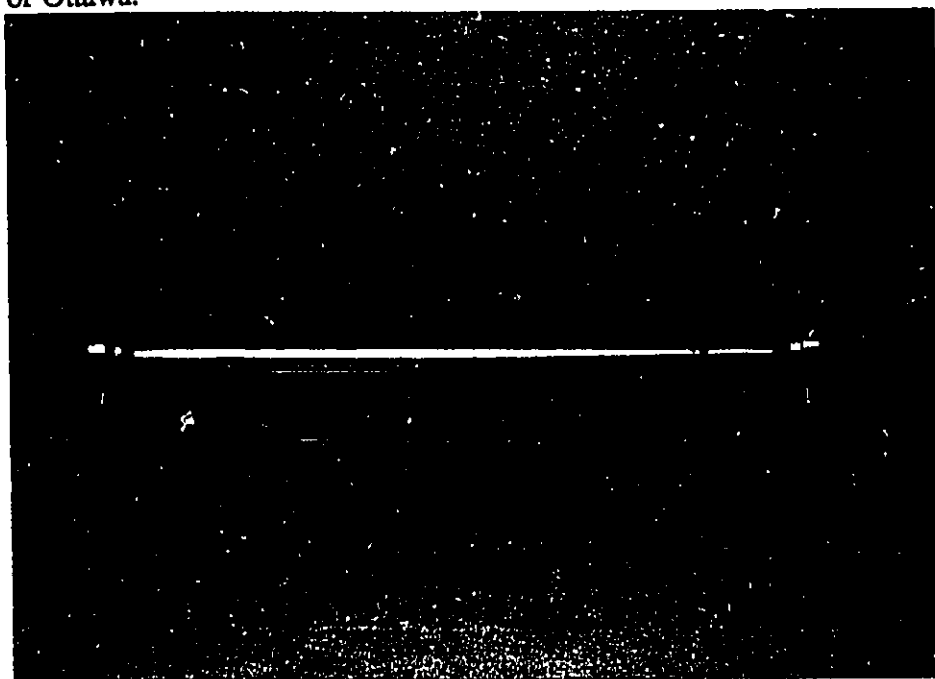


Figure 4.7 Mandrel used to wind preliminary composite tubes.

The curing technique for the pre-preg used in this preliminary stage was an existing in-house technique at IAR. The resulting tubes had small wrinkles on the surface of the gauge section due to irregular shrinkage of the gum rubber during curing. Further work was required to produce flaw free tubes (Section 6.2). These tubes were, however acceptable for preliminary testing.

4.4 Testing and Results

4.4.1 In-plane Shear Testing

The in-plane shear testing to evaluate the effect of end fittings on the strain state in the gauge section was undertaken first. The steel external shell end fitting which was designed to be easily reused was selected. The second pre-preg tube was then ground at the ends (Figure 4.8) in an attempt to ensure a uniform adhesive thickness. The inner end fittings were then inserted into the composite tube (Figure 4.9). The outer metallic end fittings were subsequently bonded to the outer surface of the composite tube with Hysol EA-9309.2 adhesive. Finally a number of 0°/45°/90° stacked rosette strain gauges (Showa N32-FA-5-120-11) were mounted on the tube as shown in Figure 4.10. A completed specimen is shown in Figure 4.11.

It was expected at the beginning of this study that an ATM (Adelaide Testing Machines, Toronto, Ontario) servo-hydraulic triaxial test machine would be available at IAR for testing all of the tubular specimens. Since the ATM testing machine was not available, a temporary test apparatus was used (Figure 4.12). The lower end of the test specimen was mounted in a three jaw lathe chuck which was secured to the bed of a large radial arm drill press located in the Mechanical Engineering Machine Shop at the University of Ottawa. The upper end of the specimen was located in a Jacob's chuck of the radial arm drill press which had been placed directly above the centre of the lathe chuck. A torque arm (Figure 4.13) was attached to the upper end of the specimen. A threaded rod was attached in series with a strain-gauged force transducer at the end of the torque arm. The other end of the threaded rod was attached to a fixed stand. By tightening the nut on the threaded rod on the fixed stand, a controlled amount of torque could be applied. The strain readings were monitored using two strain-gauge switch and balance units. Due to the limit of 18 channels, the gauges monitored are as indicated in Figure 4.10.

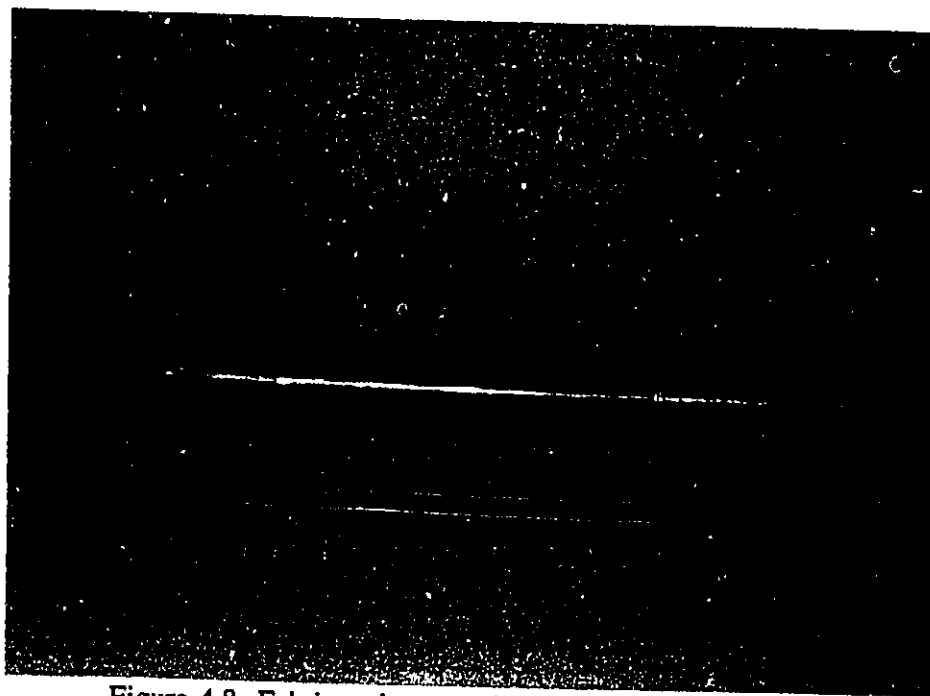
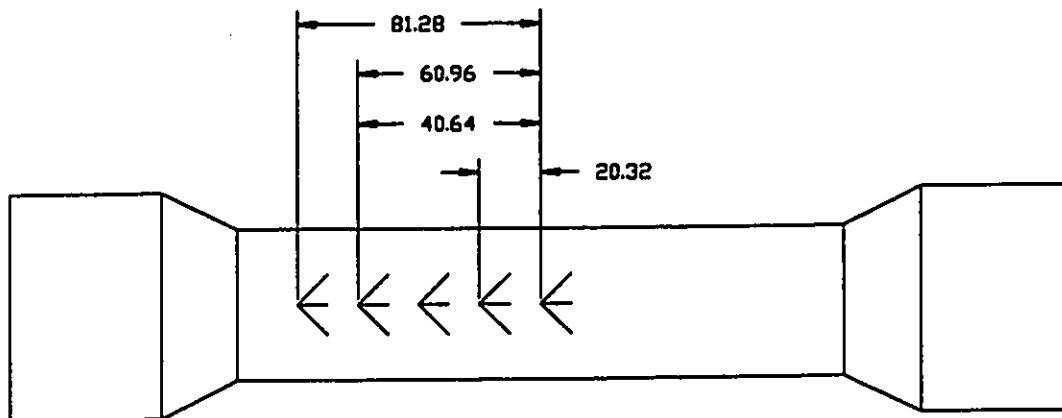


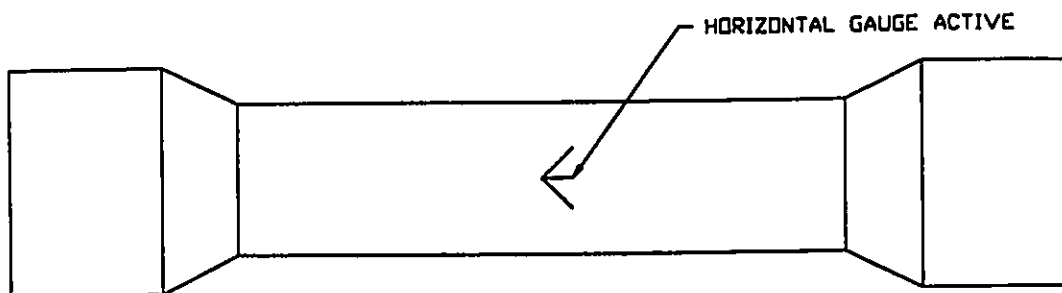
Figure 4.8 Fabricated composite tube with ends ground.



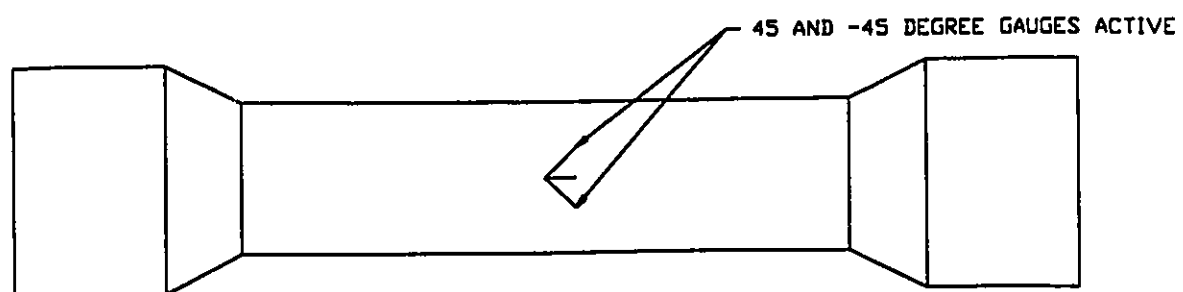
Figure 4.9 Insertion of inner plug of end fitting in composite tube.



MAIN STRAIN GAUGE ARRAY - ALL GAUGE ARMS ARE ACTIVE

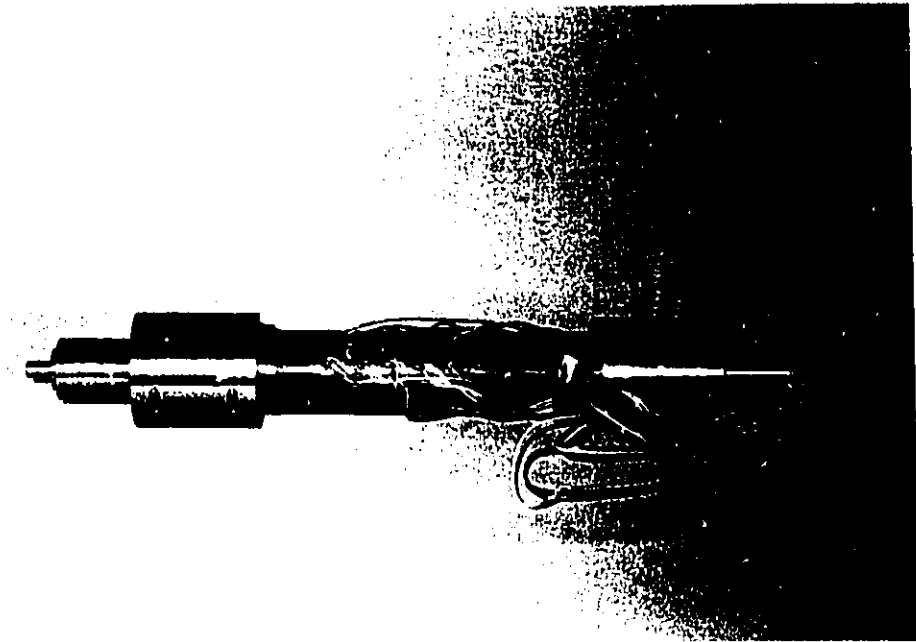


GAUGE LOCATED AT TUBE CENTER, ROTATED 90 DEGREES FROM MAIN ARRAY

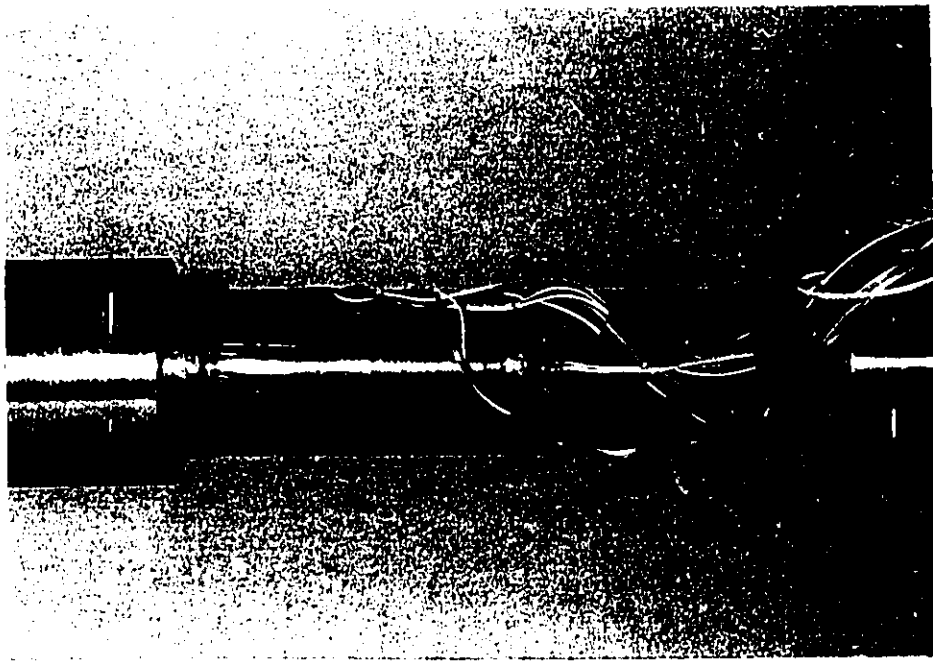


GAUGE LOCATED AT TUBE CENTER, ROTATED 180 DEGREES FROM MAIN ARRAY

Figure 4.10 Location of rosette strain gauges.



(a)



(b)

Figure 4.11 Completed tubular in-plane shear specimen-(a) top view; (b) side view.

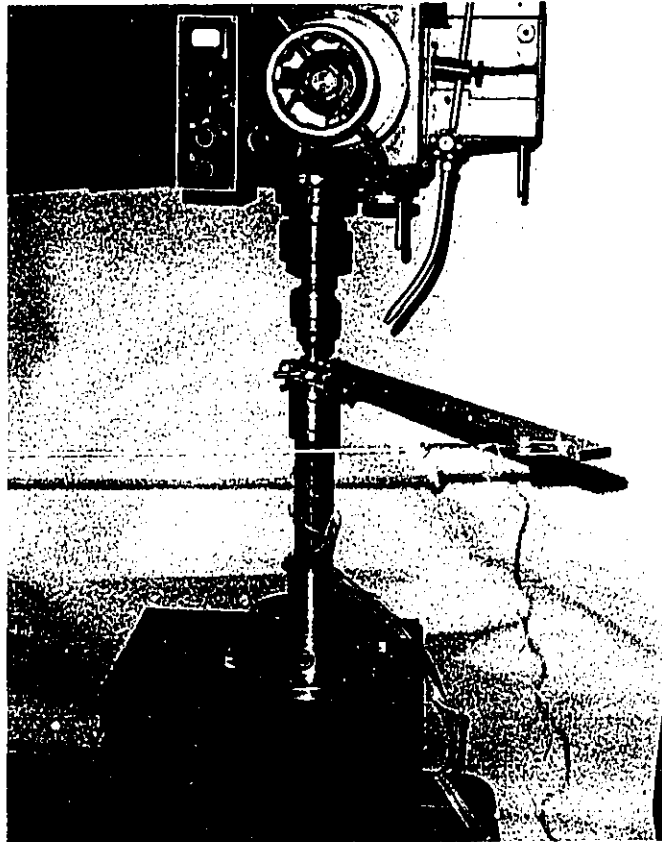


Figure 4.12 Preliminary torsion test apparatus.

The shear state within a tube is considered pure shear if the magnitudes of the strain gauges at $\pm 45^\circ$ to the tube axis are identical. Table 4.1 summarizes the strain readings for the gauges at $\pm 45^\circ$ to the tube axis for two loads at various locations along the tube. The agreement between the gauges at each location indicated a high degree of pure shear in the tube. The results of Table 4.1 generally indicate good agreement between all shear readings at each torque level. For the applied torque of 201 Nm, there is excellent agreement (maximum variation from mean is less than 3%) for all shear values within ± 40.6 mm (± 1.6 ") of the mid-point of the specimen. These results are not surprising as the specimen was designed to have an approximately 100 mm (4") long central region which would not be affected by the end fittings. In addition, the measured axial strains (Table 4.2) are insignificant as compared to the shear strains in Table 4.1. The experimentally measured value for G_{12} is 6.6 GPa which is within the normal range reported for this material.

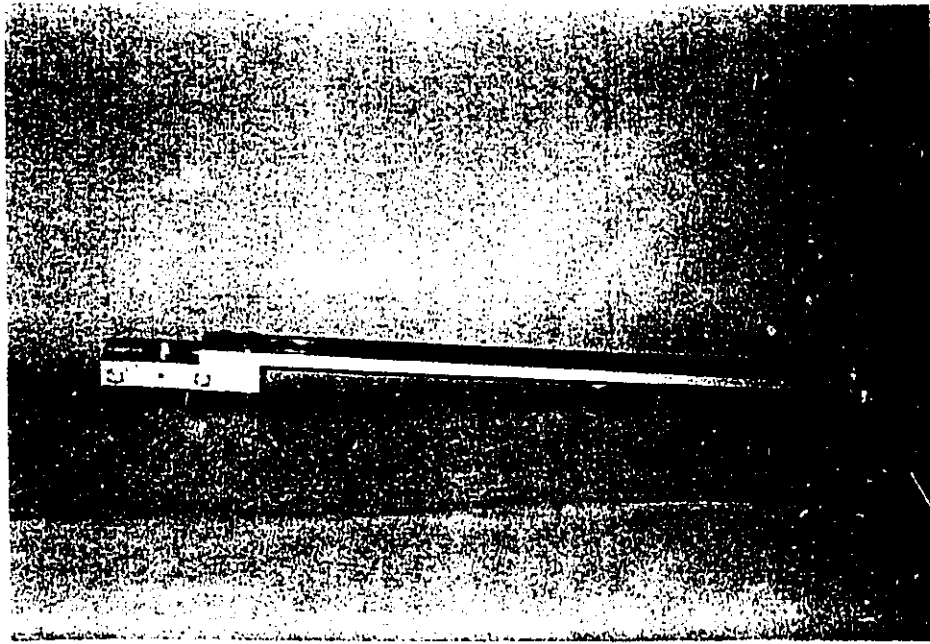


Figure 4.13 Instrumented torque arm.

Table 4.1 Strain readings at $\pm 45^\circ$ to the tube axis

Torque (Nm)	Gauge Orientation	Distance From Tube Centre (mm)					
		81.3	61.0	40.6	20.3	0	0'
119	45°	1978	2026	1910	1882	1780	1924
	-45°	-1980	-2024	-1990	-1978	-1898	-1926
201	45°	3232	3310	3090	3040	3030	3180
	-45°	-3100	-3200	-3145	-3135	-3110	-3100

1 Gauge is rotated 180° around the tube as indicated in Figure 4.10

Table 4.2 Strains parallel to the tube axis

Torque (Nm)	Distance From Tube Centre (mm)					
	81.3	61.0	40.6	20.3	0	0 ¹
119	40	42	75	2	40	40
201	100	90	140	10	70	70

¹ Gauge is rotated 180° around the tube as indicated in Figure 4.10

As is shown in Figure 4.14, the failures occurred in the gauge section near the end taper and almost exactly at the centre. The sequence of failures is not known. The ultimate shear strength of 41 MPa was, however, approximately 50% of the expected value. The expiry date for the material had been exceeded. In fact during the filament winding process, partially dried strands of pre-preg were noted; it is these which undoubtedly led to the premature failure. Since the material used to fabricate the specimen was obviously of poor quality, the failure surface was not studied.

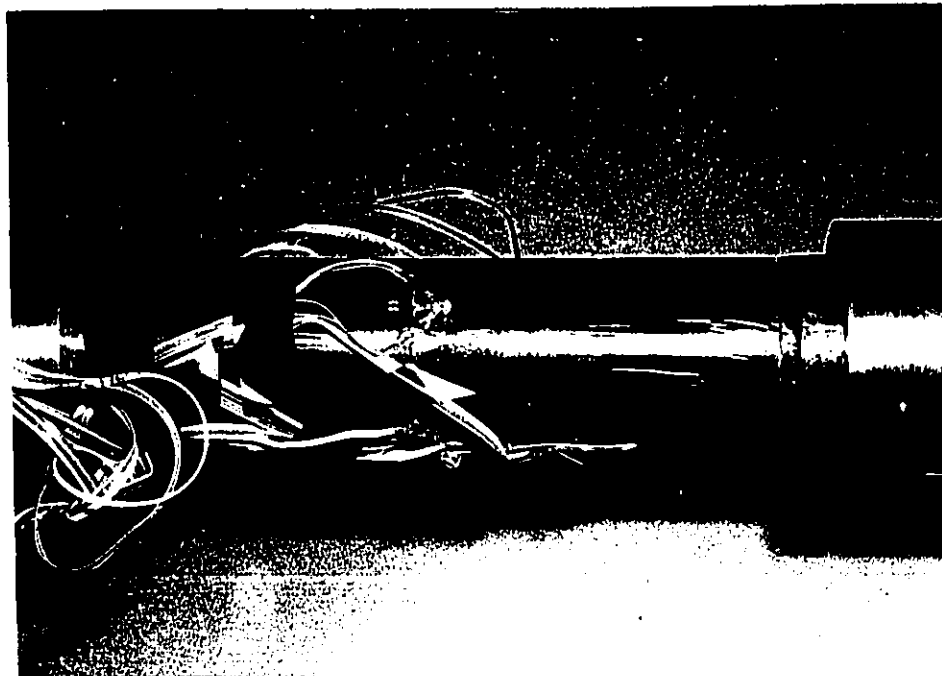


Figure 4.14 Failed tubular in-plane shear specimen.

4.4.2 Joint Evaluation

4.4.2.1 External Shell End Fitting

After the in-plane shear testing of the specimen in Section 4.5.1 the end fittings were cut from the broken tube and placed in a furnace at 600°C in order to burn off the epoxy resin. Several hours were required to burn off sufficient resin to allow disassembling of the outer shell. A second burn off period was required to allow burn off of the newly exposed composite material. The removal of the carbon fibre revealed that some residue of the burn off process had been deposited on the end fitting surfaces which deposits had to be sand blasted off. This entire procedure was time consuming and culminated in the alteration of the end fitting dimensions due to the sand blasting.

Because the tube to be used for evaluating the joint strength was thicker than the original specimen a minor modification to the external shell end fitting was required. Specifically, the pin connections which join the external shell to the inner plug were increased in diameter to withstand a new maximum load of 1075 Nm (9500 in-lbs).

The same basic apparatus that was used for the in-plane shear test was used for the joint strength testing with two exceptions. Firstly, all of the torque transmission pins were replaced with larger diameter pins due to the higher torsional loading for the thicker tubes. Secondly, the chuck that was used to support the lower specimen end fitting was replaced by an aluminum support plate which was bolted to the bed of the radial arm drill press. The redesigned apparatus appears in Figure 4.15.

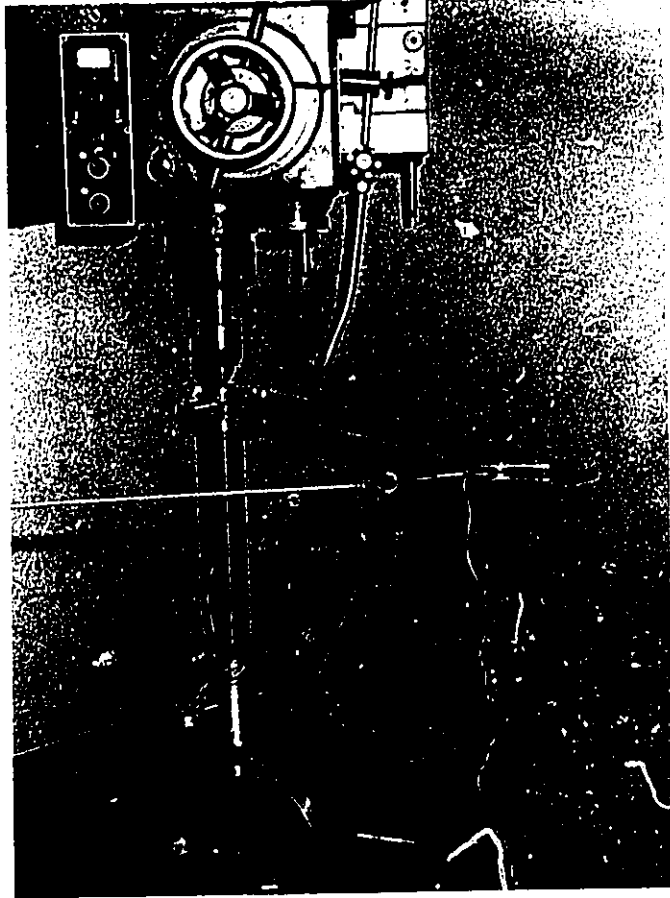


Figure 4.15 Revised preliminary test apparatus.

The composite tube was prepared in the same way as the in-plane shear specimen, except that only one strain gauge was mounted at the centre of the gauge section. Each end fitting was bonded to the composite tube using Hysol 9309.2 adhesive. The testing results for the external shell end fitting are given in Figure 4.16 and Table 4.3. The expressions used to calculate the shear stress and shear strain for Figure 4.16 are [48]:

$$\tau_{12} = \frac{2Tr_o}{\pi(r_o^4 - r_i^4)} \quad (4.1)$$

$$\gamma_{12} = |\epsilon_{-45}| + |\epsilon_{+45}| \quad (4.2)$$

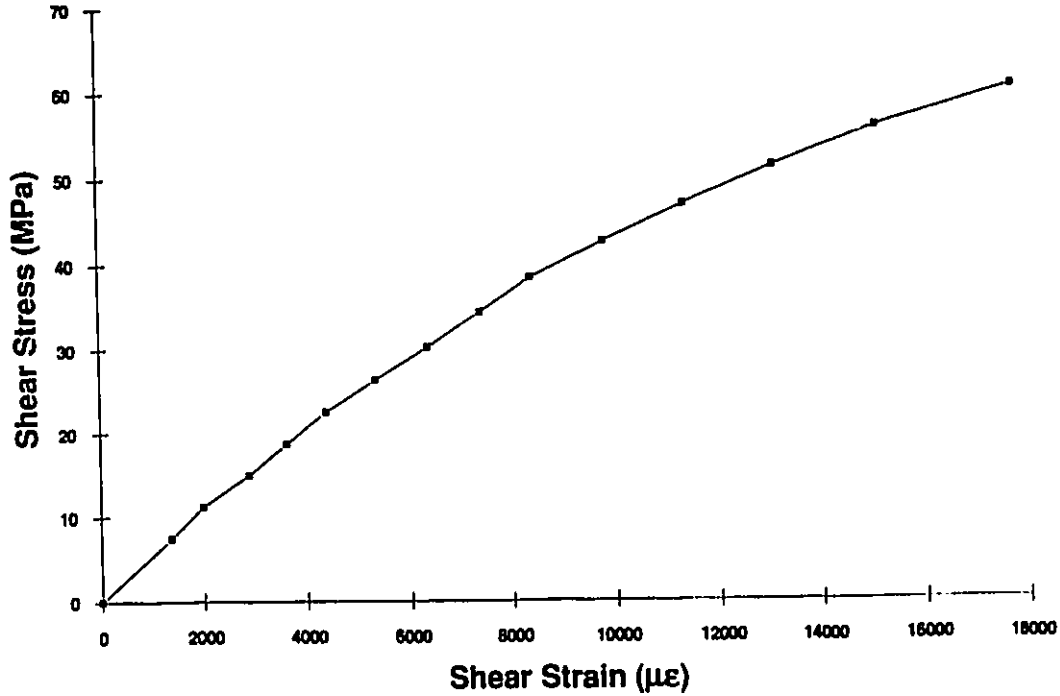


Figure 4.16 Shear stress vs shear strain for external shell end fitting joint strength test.

where r_o and r_i are the inner and outer tube radius, T is the applied torque, $\epsilon_{\pm 45^\circ}$ are the strains measured by the two gauges on the specimen. Once again it is encouraging to note that the tabular results for the two gauges at $\pm 45^\circ$ to the tube axis are very similar proving, first of all, that the testing procedure is repeatable. The axial strain levels are again very low in comparison to the shear values. Thus, a reasonably uniform shear state existed at the centre of the tube gauge section. Despite the manual loading and recording methods employed, the continuity of the results is quite good.

Table 4.3 Summary of Strain Readings For External Shell End Fitting Joint Strength Test

Torque (Nm)	Strain, +45° (µε)	Strain, -45° (µε)	Axial Strain (µε)	Comments
60.4	-720	640	-90	
91.0	-1046	948	-130	
121.8	-1516	1378	-160	
152.1	-1884	1746	-184	
184.6	-2302	2096	-204	
215.4	-2804	2555		
248.3	-3242	3140		
282.8	-3796	3638		
315.4	-4262	4164		
349.6	-4980	4860		Cracking Sound
384.2	-5670	5710	-294	
420.4	-6542	6582		
457.1	-7544	7590		
495.3	-8840	8918		Test Halted

At a load level of approximately 350 Nm, a single cracking noise was heard. Visual inspection of the specimen determined that the upper end fitting had rotated approximately 15° with respect to the tube which indicated that the bond had probably failed. Subsequent loading resulted in further rotation of the tube with respect to the end fitting and increased strain readings which were probably due to the integrity of certain portions of the bond or friction. After removing the specimen from the apparatus the external shell of the upper end fitting was easily removed. This together with the smooth surface of the exposed adhesive indicated that poor bonding between the adhesive and the end fitting had resulted in premature joint failure. The calculated adhesive shear strength for the failed specimen (Figure 4.17) was 1.78 MPa which is substantially lower than the predicted value of 10-15 MPa.

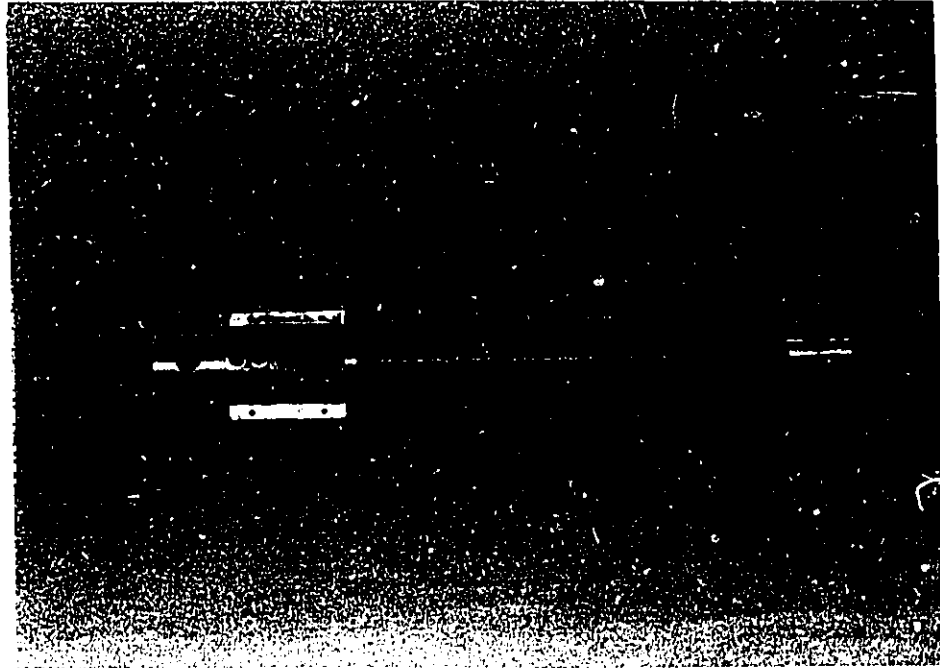


Figure 4.17 Failed external shell end fitting.

4.4.2.2 Tapered Internal End Fitting

The details for composite tube manufacturing, strain gauge attachment and testing remained unchanged for this second joint test. Some changes in the adhesive bonding procedure were required for the tapered internal end fitting. Adhesive was applied to the inner surface of one tube end and the outer surface of one end fitting and then assembled. The tube was supported vertically so that the resin flowed into the gap between the tube and the end fitting. After four hours the adhesive had set and the other end fitting was bonded to the tube. Figure 4.18 shows a tube with the tapered end fittings; the inset photograph shows the uniformity in resin distribution as well as the fillet which formed at the end of the tube end fitting.

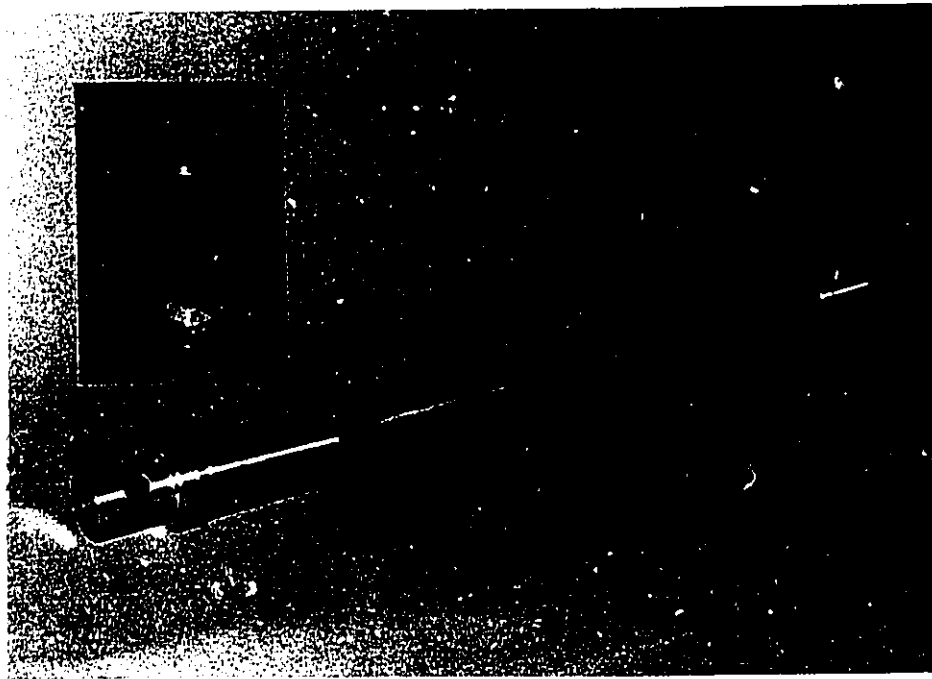


Figure 4.18 Tapered internal end fitting specimen

The test results are summarized in Table 4.4 and Figure 4.19. The second stress-strain curve on the figure is from Figure 4.16. Once again the strain readings (Table 4.4) indicate a reasonably uniform shear state with a low degree of axial loading. It is worth noting that the stress-strain curve (up to a shear strain of approximately 10 000 $\mu\epsilon$) is very similar to the one for the tube tested to evaluate the external shell end fitting. This is significant as it indicates a high degree of repeatability in the entire approach even though a simple apparatus and approximate data collection methods were used. Unlike the previous joint test, the tube failed in the gauge section (Figure 4.20).

Table 4.4 Summary of Strain Readings For Tapered Internal End Fitting Joint Strength Test

Torque (Nm)	Strain, +45° (µε)	Strain, -45° (µε)	Axial Strain (µε)	Comments
28.6	-440	260	-168	
57.3	-776	540	-243	
85.9	-1128	884	-280	
115.8	-1470	1220	-304	
145.6	-1834	1578		
175.7	-2180	1960		
206.1	-2606	2384		
236.8	-3024	2830	-334	
266.4	-3455	3280		
299.0	-3910	3760		
330.6	-4466	4367		
362.4	-5030	4940		Loud Sound
396.3	-5680	5674		
430.7	-6488	6458		Loud Sound
465.5	-7528	7310	-270	
491.3	-8170	8170		Failure

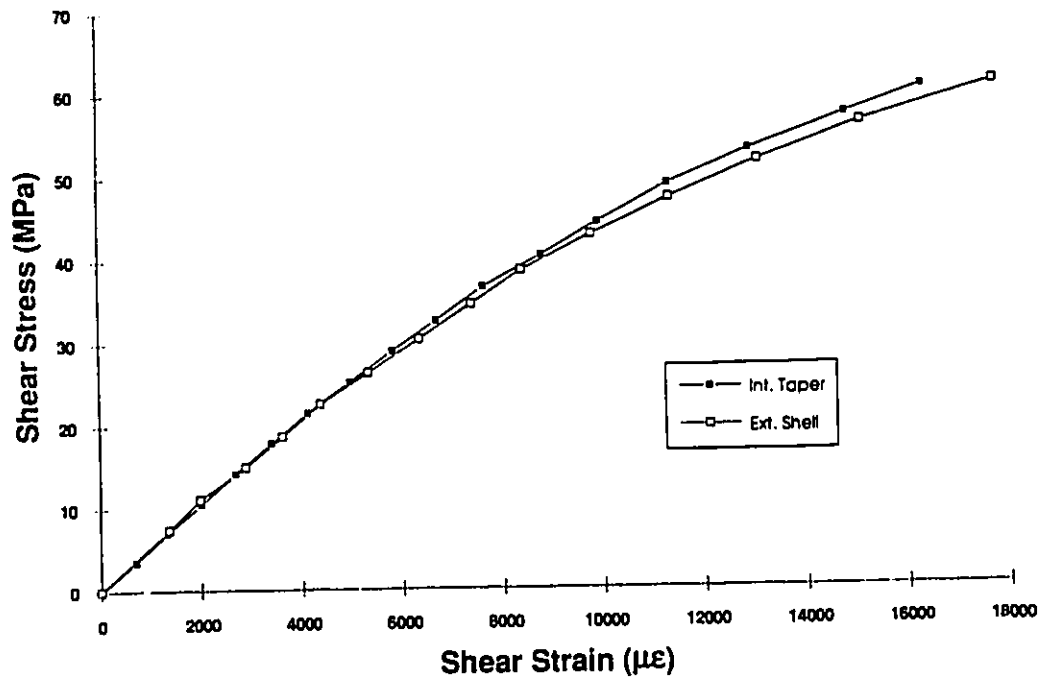


Figure 4.19 Shear stress vs shear strain for internal tapered end fitting joint strength test.



Figure 4.20 Failed tube from tapered internal end fitting joint strength test.

4.5 Preliminary Residual Stress Testing

The ultimate shear strength and strain of the composite tubes can be affected by the existence of high residual stresses and strains developed during the manufacturing process. It is therefore important to know the magnitude of these stresses and strains so that they can be neglected or taken into account. Methods for measuring the circumferential and radial residual stresses of circumferentially wound composite tube have been established at the University of Ottawa [49,50]. The experimental method consists of radially cutting a FRP ring and monitoring the resulting deflection. A circumferentially mounted strain gauge is used to measure the deflection. Expressions have been developed in the literature which utilize the measured strain for calculation of circumferential and radial stresses [50]:

$$\sigma_r = \frac{E_\theta \epsilon_{\theta_i}}{(1-KC_2+KC_3)} \left[1 - \frac{1-c^{(K+1)}}{1-c^{2K}} \left(\frac{r}{b} \right)^{(K+1)} - \frac{1-c^{(K-1)}}{1-c^{2K}} \left(\frac{b}{r} \right)^{(K+1)} c^{(K+1)} \right] \quad (4.3)$$

$$\sigma_\theta = \frac{E_\theta \epsilon_{\theta_i}}{(1-KC_2+KC_3)} \left[1 - \frac{1-c^{(K+1)}}{1-c^{2K}} K \left(\frac{r}{b} \right)^{(K+1)} + \frac{1-c^{(K+1)}}{1-c^{2K}} K c^{(K+1)} \left(\frac{b}{r} \right)^{(K+1)} \right] \quad (4.4)$$

where,

$$K = \sqrt{\frac{E_\theta}{E_r}}, \quad C_2 = \frac{1-c^{K+1}}{1-c^{2K}} c^{K-1}, \quad C_3 = \frac{1-c^{K-1}}{1-c^{2K}}, \quad c = \frac{a}{b}$$

and a and b are the inner and outer radii of the ring, r is the radial position of interest, σ_θ and σ_r are the residual stresses in the circumferential and radial directions, E_θ and E_r are the elastic moduli in the circumferential and radial directions and ϵ_{θ_i} is the measured circumferential strain on the inside surface of the ring.

A ring was cut from the failed specimen used to evaluate the tapered end fitting-composite joint. A strain gauge was subsequently mounted on the internal surface of the ring (Figure 4.21). The ring was then cut radially at 180° to the location of the strain gauge and the subsequent strain recorded. The measured circumferential strain of 910 $\mu\epsilon$ corresponded to a circumferential stress of 53 MPa and a radial stress of 100 KPa. The circumferential stress is very significant considering the shear failure of the tube was 106 MPa while the radial stress of 100 KPa is negligible.



Figure 4.21 Residual stress specimen.

4.6 Summary

The torque tests conducted in this portion of the study revealed, firstly, that a reasonably pure and uniform state of shear existed in the central 100 mm of the gauge section of the tube. If the relatively low axial loading could be eliminated, an even purer state of shear would exist.

The external shell end fitting did not perform to expectations. The re-usability of the end

fitting was over estimated in view of the substantial amount of work required to remove the composite through resin burn off. Of more importance is that the bonding between the external shell and the adhesive was very poor. In addition, the complicated nature of the design was not satisfactory. For all of these reasons, the external shell end fitting was not considered further. Alternatively, the tapered end fitting was light, easy to use and it sustained a respectable torque load without failing. In addition, it might be possible to re-use the end fittings by machining off the composite after testing. The tapered end fitting was, therefore, selected as the end fitting to be used in all subsequent tube testing.

The material to be used in future tubes, AS4/3501-6, is available from Hercules Inc. in pre-preg form only. Therefore, the tubes made from this material would have to be vacuum bagged and cured in an autoclave. The vacuum bagging technique used for this phase of the work did not produce a sufficiently good surface finish for in-plane shear test specimens. Since, surface irregularities could cause premature tube failure, manufacturing procedures which would result in good quality tubes had to be developed.

High residual stresses and strains are not likely to affect the value of the shear moduli found using tube testing; however, they could affect the shear strength and ultimate strain values. For this reason a study into the effect and potential elimination of residual stresses is considered necessary. In order to minimize the usage of material, the rings used in the subsequent residual stress study should be taken from tubes fabricated during the manufacturing study.

Chapter 5.0

Evaluation Of Tubular Specimen Residual Stress

5.1 Background

It is well known that during the manufacturing process, composite materials develop residual stresses which may limit their performance. Although the problem has been the subject of many studies, it is still not possible to explain entirely the observed effects. Residual stresses that develop during the manufacturing of hoop wound tubes are believed to be due to many separate effects which, in general, occur at different stages of the manufacturing process [49,51,52].

In the case of filament winding, a pre-stressed state is induced into the part due to the fibre tension during winding. The fibre must have a certain amount of tension as it is applied to the part; this is one of the basic requirements of filament winding. The successive build up of the layers on a circumferentially wound ring or tube results in a hoop stress which, for thick windings, is compressive on the inside surface of the ring or tube and tensile on the outer surface [51,53,54]. A compressive radial stress also develops. The stresses in cylindrical components due to winding tension have been formulated by Liu and Chamis [54]:

$$\sigma_r = \frac{2KB^{2K}c^{K-1}f(b_m)}{c^{2K}-b_m^{2K}} \left[\left(\frac{c}{r} \right)^{K+1} - \left(\frac{r}{c} \right)^{K-1} \right] - b_m^{K-1}f(r) \left[\alpha \left(\frac{b_m}{r} \right)^{K+1} + \beta \left(\frac{r}{b_m} \right)^{K-1} \right] \quad 5.1$$

$$\sigma_\theta = T(r) + Kb_m^{K-1}f(r) \left[\alpha \left(\frac{b_m}{r} \right)^{K+1} - \beta \left(\frac{r}{b_m} \right)^{K-1} \right] - \frac{2K^2b_m^{2K}c^{K-1}f(b_m)}{c^{2K}-b_m^{2K}} \left[\left(\frac{c}{r} \right)^{K+1} + \left(\frac{r}{c} \right)^{K-1} \right] \quad 5.2$$

where,

$$\alpha = K - \nu_\theta - B, \quad \beta = K + \nu_\theta + B, \quad B = \frac{E_\theta}{E_m(b_m^2 - a_m^2)} [(1 - \nu_m)b_m^2 + (1 + \nu_m)a_m^2]$$

$$f(r) = \int_r^c \frac{\rho^K T(\rho)}{\beta \rho^{2K} + \alpha b_m^{2K}} d\rho, \quad K = \sqrt{\frac{E_\theta}{E_r}}$$

and σ_r and σ_θ are the radial and circumferential stresses, a_m and b_m are the inner and outer radius of the mandrel, c is the outside radius of the composite, E_θ and E_m are the Young's modulus in the circumferential direction of the composite and of the mandrel, $T(r)$ is the tension at radius, r , ν_θ and ν_m are the Poisson's ratio in the circumferential direction of the composite and of the mandrel material. A number of researchers [52,53,55,56] have promoted the use of winding tension as a means to reduce the detrimental effects of other manufacturing parameters for thick and thin cylindrical rings.

A second parameter that affects the residual stress state is the anisotropic shrinkage of the resin during and after curing. This is most pronounced in relatively thick cylindrically wound components where cracking often takes place [51,57,58]. Shiratori [57] proposed that the anisotropic shrinkage was due to the chemical curing of the resin and thermal shrinkage during

cooling of the component to room temperature after curing.

Others have studied the development of residual stresses when the cured component is cooled from its stress free state at the cure temperature to room temperature only [49,51]. For individual laminae there is a difference between the coefficient of thermal expansion parallel to the fibres and perpendicular to the fibres. When these laminae are assembled together to form laminates the laminae are constrained from moving under thermal strain as the individual laminae expansion coefficients dictate. For multi-layered laminates with laminae at different orientations this is especially noteworthy, as it can contribute significantly to the general state of stress in the unidirectional laminates. Wong [51] derived expressions for radial and tangential stresses for circumferentially wound cylindrical components due to the cool down after curing:

$$\sigma_r = \frac{E_r(K^2 - \nu_\theta^2)(\alpha_\theta - \alpha_r)T}{(1 - \nu_r \nu_\theta)(1 - K^2)} \left[\frac{1 - C^{-K-1} \left(\frac{r}{a}\right)^{K-1}}{1 - C^{-2K}} + \frac{1 - C^{K-1} \left(\frac{r}{a}\right)^{-K-1}}{1 - C^{2K}} - 1 \right] \quad 5.4$$

$$\sigma_\theta = \frac{E_r(K^2 - \nu_\theta^2)(\alpha_\theta - \alpha_r)T}{(1 - \nu_r \nu_\theta)(1 - K^2)} \left[K \frac{1 - C^{-K-1} \left(\frac{r}{a}\right)^{K-1}}{1 - C^{-2K}} - K \frac{1 - C^{K-1} \left(\frac{r}{a}\right)^{-K-1}}{1 - C^{2K}} - 1 \right] \quad 5.4$$

where, a and b are the inner and outer radius of the composite, α_θ and α_r are the coefficients of thermal expansion (C.T.E.) in the circumferential and radial directions and $C=a/b$.

Aleong [49] experimentally measured the residual stress in thin rings and compared the results to the algebraic sum of the theoretical residual stresses for thermal cool down and winding tension for three different materials and various fibre tensions. The results of this comparison, which appear in Figure 5.1 indicate two important results. The first is that the experimental residual stress greatly exceeds the theoretical residual stress prediction. Secondly, the difference between the experimental results and theoretical results increases as the stiffness of the fibre increases. Together these observations suggest the existence of another residual stress component as yet unaccounted for.

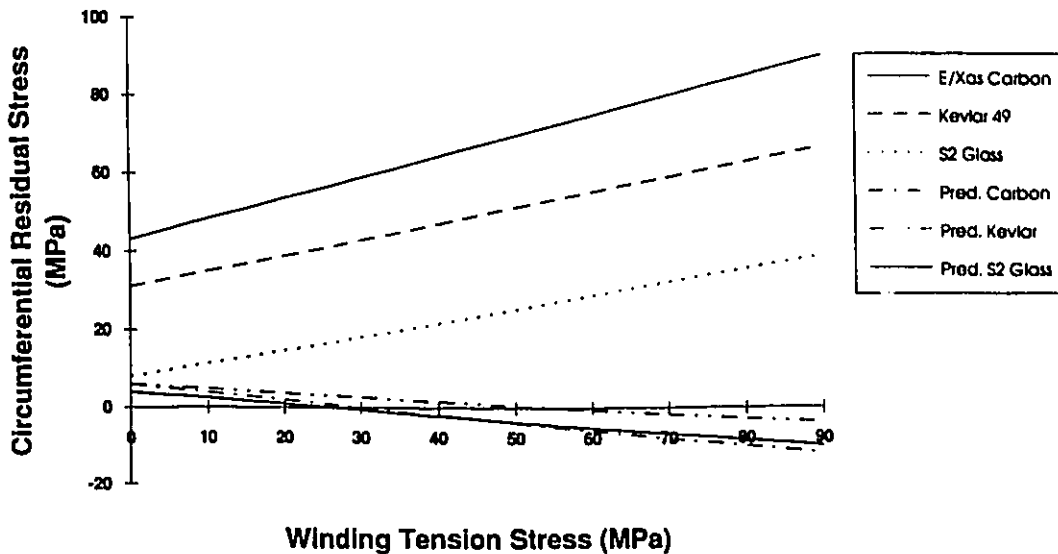


Figure 5.1 Plot of theoretical and experimental residual stress vs winding tension for three materials (modified from [49]).

Reuter Jr. [52] studied the effect of the entire manufacturing cycle on the residual stress of thin hoop wound rings manufactured using fibre glass/epoxy pre-preg material. This is a notable difference from the work discussed thus far which assumes the rings are made using a wet winding process. Thin shells were defined as having a minimum radius to thickness ratio of 20. The process of manufacturing and curing a thin ring was split into seven different phases:

- 1) Filament winding
- 2) Application of autoclave pressure
- 3) Temperature to cure temperature
- 4) Layer migration
- 5) Cooling to room temperature
- 6) Pressure removal
- 7) Removal of the mandrel

Reuter Jr. suggested that while the ring is at the cure temperature the resin softens and flows, dimensional changes occur because voids are filled, the resin shrinks and compaction takes place. This migration phase was considered the dominant mechanism in the curing process. The removal of the mandrel is a phase specific to those materials, for example, fibre-glass/epoxy, which shrinks onto the mandrel after curing. The removal of the mandrel further relaxes stress within the ring. Reuter developed a model to predict the effect of these phases on the state of stress in the ring after curing and in addition showed that by varying the fibre tension between layers the residual stress could be reduced for thin fibre-glass/epoxy rings. Figure 5.2 shows the predicted radial load intensity versus thickness for a constant 4.4 N (1 lb) tension and for a tension variation of 4.4 N (1 lb) to 26.7 N (6 lbs) from the inside to the outside of the ring. The material of this study, however, does not shrink on to the mandrel (graphite/epoxy tubes were loose on the mandrel after cool down to room temperature) and it was not established by Reuter Jr. that his model would work in this case.

It is apparent that predicting the stresses which result when manufacturing a component as simple as a thin hoop wound tube is a very complex task. It is not within the scope of this study to develop or definitively evaluate existing models but to identify whether residual stresses exist in tubular in-plane shear specimens, whether or not they can easily be eliminated and what impact they might have on the results for the in-plane shear specimen.

5.2 Experimental Investigation

5.2.1 Introduction

The focus of this study centred on two areas. Firstly, it was determined whether or not significant residual stresses existed in the tubular in-plane shear specimens. Secondly, if significant residual stresses did exist, it would be necessary to determine what controllable manufacturing parameter(s) could be used to reduce them. The experimental investigation used

sections from five long hoop wound tubes (PR/R1 to PR/R4 and PR 7¹ - See manufacturing table in Appendix C) which were initially similar and eventually identical to the tubular in-plane shear test specimens.

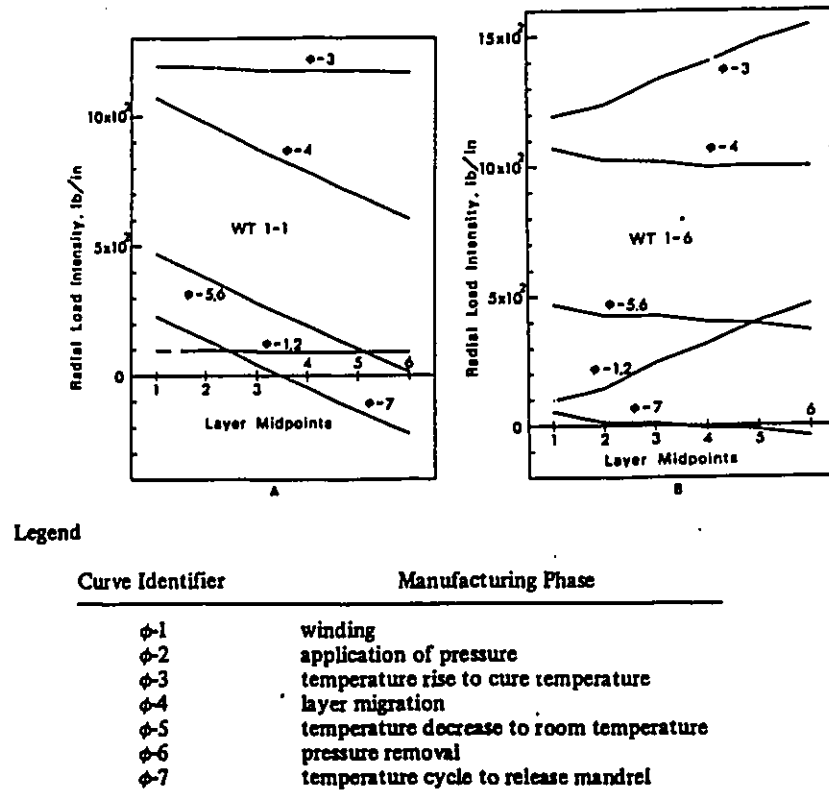


Figure 5.2 a) Load intensity vs radial position (layer no.) for fibreglass/epoxy at constant winding tension. b) Load intensity vs radial position for varying tension [52].

5.2.2 Filament Winding Of Residual Stress Specimens

Five hoop wound residual stress specimens were prepared to study various manufacturing parameters of residual stress, namely, mandrel material, winding tension and lay-up. Each tube was manufactured by the filament winding of Hercules 12K AS4/3501-6 pre-preg tow onto a tubular mandrel.

¹ Number for tubular specimens from which residual stress specimens were taken

The first two residual stress specimens were cut from the centre section of 200 mm (8 inch) long specimens which were filament wound and cured on a mild steel three-piece mandrel. Since high tension winding was used (initially thought to reduce the residual stresses) for the first of these specimens, the ends of the mandrel had holes for pins which were used to ensure that tension was maintained from layer to layer. Although unidirectional 88° tubes were used for the final in-plane shear testing, for these initial tubes a $[\pm 88^\circ]$ general lay-up was used. The 7 mm width of the pre-preg tow established the 88° angle (a more narrow tow would yield a wind angle closer to 90°). Since only the trends in residual stress were of interest this bi-directional laminate was selected. Furthermore, winding a unidirectional tube at high tension is very difficult and time consuming operation. Finally, appreciable differences in residual stress levels between the two laminate configurations were not predicted.

5.2.3 Residual Strain Test Results

In an effort to minimize residual stresses, the first tube was wound with pre-preg tow at high tension, 178 N (40 lbs). Eight layers were used so that the thickness of the specimen would be the same as that of the in-plane shear specimens (1.27 mm/0.050 inches). A 16 mm section was cut from the centre of the tube. The edges of the specimen were then ground parallel to a final specimen length of 15.24 mm (0.6 inches). This size was chosen because it was the longest specimen for which the strain gauge could be easily mounted in the middle of the inner surface. The strain gauge location is consistent with guidelines recommended for the radial cut method [49,50]. A section of the specimen opposite the strain gauge was then cut away and the resulting deflection strain read on a strain indicator. The resulting circumferential strain was 1242 $\mu\epsilon$ while the axial strain was found to be a negligible 20 $\mu\epsilon$. It was also found that the measured thickness of the tube was actually 1.52 mm (0.06 inches); therefore, seven layers were used in subsequent residual stress specimens using this batch of material.

The second specimen was wound at 17.8 N (4 lbs) of tow tension using the same steel mandrel as the first specimen. As mentioned, seven layers were used to wind the specimen in an attempt to attain the 1.27 mm wall thickness required for in-plane shear testing. This test was

conducted to verify the work of Aleong [49] which showed lower winding tension resulted in a lower residual stress. As expected, the strain level was substantially lower than in the 178 N (40 lbs) tow tension case. The residual circumferential strain was found to be 612 $\mu\epsilon$ while the axial strain was again close to zero and was thus not measured in subsequent testing.

In two of the attempts to describe cylindrical hoop wound composite behaviour it was mentioned that a change in residual stress would occur due to a change in mandrel material [53,59]. Al-Qureshi [59] mentioned the effect of mandrel material in particular, but only in the sense that a high mandrel stiffness reduces residual stress, though he does not prove this. Liu and Chamis [53] also concluded that high mandrel stiffness reduces residual stress levels. The results of a theoretical study by Liu and Chamis [53] indicate that steel mandrels produce lower residual stress than do aluminum mandrels. Plaster mandrels which are less stiff than steel, however produce similar residual stress levels to those found using a steel mandrel. This work suggests that another material parameter is responsible for the differences in residual stress levels i.e. mandrel coefficient of thermal expansion (C.T.E.). The effect of mandrel CTE has not been discussed in the literature.

Since mandrel CTE and stiffness are controllable parameters, a second middle section of the mandrel was manufactured from aluminum which has a substantially higher coefficient of thermal expansion than that of steel as well as a lower mandrel stiffness. The circumferential residual strain level for this specimen was 1078 $\mu\epsilon$ which is 75% higher than in the specimen wound on the steel mandrel. This is in agreement with the conclusion of Liu and Chamis and an indication that there is an interaction between the coefficient of thermal expansion and residual stress and/or mandrel stiffness. Thus the use of a low C.T.E./high stiffness mandrel appears to be a realistic way to reduce residual strain (and thus residual stress). Ideally, to determine whether coefficient of thermal expansion or mandrel stiffness is required for lower residual stress, a very low C.T.E. and low stiffness mandrel material should be used. Time and cost considerations resulted in the use of a stainless steel tubular mandrel. The effects of resultant residual stresses for this material on the shear state are evaluated in Section 5.3.

The preliminary residual stress investigation using $[\pm 88^\circ]$ laminates was completed at this point. Future specimens were wound using a unidirectional 88° lay-up. The next specimen which was wound on the aluminum mandrel had a residual strain of $1190 \mu\epsilon$. This value is only 10% higher than for the $\pm 88^\circ$ specimen which indicates that the bi-directional tube was a reasonable method for estimating the effect of manufacturing parameters on residual stress.

At this point a new batch of AS4/3501-6 material was used which was marginally thinner than the first batch. For this reason the final specimen used an 8 layer winding. A unidirectional lay-up was again selected, $[88]_8$, a stainless steel tubular mandrel was used and a relatively low 14.7 N tension was applied to the pre-preg tow during winding. Furthermore the dimensions of the tube were consistent with those of an actual in-plane shear test specimen. This test was done to determine the residual strains in a typical in-plane shear specimen and to compare this strain to that of the previous bi-directional specimen which was also fabricated using the steel mandrel. The resulting strain level was $571 \mu\epsilon$ which is less than 10% lower than the bi-directional specimen. This confirmed the selection of the steel mandrel for the 88° unidirectional case. A summary of the results for the residual stress specimens appears in Table 5.1.

Table 5.1 Summary Of Residual Stress Specimen Results

Specimen	Mandrel Material	Winding Tension (N)	Lay-up	Residual Strain($\mu\epsilon$)
Resid 1	Steel	178.0	$[\pm 88]_4$	1242
Resid 2	Steel	17.8	$[\pm 88, \pm 88, \pm 88, 88]$	612
Resid 3	Aluminum	17.8	$[\pm 88, \pm 88, \pm 88, 88]$	1078
Resid 4	Aluminum	17.8	$[88]_7$	1190
Resid 5	Steel	17.8	$[88]_8$	571

5.3 Contribution of Residual Stress To Failure of In-plane Shear Specimens

The best approach to conclusively determine the contribution of residual stress to the failure of in-plane shear specimens would be to test two or more sets of tubes (in torsion) until failure with each set having a different residual stress level. Based on the residual stress testing already completed, the simplest method would be to manufacture specimens using mandrels which have very different C.T.E.'s. Since such extensive testing was beyond the scope of this project, an established failure theory was used to estimate the impact of circumferential residual stress on failure.

Many failure theories have been developed for determining expected failure levels of composite materials. Nahas [60] reviewed existing theories of failure of laminated fibre reinforced composite materials. Four theories based upon a plane stress assumption were identified as being used most in practice: Maximum strain theory, Azzi-Tsai theory, Tsai-Wu theory and Maximum stress theory. The maximum strain and the maximum stress theories are the most widely preferred [60]; however, failures in different modes are assumed to occur independently. For this study it is the interaction between residual stress levels in the circumferential direction and the applied shear stress acting on a plane perpendicular to that of the circumferential residual stress that is of interest. The Tsai-Wu theory requires biaxial test data which is not readily available and thus the Azzi-Tsai failure theory was selected.

The Azzi-Tsai failure theory is a derivative of the Hill theory which is itself a derivative of the Von Mises-Hencky maximum distortional energy theory (isotropic failure theory). The Von Mises-Hencky theory was modified by Hill to include anisotropic materials and the interaction between the failure modes. The one disadvantage of the Hill theory is that it does not allow for different strengths for the tensile and compressive modes. When using the Azzi-Tsai theory one uses the principal strengths compatible with the sense of the prevailing stresses (i.e. compressive strength for negative normal stress resultant). The Azzi-Tsai failure criterion is:

$$\frac{\sigma_1^2}{X^2} - \frac{\sigma_1\sigma_2}{X^2} + \frac{\sigma_2^2}{Y^2} + \frac{\tau_{12}^2}{S^2} = 1 \quad 5.5$$

where X is the either the tensile or compressive strength in the fibre direction, Y is the tensile or compressive strength in the transverse direction, S is the principal in-plane shear strength and σ_1 , σ_2 and τ_{12} are the applied principal stresses (Figure 1.2).

The strength values for tension and longitudinal compression were taken from an I.A.R. study on the material properties of AS4/3501-6 [61]. The transverse compressive strength of AS4/3501-6 was taken from [62]. Shear failure levels, however, are not established with any degree of confidence for any material due to the controversy over the appropriateness of the established shear tests to predict shear failure. An estimate of 100 MPa was selected as the failure level to be used for the study [62,63]. Tension and compression modulus values were taken from [61] while a shear modulus value of 6.0 GPa was assumed. A summary of the material properties for AS4/3501-6 appears in Appendix B.

The circumferential and radial residual stress distributions were calculated for each specimen from the residual strain measurements using the equations 4.3 and 4.4. These stresses are summarized in Table 5.2. The radial residual stress is not accounted for in the plane stress (2-D) failure theory in Equation 5.5. It can be shown that, if a 3-D formulation is used, the contribution from a σ_3^2/Z^2 term is negligible. For these reasons the effect of the radial residual stress was not considered. The effect of residual stress on each individual specimen tested in the experimental phase is not necessary nor desired for this study. Rather, two general residual stress levels denoting high and low residual stresses were selected based on the stress levels that appear in Table 5.2. The higher stress level is in the region of ± 160 MPa and can be associated with an aluminum mandrel (Resid 4) while the lower stress level was selected as ± 85 MPa (steel mandrel, Resid 5). The shear stress resulting from torsional loading was taken to be the assumed shear strength, i.e. 100 MPa. The stresses in the material directions for each layer were

determined by transforming the total loading (torsional shear stress and residual stress) from the load axes.

Table 5.2 Summary of Residual Stress Levels

Specimen	Circ. Stress Inside Layer (MPa)	Circ. Stress Outside Layer (MPa)	Maximum Radial Stress (MPa)
Resid 1	168.7	-168.6	-2.2
Resid 2	83.1	-83.1	-1.1
Resid 3	146.4	-146.3	-1.9
Resid 4	161.6	-161.5	-2.1
Resid 5	77.6	-77.5	-1.0

Two torsional loading cases were of interest, one that produced a negative shear stress of -100 MPa and one that produced a shear stress of +100 MPa. Furthermore, inside and outside layer failure levels were determined using Azzi-Tsai failure theory since these layers experience the highest circumferential residual stress levels. The results from substituting the stresses in the material directions in Equation 5.5 are summarized in Table 5.3.

A tubular in-plane shear specimen which is wound at a wind angle of 88° has fibres arranged in a helix. When such a tube is torqued in one direction the helix is, in effect, tightened (the helix diameter is reduced), while in the opposite direction the helix tends to open up (the helix diameter becomes larger). In Table 5.3 a positive torque corresponds to the tightening mode of the helix while a negative torque corresponds to the opening mode of the helix. In the case of pure torque loading the results indicate that a tube loaded with a positive torque should be slightly stronger than one which experiences a negative torque. Furthermore, failure occurs at the same time both on the inside and the outside of the tube. When circumferential residual stress is added to the loading the failure levels change drastically. A tube which experiences positive torque loading will fail first on the outside. A tube loaded in the negative direction will fail first on the inside. The failure levels from positive to negative loading are not appreciably

different though tubes loaded in the positive direction appear to be a slightly stronger.

Table 5.3 Summary Of Failure Levels For $[88]_8$ Tubes

Loading Case	Failure Level		%Diff. From Positive Pure Torque	
	Inside	Outside	Inside	Outside
Pure Torque (+100 MPa)	1.00	1.00	-	-
Comb. Loading (160 MPa)	0.89	1.14	-11.0	14.0
Comb. Loading (85 MPa)	0.94	1.06	-6.0	6.0
Pure Torque (-100 MPa)	1.02	1.02	-	-
Comb. Loading (160 MPa)	1.14	0.95	14.0	-5.0
Comb. Loading (85 MPa)	1.08	0.97	8.0	-3.0

The results shown in Table 5.3 mainly serve to underscore the necessity of eliminating residual stress from the specimen. For both pure torque cases the failure levels of the two laminates are consistent through the thickness. When residual stress is added to the loading failure levels change drastically. There is also a considerable difference in failure levels from the inside layer to the outside layer. For positive shear loading, failure levels increase for the outer layer while decreasing for the inner layer. This trend is reversed for negative shear loading. In either loading case the effect of residual stress reduces as the level of stress reduces. The higher residual stress level (160 MPa) corresponds to the use of an aluminum (or high CTE) mandrel during the manufacturing process while the lower residual stress level (85 MPa) corresponds to the use of a steel mandrel. The results of Table 5.3 indicate, therefore, that a steel mandrel should be chosen instead of an aluminum one.

Chapter 6.0

Manufacturing Of Tubular In-Plane Shear Specimens

6.1 Introduction

Several different requirements were identified for the AS4/3501-6 tubes to meet in order to be accepted as in-plane shear specimens. Principal concerns were surface finish, uniform tube thickness, laminate consolidation and through-the-thickness defects. A minimum acceptable surface finish was one which had no serious blemishes on the surface of the tube's gauges section. The thickness of the tube must be uniform around the tube circumference to within 5% since for a constant load, a 5% increase (decrease) in thickness translates into a 5% decrease (increase) in shear stress. Five percent variation in shear stress between specimens is common for the most used shear tests [3,5,12,63] and so it is used as the maximum tube thickness variation. Laminate consolidation was determined by reflected light microscopy. During the development of manufacturing techniques tube cross-sections were periodically cut from cured tubes to determine whether or not resin rich areas existed between the laminae. Through thickness defects were detected using ultrasonic C-scan techniques. Tubes were accepted for in-plane shear specimens only if there were no through thickness defects detected. A summary table describing the manufacturing detail is given in Appendix C.

6.2 Development Of Vacuum Bagging Techniques

6.2.1 Conventional Bagging

As a first attempt tube PR/R1 was cured using a practice similar to that used for flat panels. The filament wound tube was wrapped with a release film and then wrapped in a breathing layer (a fibrous material which allows the escape of air and volatiles during cure). This package was enclosed in a vacuum bag as shown in Figure 6.1 and then cured in the autoclave using the recommended AS4/3501-6 cure schedule. The cured tube had a very rough surface finish with many deep and irregular impressions on its outer surface which were caused by the wrinkling of the vacuum bag. The development of the final bagging techniques involved the evaluation of many different approaches, beginning with traditional methods. Many valuable details were found during this procedure which are applicable for composite tube fabrication in general. For this reason the different approaches are described and discussed in detail.

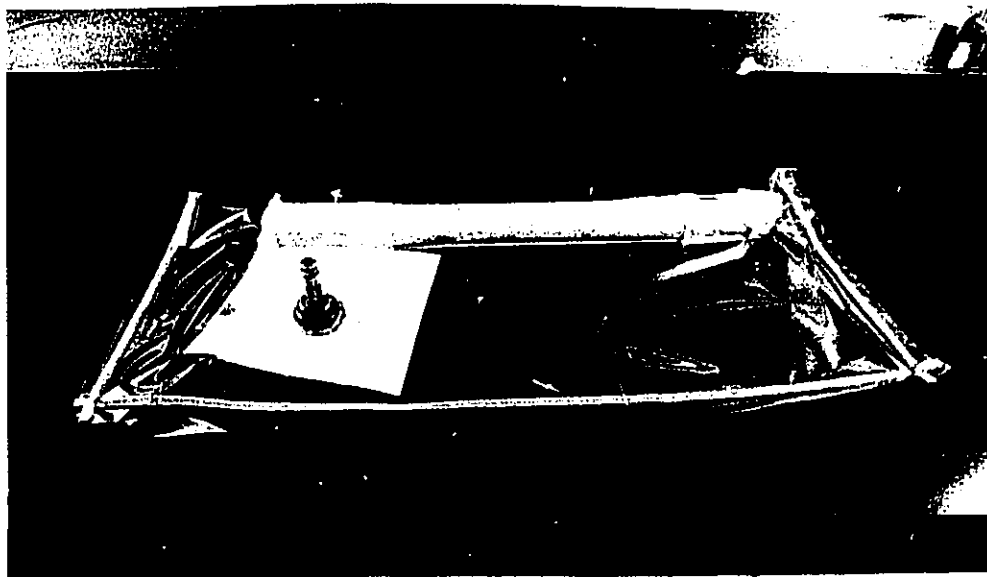


Figure 6.1 Tube enclosed in vacuum bag.

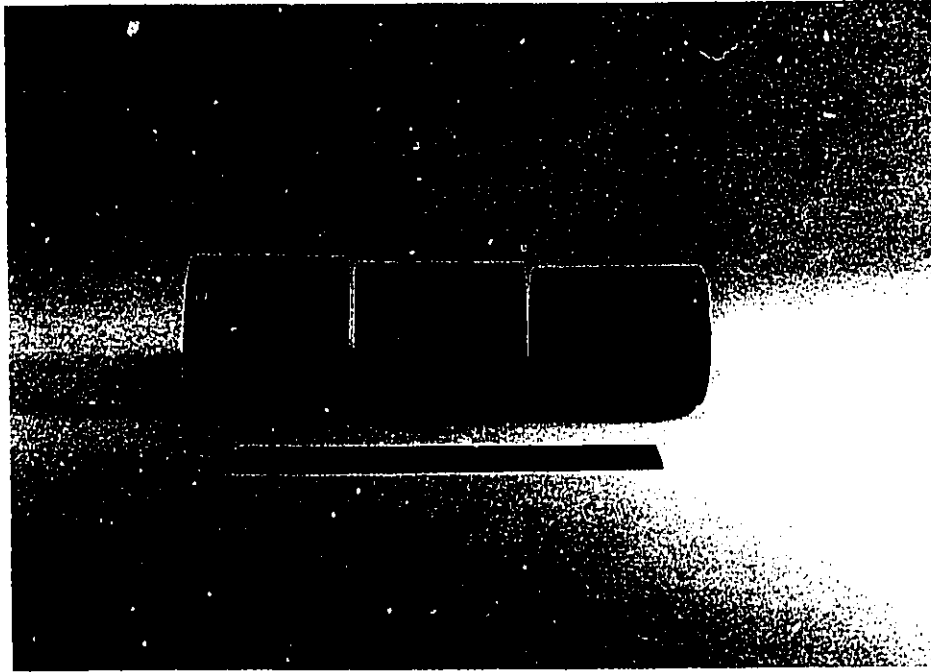


Figure 6.2 Surface finish of tubes PR/R1, PR/R2 and PR/R3 (left to right).

6.2.2 Shrink Tape Wrap Technique

The second attempt to meet the specified requirements consisted of wrapping the tube in shrink tape prior to the addition of the release film. A tube (PR/R2) was first wrapped using a 25.4 mm wide 3% shrink tape (Airtech B-500-OPP) which was applied by hand using high tension. Heat was then applied to consolidate the tape wrap using a hand held dryer. A second layer of 37.6 mm wide 30% shrink tape (Airtech A575) was also applied to the tube by hand also under high tension. Heat was again used to consolidate the wrap. The bagging procedure from that point was as for the first technique. Figure 6.3 shows the degree of overlap for both shrink tape layers.

After curing well defined creases in helical patterns around the tube were evident on the tube surface. One helical pattern corresponded exactly to the overlap of the A575 tape (overlapped by one half of the tape width). A second helical crease pattern occurred (but was

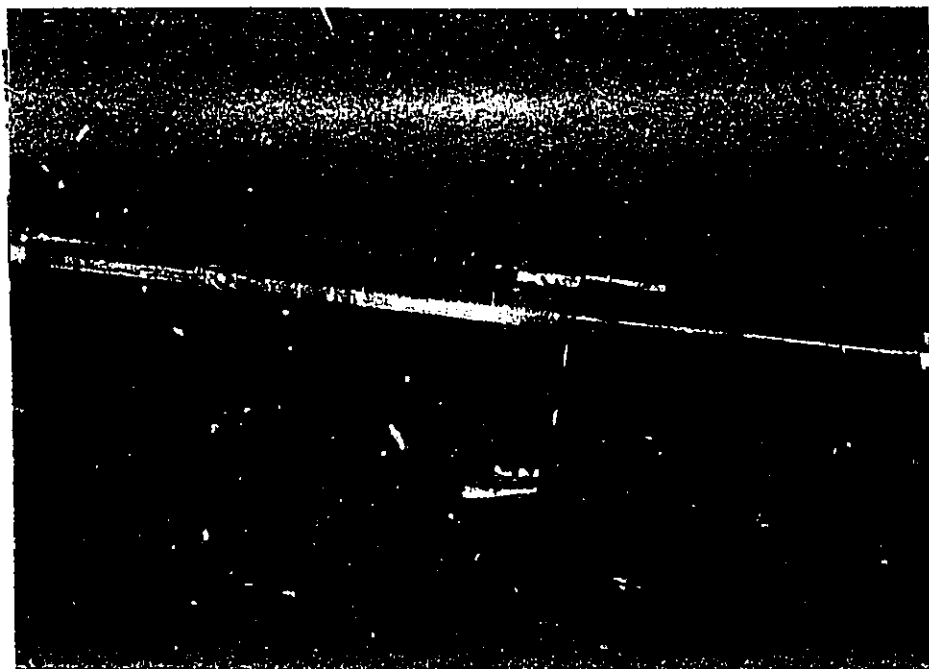


Figure 6.3 Double shrink tape wrapping technique (PR/R2 and PR/R3).

wavy in nature) which corresponded to the B-500-OPP tape overlap. The waviness was due to the buckling of this tape beneath the higher shrinkage tape.

Two more tubes were cured using this technique (PR/R3 and PR/R4) except that pre-heating of the shrink tape was abandoned, tension was lessened and the degree of overlap was increased and then decreased. In each case the helical pattern was reduced in severity, however, it could not be eliminated.

The B-500-OPP material was originally used as a release film for the higher percentage shrink tape. For the next approach it was removed in an attempt to improve the surface finish by eliminating the wavy helical pattern. Thus only the A575 high shrink tape was used. The cured tube is shown Figure 6.4 The helical pattern is even more pronounced than in the previous attempts. Further attempts to use shrink tape as the primary means for producing a good surface finish were abandoned at this point.

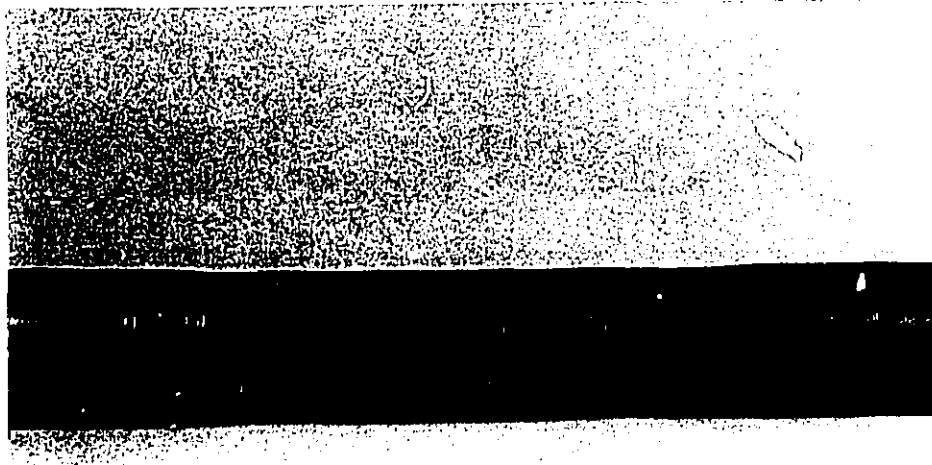


Figure 6.4 Surface of single shrink tape cured tube (PR 1).

6.2.3 Aluminum Sleeve Bagging Technique

The consolidation of the laminae was, however, quite good for all of the tubes which utilized shrink tape. Figure 6.5 shows a section of tube PR/R3 viewed under a microscope at a magnification of 67.5 times. The figure is typical of the entire tube cross section with no obvious resin rich regions or defects. Ultrasonic examination of these tubes was not possible as the surface roughness of the tubes scattered the sound beam in the region of the creases.

The helical patterns which resulted during the curing of the tubes wrapped with shrink tape were due to the tape over-lap. When the resin flowed at the cure temperature it was believed that it flowed into the overlapped regions. It was therefore proposed that a continuous and semi-rigid material layer be used beneath the shrink tape to provide a smooth surface finish while taking advantage of the good consolidation afforded by the shrink tape. To accomplish this, a layer of aluminum foil, 0.08 mm (0.003 inch) thick, was wrapped around the tube (tube PR 2) gauge section. The foil was a particularly good choice because any continuous layer



Figure 6.5 Typical view of tube PR/R3 at a magnification of 67.5 times.

beneath the shrink tape would necessarily have one seam running along the axis of the tube. The 0.08 mm thickness should leave a negligible seam. A layer of 76 mm wide A575 shrink tape was wrapped (with 50% overlap as before) around the entire tube to complete the process (see Figure 6.6). This combination was then cured using the normal bagging procedure. The resulting tube surface was excellent except that in several places the vacuum bag pulled the foil away from the tube causing wrinkling. Resin and fibre flowed into the wrinkles during curing which translated into long axial creases in the tube surface (Figure 6.7).

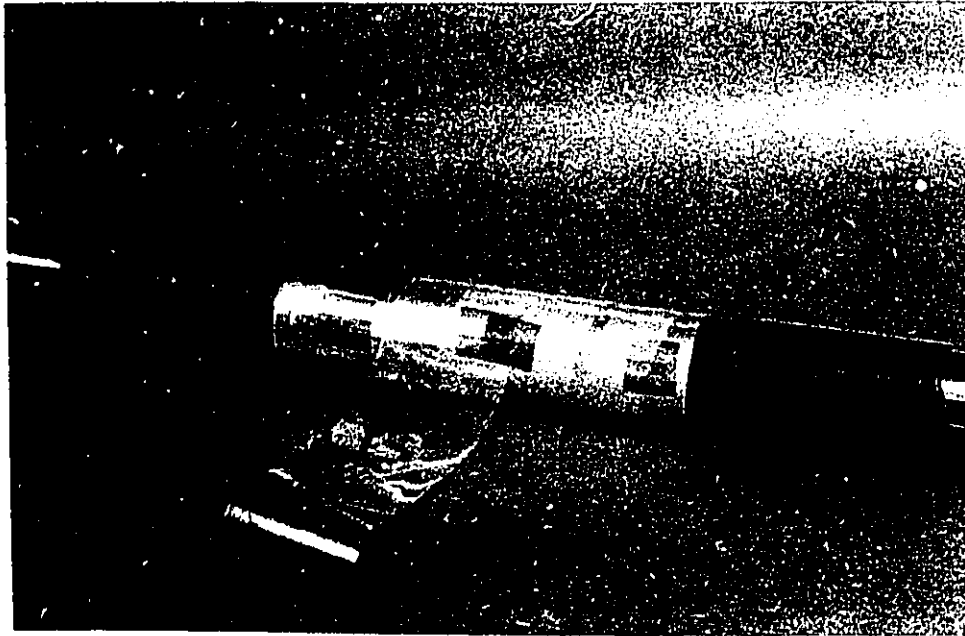


Figure 6.6 Aluminum foil and single shrink tape wrapping technique (Tube PR 2).



Figure 6.7 Cure tube with axial seams (Tube PR 2).

Since the thin seam and the good surface finish away from the creases were positive results, a second attempt using aluminum foil was conducted. This time two layers of foil were wrapped around the gauge section (tube PR 3) with the two axial seams at 180° to one another. A special effort was made to wrap the foil tightly around the tube so that the vacuum would not cause wrinkling. The cured tube, however, suffered from similar creases to those of PR 2, though the number of creases were reduced. As a result, the use of aluminum foil was deemed unsuitable for further study.

In spite of the wrinkling, the use of aluminum foil did produce regions of excellent surface finish thus a stiffer wrapping layer was required which should yield both an excellent tube finish and good consolidation. The obvious limitation is that a thicker aluminum layer would produce a thicker axial seam. To check the validity of the method a 203 mm long, 7.6 mm (0.03 inch) thick aluminum sheet was rolled into a cylindrical sleeve to fit the gauge section of a tube (tube PR 4). The sleeve was placed on the tube by sliding it over the built up ends of the tube and onto the gauge section. It was then tightly squeezed around the tube, while mylar tape was used to secure the position. Again shrink tape was wrapped tightly over the entire tube and the combination vacuum bagged and cured (Figure 6.8). The cured tube (one half is shown in Figure 6.9) had an excellent surface finish generally, however the axial seam as expected was thick. Furthermore, a thick seam resulted along the end of the gauge section which could result in early failure of the tube (see arrow).

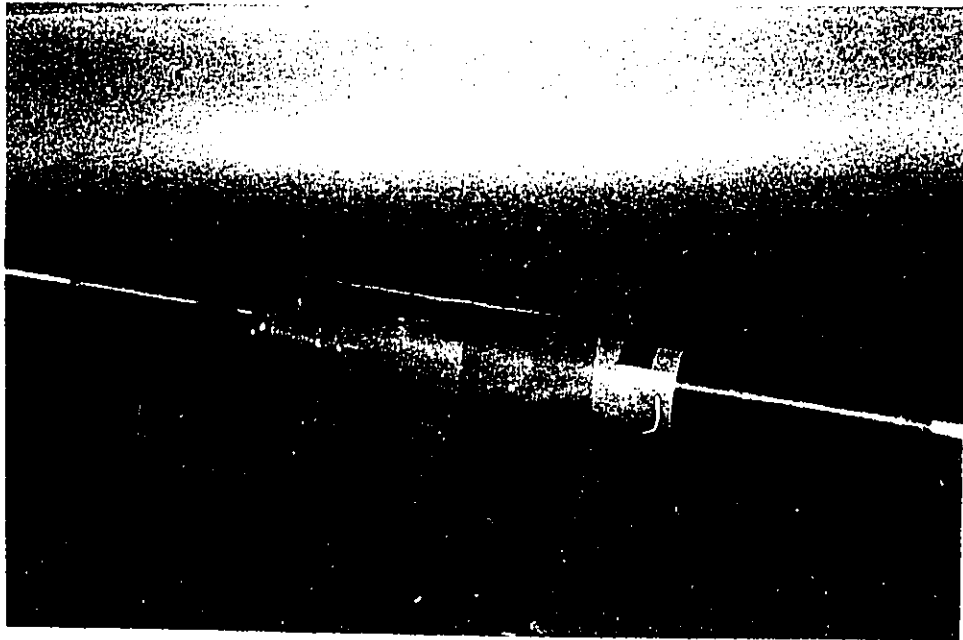


Figure 6.8 Aluminum sleeve with single shrink tape wrapping technique (PR 4).



Figure 6.9 One end of tube PR 4 after cure.

The tube was C-scanned to check for defects. As expected, there was one very dark region (indicating a geometric discontinuity or defect) all along the tube seam. A cross-section was cut from the tube and photographs taken of the seam at a magnification of 300x. The microphotograph which appears as Figure 6.10 shows that the seam was a predominantly resin rich region on the outside of the tube (see arrow). Thus the dark region was believed to be caused by the irregularity of the seam itself rather than any internal voids or delaminations.

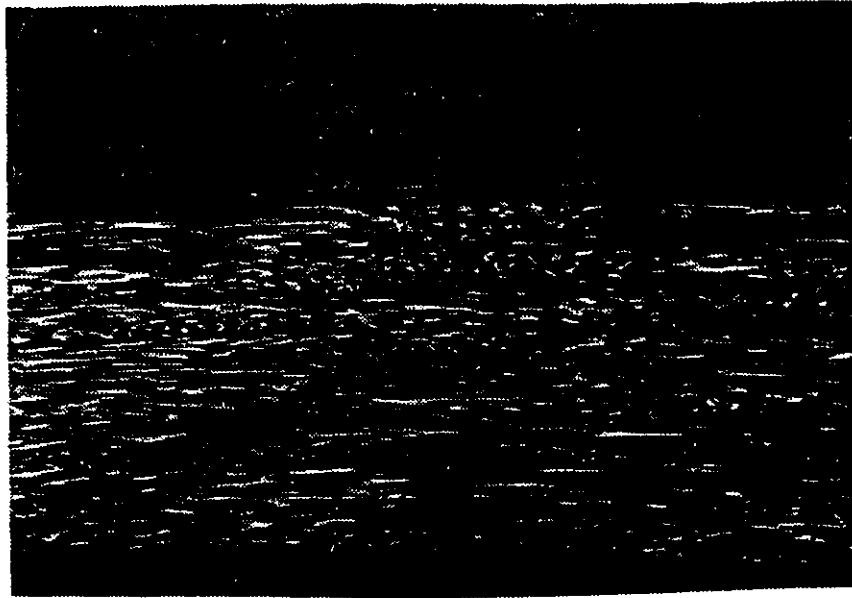


Figure 6.10 Microscopic view of seam overlap (PR 4).

In an attempt to reduce the thickness of the axial and end seams all of the edges of the aluminum sleeve were sanded using a belt sander. The edges were reduced to a nominal thickness of $1.3 \text{ mm} \pm 0.5 \text{ mm}$. The modified aluminum sheet was employed to fabricate a tube (Tube PR 5); however, there were large axial and circumferential seams at the edge of the 203 mm gauge section. A second and third attempt at thinning the sleeve edges (Tube PR 6 and PR 7) resulted in some decrease in seam thickness but not enough to consider the tubes acceptable for in-plane shear specimens.

Two additional types of joint were tried in an effort to eliminate the axial seam. The first

was a butt joint (Tube 0) which consisted of cutting the aluminum sleeve to a length which exactly corresponded to the tube circumference. When the sheet was rolled, the two edges of the seam were to just meet when the sleeve was placed around the tube gauge section. Furthermore a 0.08 mm thick teflon film was placed between this butt joint and the wound composite to prevent resin from escaping. This technique was an improvement in terms of the irregularity of the outer diameter (and thus the tube thickness) but the teflon film wrinkled considerably leaving many small creases all along the seam. The second approach (Tube_1) was to sand the edges of the aluminum sheet in a scarf joint configuration. The resulting seam was acceptable, but the process was inconsistent. A subsequent tube, Tube 2, had a larger seam while Tube 1 and Tube 3 had seams which were very small.

The problem of the circumferential seam at the ends of the gauge section was addressed at the same time as the axial seam. The approach was to extend the aluminum sleeve into the tapered section of the specimen ends. For the first attempt the sleeve was extended by 12.7 mm so that 6.35 mm of the sleeve covered the taper (Tube 0). Slits were cut into the 6.35 mm extensions so that they would easily accommodate the taper by flaring out. This effort eliminated the end seam in the gauge section and resulted in a smooth transition; however, there was a considerable build up of resin at the interface of the flared sleeve and the rest of the composite tube end. For this reason the sleeve length was changed to 254 mm so that the entire taper was covered by the flared aluminum sleeve (Tube 1 to Tube 8). The result was a smooth transition from the gauge section to the larger outer diameter of the tube ends. This transition is shown in Figure 6.11. Both of the above methods were deemed acceptable for fabricating in-plane shear specimens. All tubes manufactured after Tube 0 and Tube 1 had short sleeves placed on the tube ends to produce the same smooth surface as exists in the gauge section. This bagging technique was very successful in terms of producing an excellent surface finish, very good consolidation and uniform thickness.

As mentioned previously, the fabrication of the scarf joint was a difficult technique and gave inconsistent results. It was therefore decided to replace the aluminum sleeve with a stiffer material so that the sheet thickness and resultant seam thickness could be reduced.

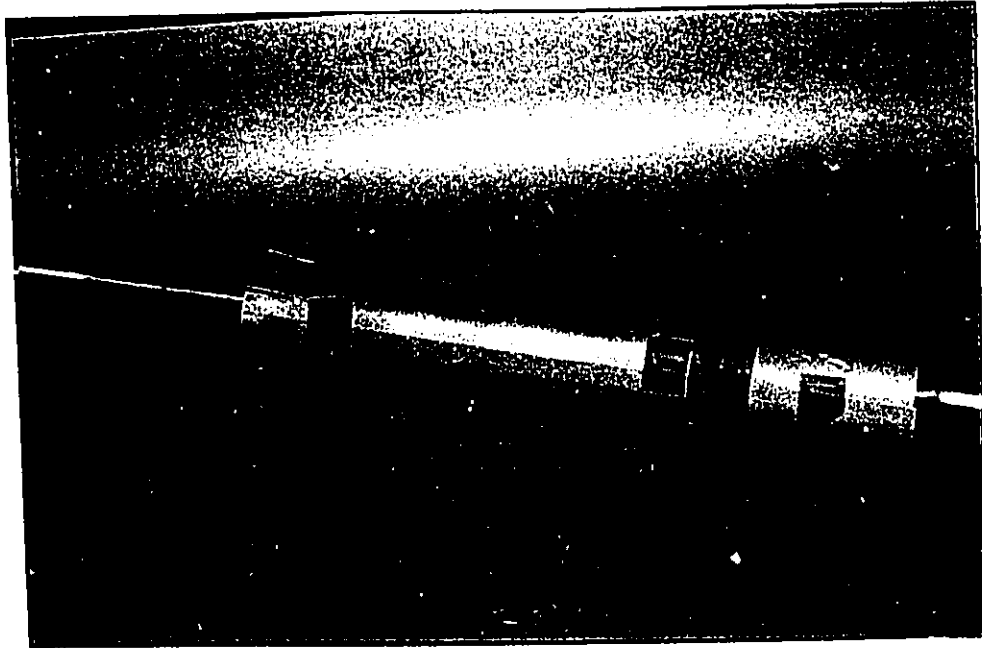


Figure 6.11 Aluminum sleeve with flared ends technique (Tube_2).

6.2.4 Steel Sleeve Bagging Technique

The material chosen to replace the aluminum sheet was cold rolled mild carbon steel shim stock which is available in 30 cm (12 inch) wide rolls. Both a 0.1 mm (0.004 inch) thick and 0.15 mm (0.006 inch) thick shim were evaluated. A 0.15 mm thick sleeve was attempted first because it was more rigid than the 0.1 mm stock. Neither of the two materials could be rolled into a sleeve so that they had to be wrapped around the tubes without being pre-rolled. This was particularly difficult in the case of the 0.15 mm thick sleeve as it tended to spring open. A completely wrapped tube can be seen in Figure 6.12. As in all previous cases using this method, a layer of shrink tape was hand wound under high tension over the sleeve. For this technique the tape was pre-heated using a blow dryer to ensure that the steel was tightly pressed against the composite. This was not necessary with the aluminum sleeve since it was deformed to fit the composite tube exactly. The axial seam in this case was excellent and this was confirmed by the C-scan results. There were however several internal voids detected by C-scan within other parts of the gauge section. When the sleeve was being removed it snapped open and resumed

its former flat shape which suggested that the sleeve may not have sufficiently compacted the tube enough during curing. Another tube was made using the more flexible 0.1 mm thick steel sleeve material, however, voids were again detected. Microscopic analysis showed that these voids were in fact between the individually wound layers confirming the belief that proper compaction was not being achieved.

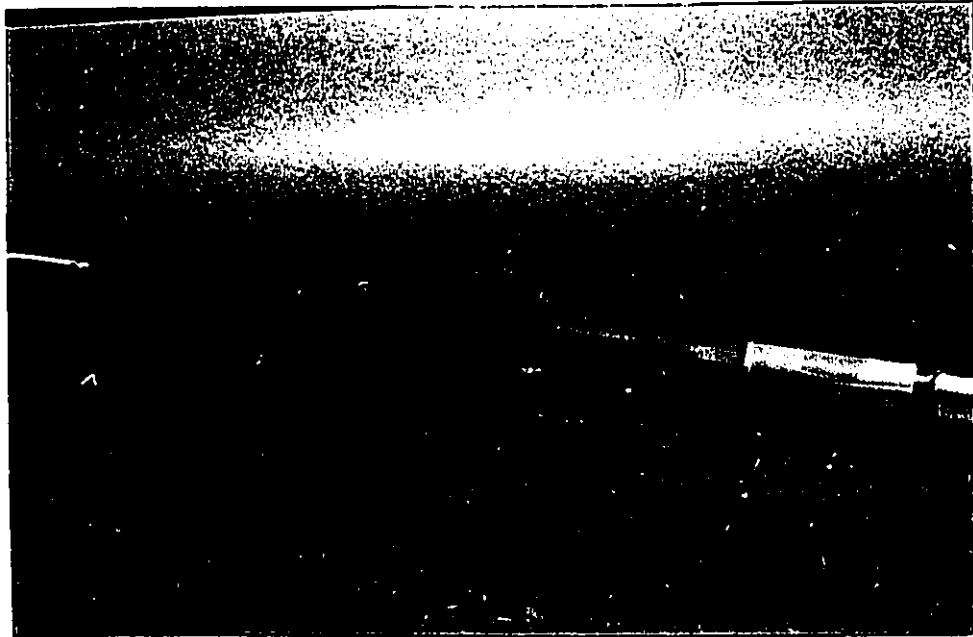


Figure 6.12 Mild steel sleeve wrapping technique

It is possible that the pre-preg was not consolidating prior to tube curing because of aging of the pre-preg. Since it was not desirable to increase the fibre tension to improve as-wound lamina consolidation, as this would result in higher residual stresses, it was therefore necessary to identify other methods which could be implemented during the filament winding phase of manufacturing to produce a highly compacted wound tube. Two changes were made to achieve this objective. First, the pre-preg tow was heated just prior to its application to the mandrel. The heat served to soften the resin enough to allow resin from the layer being wound and the one beneath on the mandrel to mingle somewhat. Furthermore, the heat increased the chance that moisture and volatiles which might cause interlaminar voids were liberated before curing. Secondly, a roller was used to apply a small amount of pressure to the heated tow filaments as

they were being wound onto the mandrel in a further attempt to guarantee good compaction between the laminae. This revised winding method was first utilized with tube 7.

Additional changes were also made for the curing of Tube 7 because of one further problem which was of an intermittent nature. Occasionally, a tube would have small surface pits (which were not detected by C-scan) after the cure due to resin starvation. With the latest two tubes this problem was especially serious and so steps were taken to eliminate it. Resin starvation also occurs during the curing of composite panels; the solution is to use a dam. In essence, this is a process of sealing the edges of the panel so that resin cannot flow into the surrounding bagging materials. Kevlar fibre wicks are used so that entrapped air can escape from the laminate through the dam to the vacuum. The same approach was used for this tube. A Kevlar fibre wick was placed axially on the outside of the shrink tape and extended until it reached the end of the mandrel. The release film which normally encloses both the tube and mandrel was cut so that only the tube was covered. The release film was then taped to the mandrel, making the dam, while ensuring that a small portion of the wick extended outside the release film. The completed tube is shown in Figure 6.13. The bagging procedure continued normally from this point on.

The resulting cured tube (Figure 6.14) was considerably better than all of the previous tubes in all respects. The seam was very thin and did not appear on the C-scan. Furthermore the C-scan result indicated that there were not any suspicious areas nor any surface irregularities. The resin starvation noticeable on several of the other tubes was not apparent in this case. A subsequent tube, Tube 8, was manufactured with this technique and the results were once again very good.

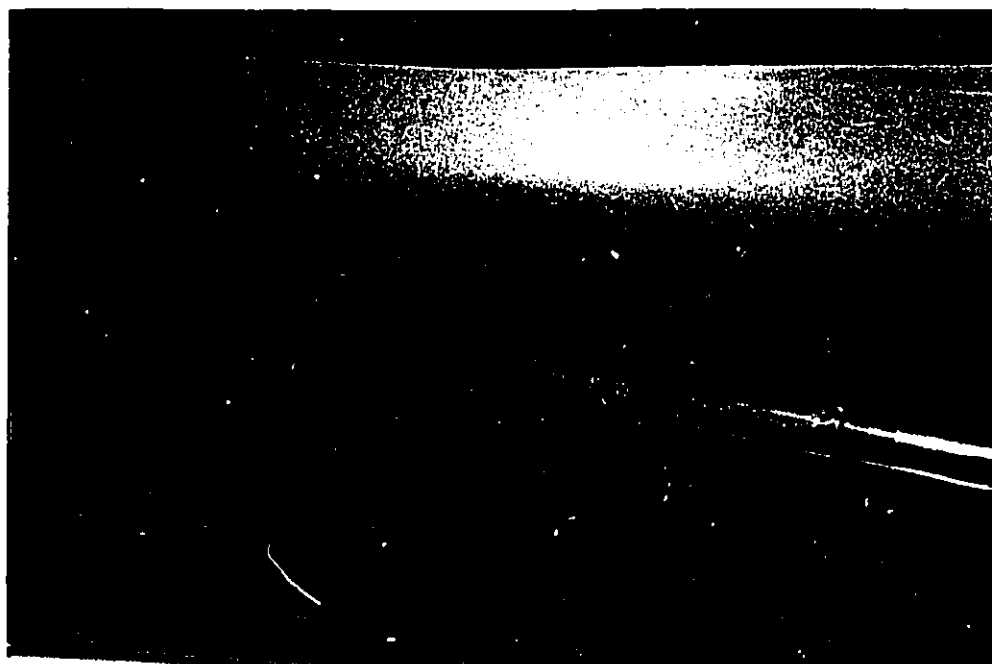


Figure 6.13 Tube with dam and Kevlar fibre wick

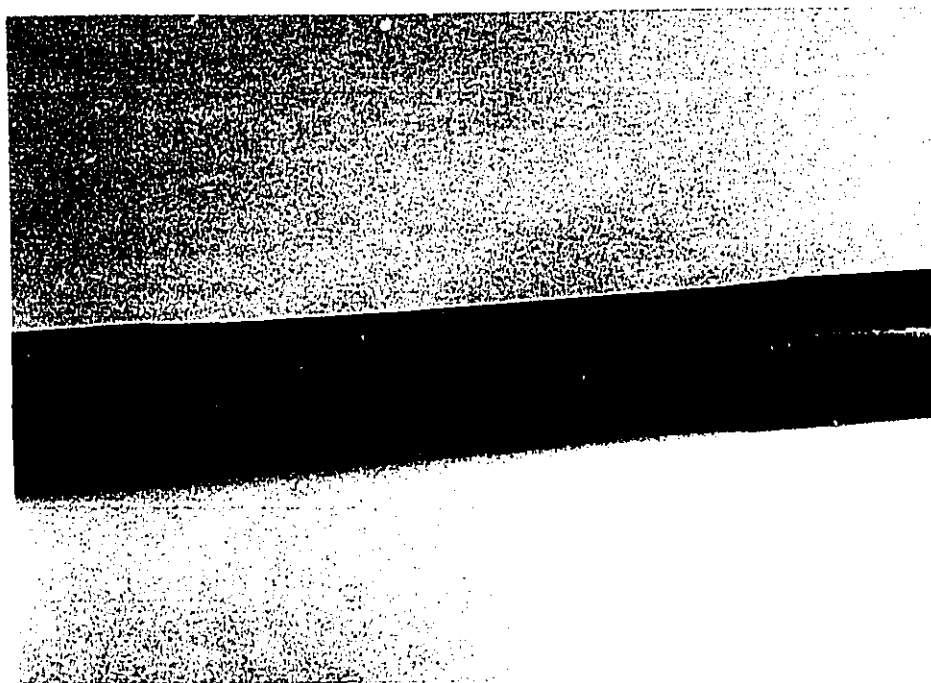


Figure 6.14 Tube 7 - produced using an advanced filament winding technique and damming.

6.3 Summary

A total of nineteen tubes were used in the development of a final manufacturing process. The general filament winding procedure is the same as that discussed in the previous chapter. Sections were taken from four of these, PR/R1 to PR/R4, to serve as residual stress specimens (Resid 1 to Resid 4) for the study of the previous chapter. Five more tubes, PR 1 to PR 4 and PR 7, served only to improve tube manufacturing with two others, PR 5 and PR 6, acting as both manufacturing tubes and as preliminary test specimens (see the following chapter). In addition, a section was cut from tube PR 7 to act as a residual stress specimen (Resid 5). Six tubes, Tube 0 to Tube 3, Tube 7 and Tube 8, were accepted as test specimens, by either meeting or exceeding the stated requirements. Three tubes, Tube 4 to Tube 6 met all dimensional requirements but ultrasonic C-scans of each indicated through thickness defects. A complete summary of the use of and manufacturing details for each tube is presented in tabular format in Appendix C.

A major accomplishment of the manufacturing study was the reduction of the tube seam Thickness. Tube four had an exceptionally thick seam (~ 0.38 mm) while the latest tubes (especially Tube 7 and Tube 8) had seams which were very thin (Figure 6.15). Six in-plane shear specimens were prepared from the 19 tubes that were manufactured: Tube 0 to Tube 3 (made using the aluminum sleeve) and Tube 7 to Tube 8 (made using the steel sleeve). In each case it was assumed that fibre consolidation was acceptable based on the microscopic analysis of other specimens manufactured using the same techniques. The geometric tolerances attained in the manufacturing of these specimens is the major indication of their acceptability.

The particular concerns for geometric tolerance were tube thickness, roundness, concentricity and straightness. The thicknesses of the resin rich seams for the aluminum and mild steel sleeve bagging techniques have only been discussed in a qualitative sense. It was of interest to measure thickness of the seam as well as the thickness of the tubes. Thickness measurements around the circumference of a tube typically varied by less than ± 0.08 mm. This corresponds to a $\pm 6.3\%$ thickness variation which is close to the desired variation of $\pm 5\%$. In

addition, the inner diameters of the tubes were very similar at 49.52 ± 0.08 mm as the same steel mandrel was used in each case. For each tube, the roundness tolerance was also ± 0.08 mm.

For the four tubes manufactured using the 0.8 mm thick aluminum sleeve (Tubes 0,1,2 and 3) and the single tube using the steel sleeve (Tube 7), the outside diameters were measured both at the seam and 90° rotation from the axial seam at three points along the gauge length (Table 6.1) before testing. Prior to the measurement of the outside diameters the seams were hand sanded using emery paper. If any black coloration appeared then the sanding was stopped.

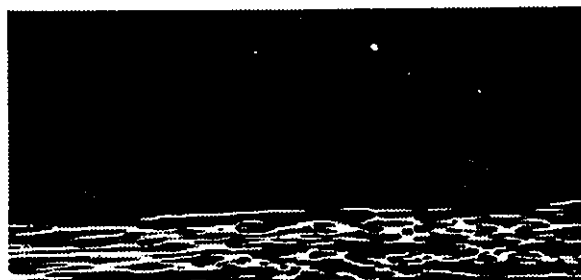
Table 6.1 Summary of Diameter Measurements and Differences in Seam Thicknesses

Specimen Number	Location	Tube Diameter (mm)			Average Diameter	% Diff in Thickness
		0	102	203		
Tube 0	Rot. 90°	52.20	52.17	52.17	52.18	
	Seam	52.22	52.40	52.32	52.32	5.2
Tube 1	Rot. 90°	52.15	52.17	52.20	52.17	
	Seam	52.35	52.45	52.32	52.38	7.7
Tube 2	Rot. 90°	52.32	52.35	52.35	52.34	
	Seam	52.58	52.45	52.38	52.48	5.0
Tube 3	Rot. 90°	52.17	52.17	52.15	52.16	
	Seam	52.50	52.48	52.25	52.41	9.3
Tube 7	Rot. 90°	52.30	52.27	52.25	52.27	
	Seam	52.30	52.27	52.20	52.26	-0.6

For all five tubes the outer diameter of the tube at 90° rotation away from the seam varied by less than ± 0.03 mm. The differences in thickness of the seams for the tubes that were fabricated using the aluminum sleeve bagging technique were substantially greater than for Tube 7 which employed the mild steel sleeve as part of the bagging technique. During the

testing phase of the project it would be seen whether the larger seam thicknesses for the tubes fabricated using the aluminum sleeve bagging technique would have any significant impact on the in-plane shear test results.

Additional measurements were made for Tube 7 in an attempt to further quantify the dimensional variation of tubes made using the final manufacturing technique. The tube ends were initially internally bored to remove any variation in roundness, fitted with internal plugs and placed between centres in a lathe for the measurements. Two parameters were of interest: tube concentricity and straightness. A dial indicator was placed on the tube surface at various locations along the tube length and the tube was then rotated slowly by hand. The dial indicator fluctuated by ± 0.08 mm (± 0.003 inches). This result was duplicated all along the tube. Thus the tube was considered concentric to within ± 0.08 mm. By running the indicator along the tube gauge section the straightness of the tube was found to be ± 0.08 mm (± 0.003 inches) on the seam and ± 0.025 mm (± 0.001 inches) away from the seam.



(a)



(b)

Figure 6.15 Comparison of (a) thick seam (PR 4) and (b) thin seam (Tube 7).

Chapter 7.0

In-plane Shear Testing Results

7.1 Introduction

Four series of tests were conducted. The first consisted of testing a solid aluminum cylinder to consolidate the testing procedures and to verify that shear modulus results found from torque testing compared favourably with known values. The second test schedule was for two tubular specimens considered to be preliminary in-plane shear specimens. The third group of specimens were the actual tubular in-plane shear specimens. The final group were flat coupon specimens for the $\pm 45^\circ$ tensile, 10° off-axis and 90° Iosipescu in-plane shear tests as well as four laminate tensile tests. The laminate configurations (taken from [13]) were $[\pm 55, \pm 50]_{4S}$, $[\pm 35, \pm 40]_{4S}$, $[\pm 65, \pm 50]_{4S}$ and $[\pm 25, \pm 40]_{4S}$.

7.2 Torsion Testing Equipment

Testing was conducted using the ATM triaxial hydraulic test machine located at the Institute for Aerospace Research structures laboratory at the National Research Council. Originally this equipment was designed for rotational and axial fatigue testing of small diameter thin-walled tubes. It was thus necessary to modify this machine so that it was capable of testing tubes in static torsion. In addition, the maximum capacity of the equipment was only 170 Nm which was approximately one third the required torque to fail the tubular in-plane shear specimens. To ensure that sufficient torque capability would be available, the A.T.M. equipment

torque capacity was increased by a factor of 10 to 1700 Nm using a torque multiplier. The details of the ATM upgrade are presented as Appendix D along with test procedures developed for use of the ATM at the increased capability. The upgraded equipment appears in Figure 7.1 with a schematic of the machine in Figure 7.2. The specimen is attached between the upper and lower universal joints. The lower motor support can be moved axially by an hydraulic actuator. It is possible to automatically maintain a zero axial load during torque loading using the axial actuator control. The load cell is used to monitor torque and axial load.

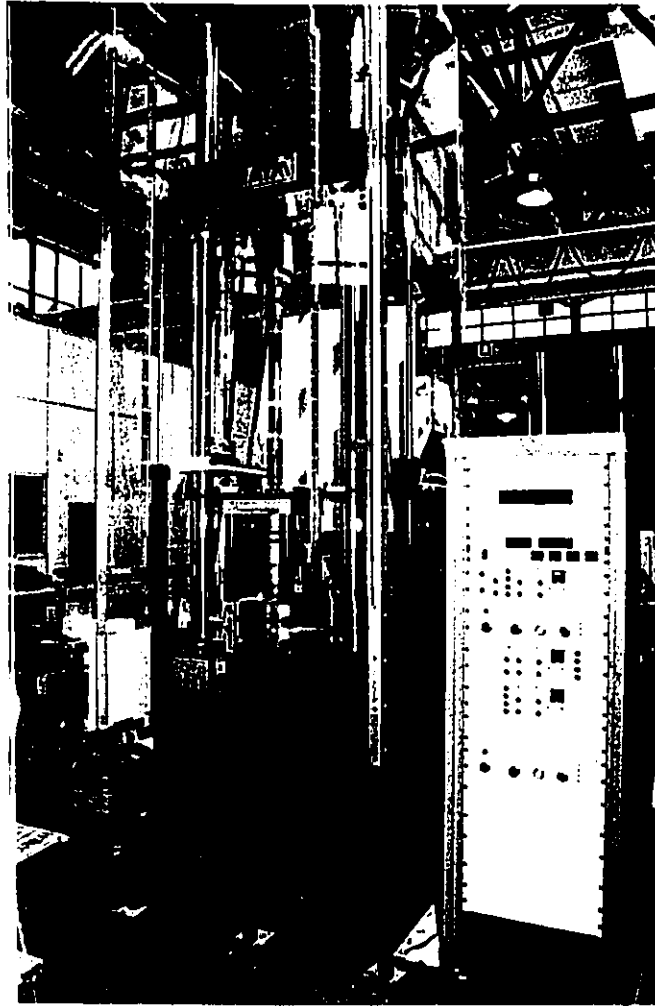


Figure 7.1 Upgraded ATM triaxial test machine.

The loading rate used for all of the testing was 1.13 Nm/min (corresponded approximately to a displacement rate of 0.1 mm/min on tube outer surface). All of the load and strain data was

collected using a PC based data acquisition system. The maximum capacity of the ATM load cell was 1695 Nm. All of the testing was in the range of 400 Nm - 1000 Nm. During testing special attention was paid to the magnitude of axial loading.

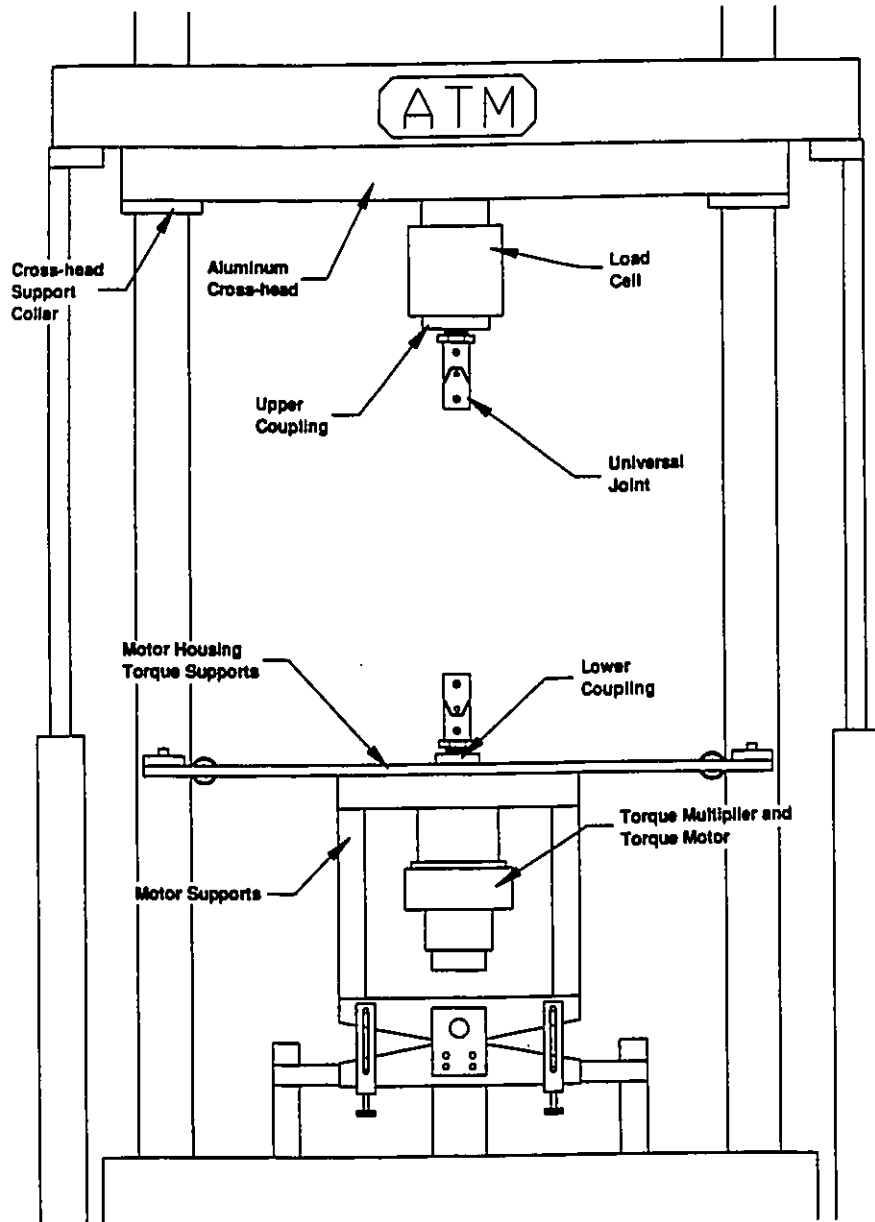


Figure 7.2 Schematic of ATM triaxial test machine.

7.3 Testing Of Aluminum Specimen

An aluminum rod (406 mm long and 38 mm diameter) was manufactured to provide an inexpensive means to check the ATM system and test procedures. In particular, it was necessary to check the load cell calibration using a specimen with known properties. The aluminum rod was loaded eight separate times in both the counter-clockwise and the clockwise directions up to a maximum load of 420 Nm (3700 in-lbs) which corresponds to shear strain of approximately 1430 $\mu\epsilon$. Two triaxial rosette strain gauges (Showa N32-FA-5-120-23) were used to measure the strain. Both gauges were attached in the circumferential direction at the centre of the rod length. The 45° gauge of the rosette was aligned with the tube axis to measure any axial strain while the 0° and 90° gauges were aligned at $\pm 45^\circ$ to the specimen axis to determine the shear strain. The shear strain was calculated using [12]:

$$\gamma_{12} = |\epsilon_{45^\circ}| + |\epsilon_{-45^\circ}| \quad (7.1)$$

while the shear stress was calculated using [48]

$$\tau_{12} = \frac{2T}{\pi r^3} \quad (7.2)$$

where γ_{12} is the shear strain, ϵ_{45° and ϵ_{-45° are the strain in the gauges at $\pm 45^\circ$ to the specimen axis, τ_{12} is the shear stress, T is the torque and r is the radius. A typical stress-strain curve for the aluminum specimen appears in Figure 7.3 (the remaining stress-strain curves appear in Appendix E). Table 7.1 displays the shear modulus determined from each of the eight trials. It is possible to apply a torsional load to the specimen in either the clockwise or counter-clockwise directions using the ATM machine. To ensure that there was no machine bias in either direction, two tests were conducted using the clockwise direction and six tests were conducted using the counter-clockwise direction. In each case the shear modulus is close to the reported value for aluminum of 26 GPa [48] indicating that the system is operating satisfactorily. In addition, no axial loading was observed during the testing. This was indicated by a zero strain reading which was measured for the strain gauges mounted along the tube axis. The ATM digital indicator was also used to display the axial load cell reading which was zero at all times during

testing of the aluminum specimen.

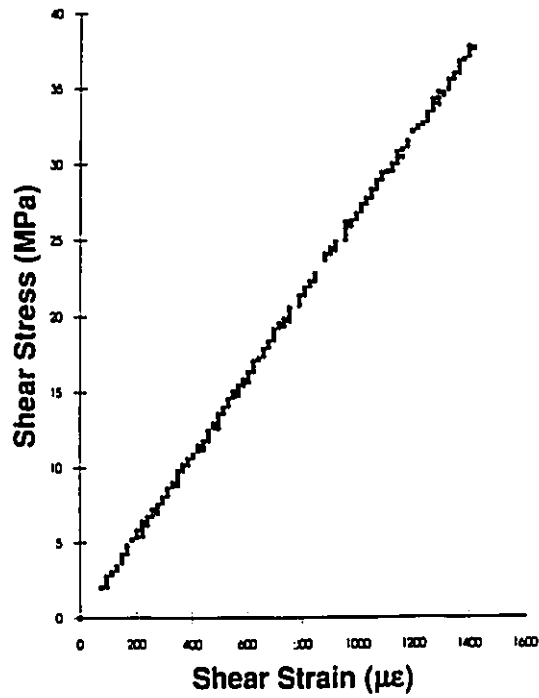


Figure 7.3 Typical shear stress-strain curve for the aluminum specimen loaded to 420 Nm.

Table 7.1 Summary Of Aluminum Specimen Testing Results

Trial Number	Torque Direction	Shear Modulus (GPa)
1	C-Clockwise	26.0
2	C-Clockwise	26.0
3	C-Clockwise	26.0
4	C-Clockwise	26.3
5	C-Clockwise	25.8
6	C-Clockwise	26.4
7	Clockwise	26.7
8	Clockwise	27.0
Average G_{12} (GPa)		26.3
Coefficient Of Variation (%)		0.01

7.4 Testing of Composite Tubes

7.4.1 Specimen Preparation

7.4.1.1 Attachment Of End Fittings

The procedure for attaching the tapered internal end fitting is similar to the process described in Section 4.5, however, it is described in detail again as there are some notable differences.

Prior to bonding the tube and end fittings the internal surfaces of both ends of the tube were machined. Firstly, the internal surface which comprises the joint (on either end) was machined to ensure that both ends of the tube were concentric with respect to the end fittings. This was done so that there would be a consistent adhesive thickness to ensure that the joint would not experience eccentric loading which may lead to premature failure. Secondly, a 12.7 mm long and 0.25 mm deep shoulder was machined into the tube ends to receive the corresponding shoulder on the end fitting. This was used to ensure proper alignment of the end fitting inside the tube.

After the tube was machined the surfaces of both the tube and the end fitting were prepared for bonding. Initially the tubes were sanded using 600 grit emery paper and then cleaned using acetone. The adhesive (Hysol 9309.3) was then prepared and applied to the inside of one end of the tube. The end fitting for that end was then sanded using 600 grit emery paper to remove aluminum oxides and then cleaned using acetone. Adhesive was then applied to the end fitting surface. The end fitting was then seated inside the previously prepared tube end. Since the adhesive was highly viscous, the tube was clamped vertically so that the adhesive would flow down into the joint and any excess would form a fillet at the joint tube interface. When the adhesive hardened (after 4 hours) the second joint was bonded using the same procedure. The adhesive of both ends was allowed to cure for three days before testing.

7.4.1.2 Attachment Of Strain Gauges

Strain gauges were used to measure the strains during testing. During the time that the joint was curing the strain gauges were attached to the specimen gauge section. Showa triaxial 120 Ω rosette strain gauges were used on all of the specimens. A rosette gauge was bonded at the centre of the gauge section at each of rotational positions of 90°, 180°, and 270° to the tube seam for all composite specimens. For the two preliminary specimens, PR 5 and PR 6, a rosette gauge was bonded at the joint-gauge section interface and another gauge was attached between the joint and the centre of the gauge section. These latter two gauges were used to check that no strain gradients exist at the centre of the gauge section. The individual gauges of each rosette were aligned at 0°, +45° and -45° to the tube axis.

The surface beneath each gauge was first abraded using 600 grit emery paper. The surfaces were then wiped with isopropyl alcohol. Each gauge was then located on the specimen using a microscope, bonded and then checked for orientation and position. A completed specimen appears in Figure 7.4.

7.4.2 Preliminary Composite Tube Testing

Composite tube specimens PR 5 and PR 6 (see manufacturing summary table, Appendix C) were tested in order that the procedures established and verified for the aluminum specimen could be evaluated for the AS4/3501-6 tubes. Several results of the aluminum specimen testing, while encouraging, would have to be verified for the composite tubes. In particular, since the composite tube has a non-isotropic structure, it is possible that the level of pure shear developed could be different from that of the isotropic aluminum. Secondly, the potential for axial loading is much greater in helically wound tubes. Furthermore, the end fittings, though proof tested in a simpler apparatus for the wet wound tubes, had not been used to fail an actual AS4/3501-6 tube of the dimensions selected for in-plane shear testing. Finally, these two specimens were used to again ensure that strain readings at the centre of the gauge section were free of effects from the end fittings. Tubes PR 5 and PR 6 were used for

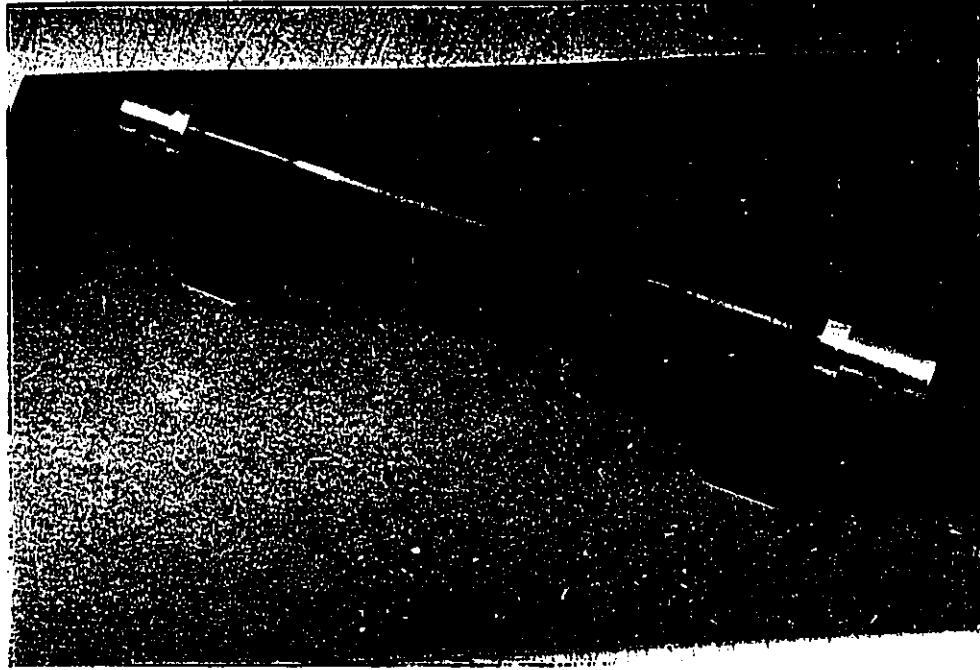


Figure 7.4 Prepared specimen ready for testing.

preliminary testing as each had thick axial and circumferential seams.

Five rosette strain gauges were mounted on each specimen. Three rosette strain gauges were mounted at 180° to the tube seam, one at the edge of the gauge length, one between the edge and the centre of the gauge length and one at the centre of the gauge length. The remaining two biaxial gauges were mounted at 90° and 270° to the axial seam at the centre of the gauge length. The gauges at 180° to the tube seam were used to check for end effects while the three gauges around the centre of the gauge length were used to verify that the shear strain was the same for all locations on the tube circumference. Equation 7.1 was used again to calculate the shear strain while the shear stress was calculated using equation 4.2 which is repeated below [48]:

$$\tau_{12} = \frac{2Tr_o}{\pi(r_o^4 - r_i^4)} \quad (7.3)$$

where r_o is the outer tube radius while r_i is the inner tube radius.

It is important to determine the level of pure shear developed in the specimen. This is indicated by the similarity between the absolute magnitudes of the two gauges (of the rosette) mounted at $\pm 45^\circ$ are to one another; pure shear occurs when the strain magnitudes measured by both gauges are identical. In order to measure the degree of pure shear developed during the testing, the normalized strains for the two gauges at $\pm 45^\circ$ to the tube axis of each rosette were plotted. The normalized strain is calculated by dividing the strain measured by the $\pm 45^\circ$ gauges by the average of the absolute magnitudes of the two gauges. A value of 1.0 represents pure shear. Figure 7.5 and 7.6 show the normalized strain readings for the gauges mounted around the circumference at the centre of the gauge section versus the shear stress for both specimens. For specimen PR 5, the gauge readings at 90° and 180° to the tube seam are within $\pm 0.05\%$ of 1.0, while the readings for the gauge at 270° are within $\pm 13\%$ of 1.0. For specimen PR 6 the gauge readings at 180° and 270° to the tube seam are within $\pm 0.05\%$ of 1.0 while the gauge readings at 90° are within $\pm 14\%$ of 1.0. The gauges which indicated a less pure shear state were in different locations in the two tubes which suggested that the orientation of the rosette gauges may not be at $\pm 45^\circ$ to the tube axis. After examination of the tubes, this was found to be the case. Thus, greater care was taken to avoid misalignment of the rosette gauges for the remaining tubes.

Shear stress-strain diagrams were plotted using the readings from three rosette gauges mounted along the tube length in order to determine whether or not a strain gradient existed. The diagrams for PR 6 appear in Figure 7.7. The shear strain as measured by the rosette gauges at the centre of the gauge section and at 50 mm from the centre are almost identical. The shear strain measured at the end of the gauge section is slightly higher than that of the other two for a given load. Based on these results it is reasonable to assume that end effects are not significant.

The stress-strain diagrams for PR 5 and PR 6 are shown in Figure 7.8. The readings from the rosette gauges mounted around the tube circumference and at the centre of the gauge section were used to measure the strain. Since one rosette gauge was mounted improperly in each case,

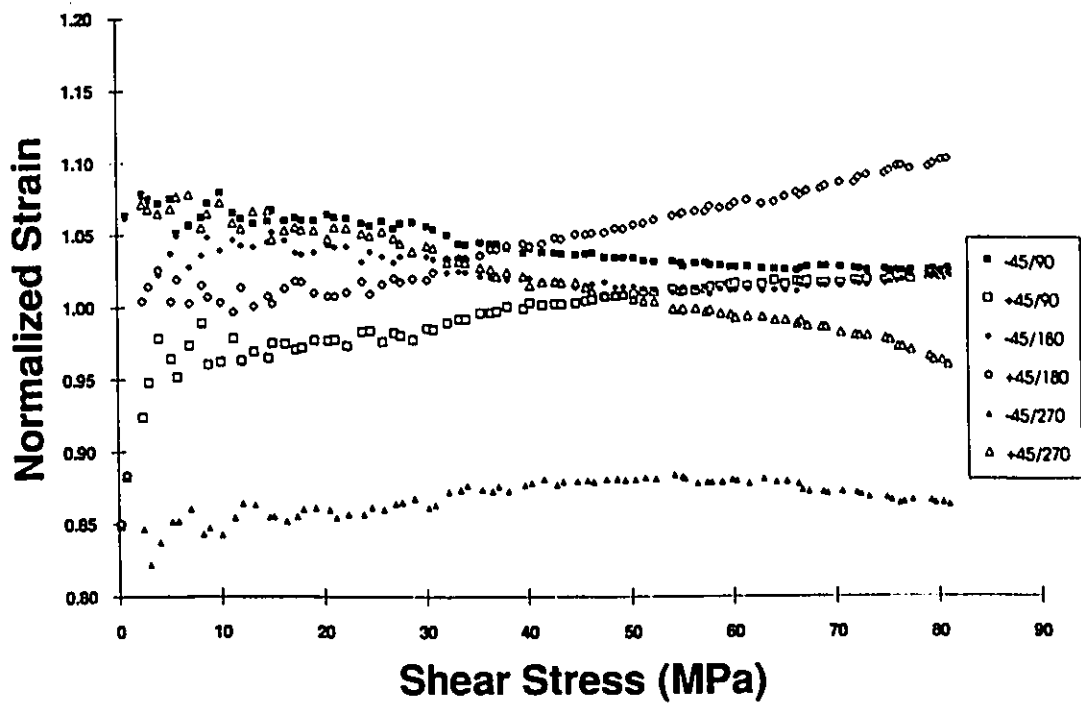


Figure 7.5 Normalized strain curves for specimen PR 5.

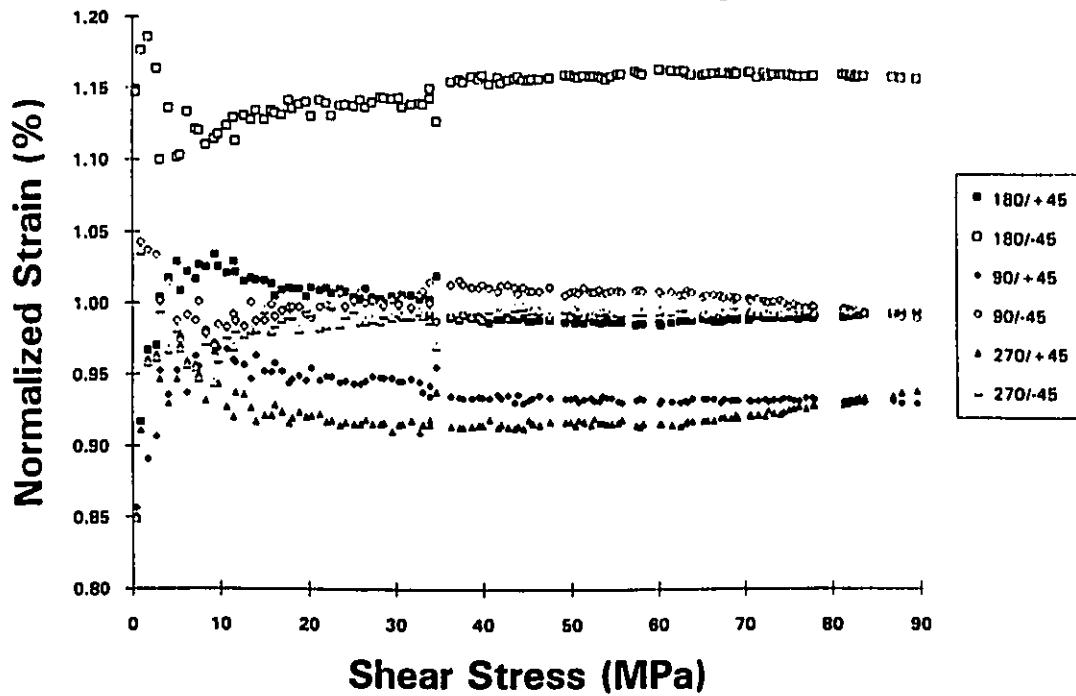


Figure 7.6 Normalized strain curves for specimen PR 6.

the strain readings are the average of the strains measured by the other two rosette gauges. The curves for the two specimens are very similar thus indicating good repeatability.

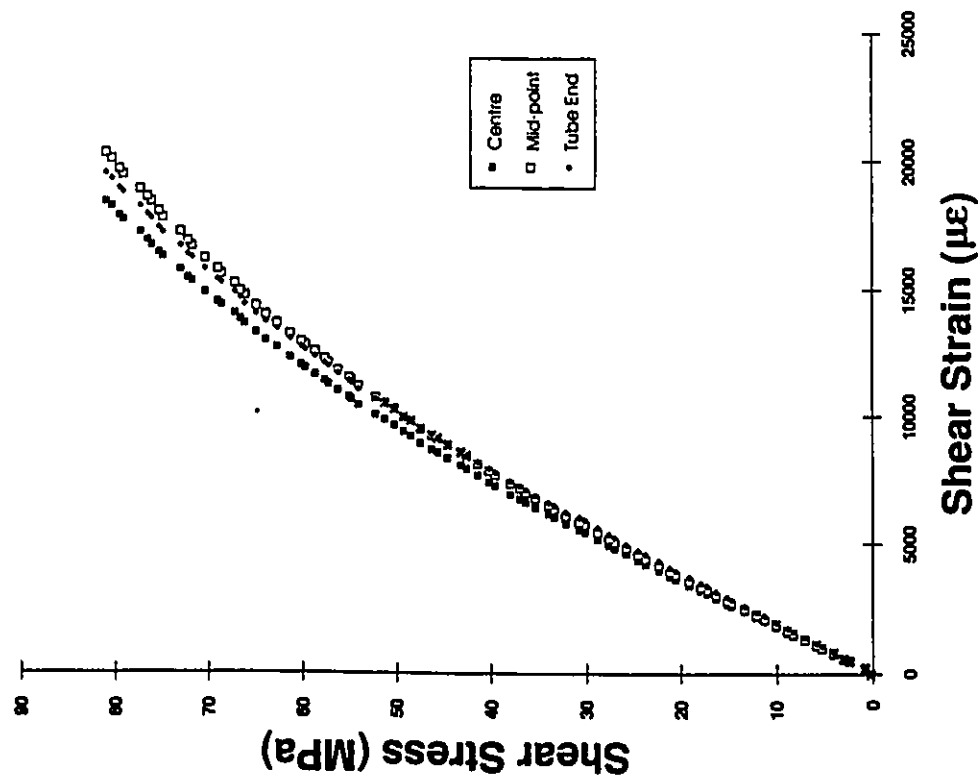


Figure 7.7 Shear stress strain diagrams for individual gauges of PR 6.

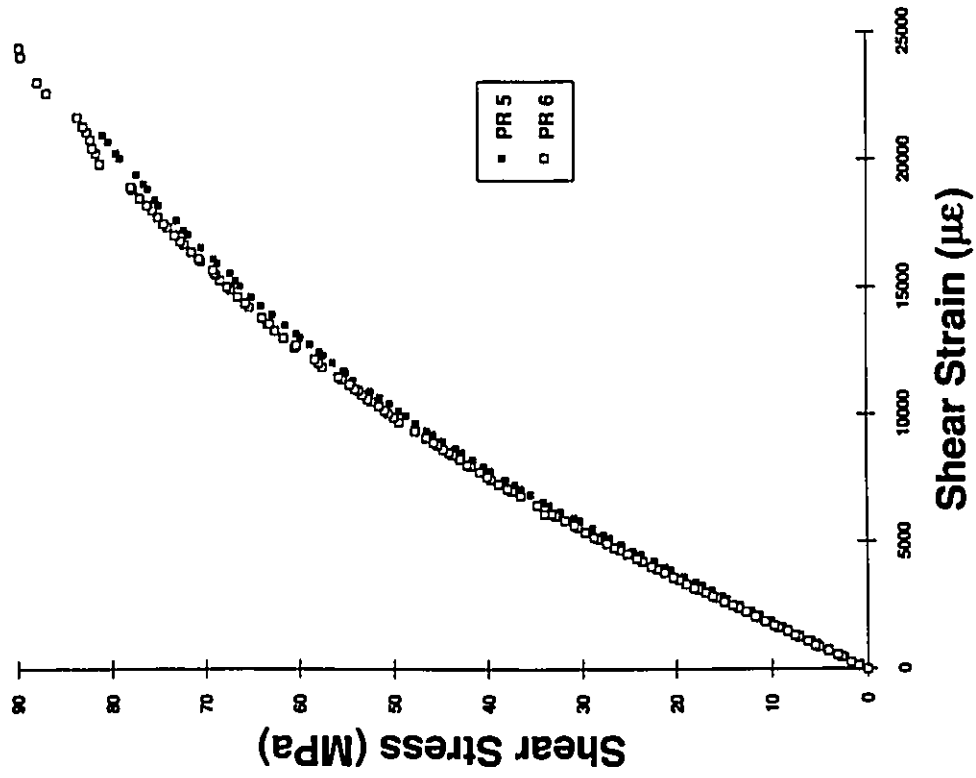


Figure 7.8 Shear stress-strain curves for PR 5 and PR 6.

The shear modulus values for both tubes were calculated by first fitting the stress-strain data of Figure 7.8 with a fourth order least square polynomial of the form:

$$\tau_{12} = C_1 + C_2\gamma_{12} + C_3\gamma_{12}^2 + C_4\gamma_{12}^3 + C_5\gamma_{12}^4 \quad (7.4)$$

where C_n are constants, τ_{12} is the in-plane shear stress and γ_{12} is the in-plane shear strain. The coefficient of correlation for both curves was above 0.99. In the aerospace industry secant modulus values are commonly measured at 5000 $\mu\epsilon$. Table 7.2 summarizes the shear modulus and ultimate shear strength values for both PR 5 and PR 6.

Table 7.2 Summary Of Preliminary Tube Test Results

Specimen Number	Shear Modulus (GPa)	Ultimate Shear Strength (MPa)
PR 5	5.4	81.9
PR 6	5.4	90.0

7.4.3 Tubular In-plane Shear Specimen Testing

7.4.3.1 Introduction

Six specimens were used to determine the in-plane shear modulus and strength of AS4/3501-6. End fittings and strain gauges were attached to the specimen in the same fashion as described previously. Three rosette strain gauges were mounted around the circumference of each tube at the centre of the gauge section to measure the shear strain. The loading rate remained unchanged.

Five specimens were tested using a positive torque while the last tube was tested using negative torque (negative and positive loading were described in Section 5.3). There are several reasons which suggest that it is necessary to verify that torque direction does not affect the shear test results. Marlowe et. al. [32] and Ellis and Hoskins [31] reported a difference in buckling

load levels in tubular laminates when the direction of torque was reversed. A large factor of safety was used in the design of the specimens for this reason; however, experimental verification of the design was still desirable. Since the composite tube has an 88° helical wind which is not a true 90° (hoop) unidirectional laminate, the loading will not be a 100% in-plane shear loading. Theoretically, at least, the difference should be negligible. Finally, it has been shown using the Azzi-Tsai failure theory (Section 5.3) that changing the torque direction would not affect the failure level. Testing of one specimen would not confirm the above assumptions, however, it could give some indication if there are any significant differences.

7.4.3.2 Test Results

Normalized shear strain curves were plotted for each of the six specimens tested. Figure 7.9 presents the normalized strain curves for all three gauges mounted on Tube 7. The solid symbols represent the readings from the tensile arm of each rosette gauge and the open symbols correspond to the compressive arm readings. The majority of the normalized stress-strain values for Tube 7 are within $\pm 5\%$ of 1.0 (pure shear). The normalized strain curves for the other four specimens (Tube 0 - Tube 3) are within $\pm 10\%$ of 1.0 (Appendix E).

The normalized strain values for Tube 8 (Figure 7.10) are significantly different from the values for those specimens tested using positive torque. Even though the values are within $\pm 10\%$ of 1.0, the tensile and compressive readings are very tightly grouped together throughout the entire test.

The in-plane shear modulus was calculated in the same manner as for tubes PR 5 and PR 6. The shear strain was the average of the shear strain found for each of the three rosette gauges. As before, the coefficient of correlation in each case was above 0.99. Table 7.3 summarizes the shear modulus and shear strength results for Tube 0-Tube 3, Tube 7 and Tube 8.

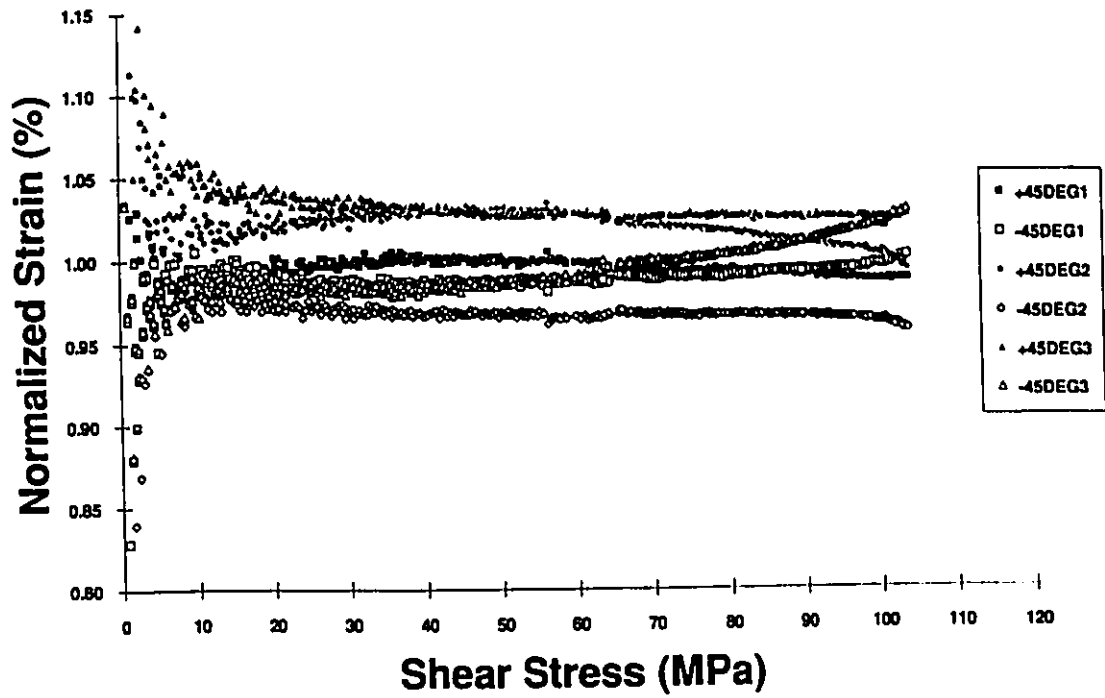


Figure 7.9 Normalized strain curves for Tube 7.

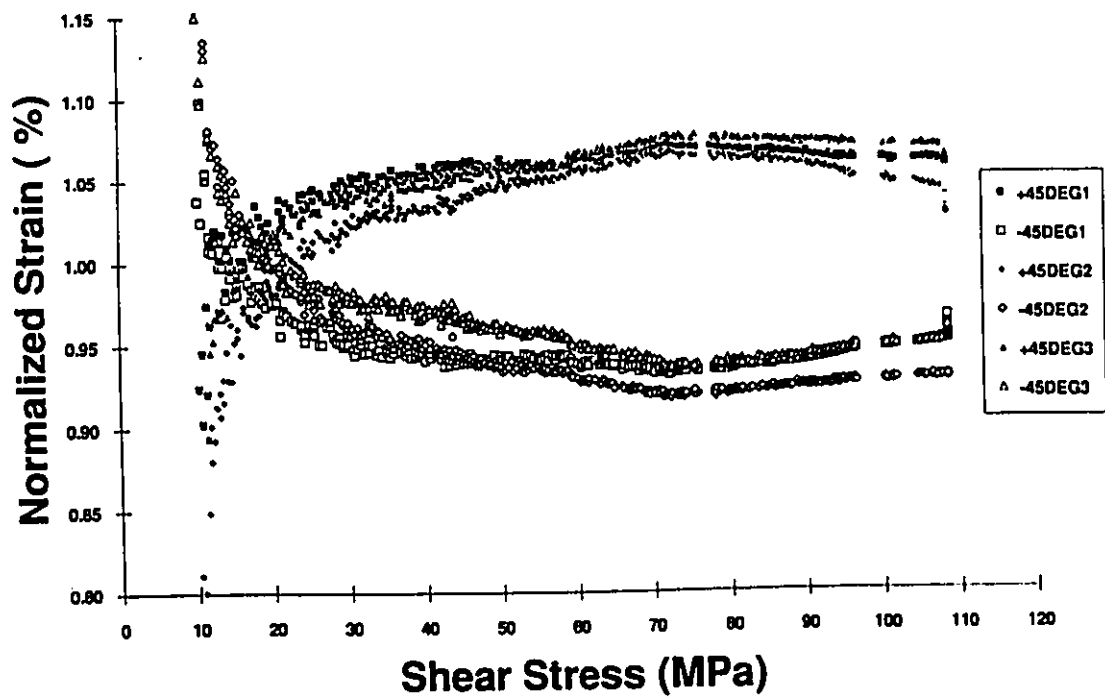


Figure 7.10 Normalized strain curves for Tube 8.

Table 7.3 Summary Of Test Results For Tubular In-plane Shear Testing

Specimen Number	Shear Modulus (GPa)	Ultimate Shear Strength (MPa)
Tube 0	5.7	108.9
Tube 1	5.6	107.2
Tube 2	5.7	110.4
Tube 3	5.3	92.6
Tube 7	5.2	103.7
Tube 8	5.4	99.71
Average	5.5	103.8
Coefficient Of Variation (%)	3.5	5.8

7.5 Coupon Specimens

The specimen dimensions for the other three in-plane shear tests and the two laminates specimens were determined in [13] according to various criteria. This information will not be repeated here, but the specimen drawings are in Appendix F. The specimens for the three in-plane shear test methods were fabricated previously [13].

End tabs were bonded to the 10° off-axis specimens using a two part adhesive (Hysol 9056) cured at room temperature. The end tab dimensions were consistent with ASTM D-3039. Experience in the laboratory, however, has shown that the use of bonded tabs can result in damage directly under the end-tabs for the $\pm 45^\circ$ tensile and laminate specimens. A second method employing Norton 320 Screen-Bak screen placed between the gripping surfaces of the specimen and aluminum plates in contact with the tensile testing machine grips was used for these specimens (Figure 7.11).

A ninety degree biaxial strain gauge (Showa N22-FA-5-120-11) was bonded to one face

of the test section of each of the $\pm 45^\circ$ tensile specimens. Three axis rosette ($0^\circ/45^\circ/90^\circ$) strain gauges (Showa N32-FA-5-120-23) were bonded to one face of the 10° off-axis specimens and the laminate specimens. In addition, the first four of the laminate specimens had the same rosette strain gauges bonded back-to-back to monitor bending. Since no bending was observed during testing of these four laminate specimens the remaining specimens had strain gauges bonded on one side only. Two 90° biaxial strain gauges (Micro-Measurements CEA-06-062WT-120) were bonded back-to-back on all the 90° Iosipescu specimens as well. Each 90° Iosipescu specimen was checked to ensure that both back-to-back biaxial gauges were located with respect to one another to within ± 0.05 mm and that each gauge was rotated with respect to the other by no more than $\pm 0.5^\circ$. In the case of the Iosipescu specimens, however, the readings from the four strain gauges were also used to indicate whether or not a region of pure shear existed in the gauge region.

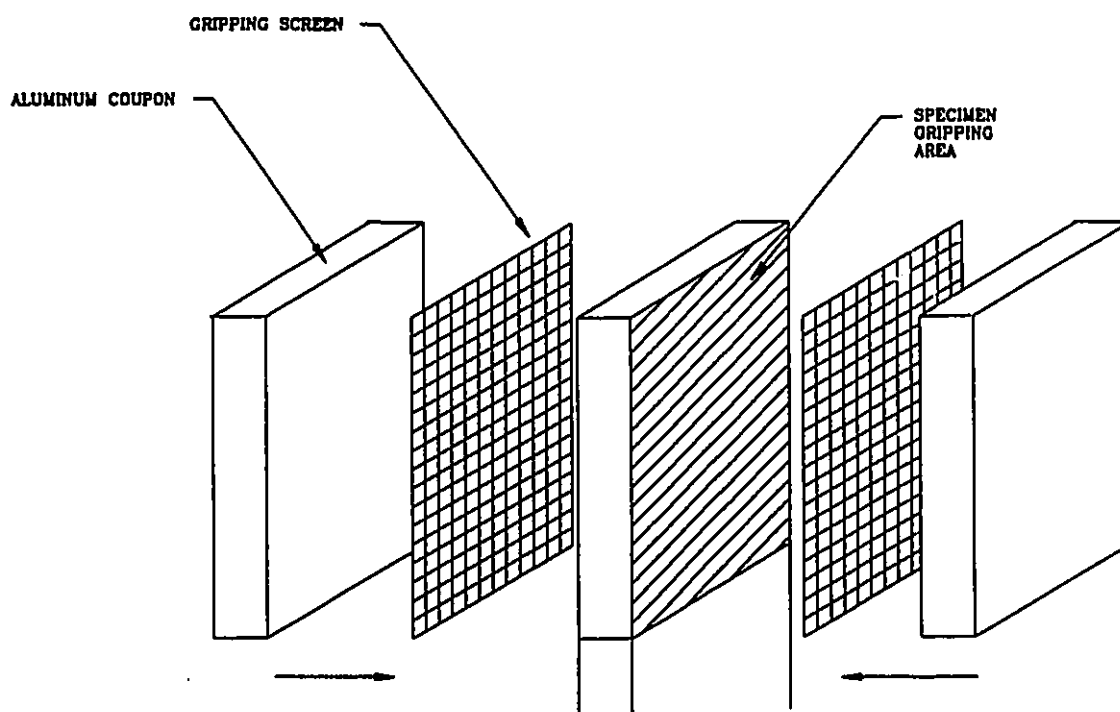


Figure 7.11 Tabbing method used during testing of $\pm 45^\circ$ tensile and laminate specimens.

7.5.1 Testing

Twelve Iosipescu, ten 10° off-axis, nine ±45° tensile, five [±55,±50]_{3S}, eight [±35,±40]_{3S}, five [±65,±50]_{4S} and five [±25,±40]_{4S} specimens were prepared. All specimens were tested using an Instron Universal Testing Machine (Model 1125) equipped with a 100kN load cell at a loading rate of 0.1mm/min. The strain gauge and load cell data were collected using the same Sciometric data acquisition system and 386 PC as was used in the tubular specimen testing.

Previous testing using the original version of the Iosipescu test specimen and fixture [5] had shown that the method was susceptible to in and out of plane bending. Therefore, during this testing the agreement between the four Iosipescu strain gauges was monitored to evaluate the shear state. At approximately 2500 µε the test was halted momentarily to compare the readings. If the four strain gauge readings were not within 10% of the mean value then the specimen was unloaded and realigned in the Iosipescu test fixture. Aligning the specimens to meet this criterion was extremely difficult. In fact, only six of the 90° Iosipescu specimens passed this initial screening of the specimen data.

7.5.2 Results

The expressions used to calculate the shear stress and strain from the test data are well established. For the ±45° tensile specimens the shear stress and strain were determined using [12]:

$$\tau_{12} = \frac{\sigma_x}{2} \quad (7.5)$$

$$\gamma_{12} = \epsilon_x - \epsilon_y \quad (7.6)$$

where τ_{12} is the shear stress, σ_x is the tensile stress, γ_{12} is the shear strain, ϵ_x is the load axis

strain gauge reading, and ϵ_y is the strain gauge reading at 90° to the load axis. The expressions used to calculate shear stress and shear strain for the 10° off-axis specimen are [12]:

$$\tau_{12} = 0.171\sigma_x \quad (7.7)$$

$$\gamma_{12} = -1.282\epsilon_{0^\circ} + 1.879\epsilon_{45^\circ} - 0.598\epsilon_{90^\circ} \quad (7.8)$$

where ϵ_{0° , ϵ_{45° and ϵ_{90° are the strain readings from the rosette gauge where 0° denotes the load direction. Simple relations were also used to calculate the shear stress and shear strain of the Iosipescu specimen [12]:

$$\tau_{12} = \frac{P}{wt} \quad (7.9)$$

$$\gamma_{12} = |\epsilon_1| + |\epsilon_2| \quad (7.10)$$

where P is the compressive load applied to the Iosipescu fixture, w is the width between the specimen notches, t is the specimen thickness, and ϵ_1 and ϵ_2 are the strain readings in the gauges at $\pm 45^\circ$ to the load direction.

The laminate specimen strains were measured using one biaxial strain gauge. Although only the gauge in the direction of the load was of interest as it was used to measure the tensile strain. The laminate tensile stress was determined using:

$$\sigma_{xx} = \frac{P}{wt} \quad (7.11)$$

where P is the tensile load, w is the specimen width and t is the specimen thickness.

As with the tubular specimen testing the shear stress-strain response of each specimen was nonlinear. For this reason a fourth order least square polynomial curve was fitted to the data for each specimen. The correlation coefficient in each case was over 0.99 indicating excellent agreement between the curve fit and the shear stress-strain data. The laminate tensile moduli were evaluated at 5000 $\mu\epsilon$. The moduli and shear strengths as well as the averages and coefficients of variation for each specimen are summarized in Table 7.4. The in-plane shear moduli obtained from the other test methods exhibit large coefficients of variation compared to the tubes (8.8%-15.4% versus 3.5%) The lowest coefficient of variation was for the 10° off-axis shear strength (4.9%); however, the 90° Iosipescu failure coefficient of variation was 20.7%. The coefficient of variation for the tubular specimens was again quite low (5.5%).

It is of particular interest that the results of only four $\pm 45^\circ$ tensile specimens and four 90° Iosipescu specimens are presented in Table 7.4. Normalized strain curves were prepared for the results for all $\pm 45^\circ$ tensile and 90° Iosipescu specimens (Appendix E). As has been discussed an attempt was made to ensure that the strain readings for all four gauges of each 90° Iosipescu specimen agreed closely. This screening, however, only covered the testing up to 2500 $\mu\epsilon$. Normalized strain curves were produced for each specimen to determine the shear strain state of the specimens over the entire testing range. Using the criterion from section 7.5.1 (i.e. $\pm 10\%$ variation between readings), only four specimens of the six which passed the original screening passed this criterion for the entire loading range. When the criteria of $\pm 10\%$ maximum variation in normalized strain was applied to the $\pm 45^\circ$ specimens only four test results were acceptable.

Table 7.4 Summary Of Results For Three Coupon In-plane Shear Tests

Specimen Number	Shear Modulus, G_{12} (GPa)	Shear Strength (MPa)
$\pm 45^\circ$ Tensile		
Shear1-2	4.5	57.5
Shear1-4	6.7	83.3
Shear1-8	6.6	114.8
Shear1-9	6.7	104.6
Average	6.1	90.0
Coef. of Variation (%)	15.4	24.4
10° Off-Axis Tensile		
Shear2-0	6.4	78.9
Shear2-1	6.1	75.9
Shear2-2	6.3	75.2
Shear2-3	6.3	72.7
Shear2-4	4.8	79.1
Shear2-5	6.8	69.0
Shear2-6	6.3	78.4
Shear2-7	5.5	82.7
Shear2-8	5.8	73.4
Shear2-9	6.2	74.9
Average	6.0	76.0
Coef. of Variation (%)	8.8	4.9
90° Iosipescu		
Shear3-0	3.9	45.0
Shear3-2	4.4	30.1
Shear3-9	4.6	55.6
Shear3-10	3.9	45.3
Average	4.2	44.0
Coef. of Variation (%)	13.0	20.7

The testing results for the four laminate configurations are in Table 7.5. As expected the coefficients of variation for the laminate tensile moduli were quite low (5.0%, 6.0%, 2.4%, 1.8%). The tensile moduli will be used to evaluate the performance of the four shear tests in Section 8.4.

Table 7.5 Summary Of Results For Four Laminate Tests

Specimen Number	Tensile Modulus, E_{xx} (GPa)
Laminate Configuration $[\pm 35, \pm 40]_{3S}$	
Lam1-1	26.3
Lam1-2	26.7
Lam1-3	28.7
Lam1-4	28.8
Lam1-5	25.5
Lam1-6	25.8
Lam1-7	26.8
Lam1-8	24.9
Average	26.7
Coefficient of Variation (%)	5.0
Laminate Configuration $[\pm 55, \pm 50]_{3S}$	
Lam2-1	14.0
Lam2-2	13.4
Lam2-3	13.0
Lam2-4	15.3
Average	13.9
Coefficient Of Variation (%)	6.0
Laminate Configuration $[\pm 65, \pm 50]_{4S}$	
Lam3-1	14.1
Lam3-2	14.3
Lam3-3	14.3

Lam3-4	13.6
Lam3-5	13.5
Average	14.0
Coefficient Of Variation (%)	2.4
Laminate Configuration $[\pm 25, \pm 40]_{45}$	
Lam4-1	45.4
Lam4-2	43.0
Lam4-3	44.2
Lam4-4	44.6
Lam4-5	44.8
Average	44.4
Coefficient Of Variation (%)	1.8

7.6 Fibre Volume Fraction Testing

Composite strengths and stiffnesses are functions of fibre volume fraction, thus, it is necessary to determine the fibre volume fraction of the specimens that were tested. In Section 8.4 the fibre volume fraction values will be used to adjust the in-plane shear moduli found using the four in-plane shear tests to represent values associated with the volume fractions of the laminates.

A resin burn-off method [64] has been developed at IAR for the determination of the fibre volume fraction of carbon fibre reinforced epoxy composites. The method consists of cutting small pieces from the composite specimens and then burning them in an oven with an inert atmosphere (carbon fibres oxidize at high temperature). The weight difference of the samples before and after burn-off is recorded and the fibre volume fraction calculated. Table 7.6 summarizes the fibre volume fraction values for the four in-plane shear and four laminate specimen groups.

Table 7.6 Summary of Fibre Volume Fractions

Specimen Type	Fibre Volume Fraction (%)
10° Off-axis Tensile	59.5
±45° Tensile	61.4
90° Iosipescu	62.7
Shear Tubes	56.2
[±55,±50] _{4S} Laminate	62.2
[±35,±40] _{4S} Laminate	56.4
[±65,±50] _{4S} Laminate	61.8
[±25,±40] _{4S} Laminate	59.2

7.7 Summary

The results and observations were presented for preliminary torsion testing, tubular in-plane shear testing, coupon shear testing and laminate testing. The success of the tubular in-plane shear specimen for measuring in-plane shear properties is evaluated in Sections 8.2-8.4.

Chapter 8.0

Evaluation Of Tubular In-Plane Shear Specimens

8.1 Introduction

This study has been undertaken primarily to compare the 90° Iosipescu, $\pm 45^\circ$ tensile, 10° off-axis and tubular in-plane shear tests as to their appropriateness for determining in-plane shear properties. The four tests are compared in the following sections on the basis of their ability to produce a pure shear state, the consistency of the results, and whether the test results can be used to accurately predict the tensile moduli of shear sensitive laminates.

8.2 Tubular In-plane Shear Testing

For both the shear modulus and shear strength of the six in-plane shear specimens the coefficients of variation are very low which demonstrates the repeatability and consistency of the manufacturing and testing procedures. The repeatability is demonstrated in Figure 8.1 which shows the stress-strain curves of all six tubular specimens; it is difficult to distinguish one curve from the other. In addition, the results show that any small differences that might exist within the specimens due to the variations in manufacturing (aluminum sleeve vs. steel sleeve) do not affect the test results. It is also important to note that Tube 8 which was tested using negative torque has shear modulus and shear strength values which are very similar to those of the specimens tested using positive torque.

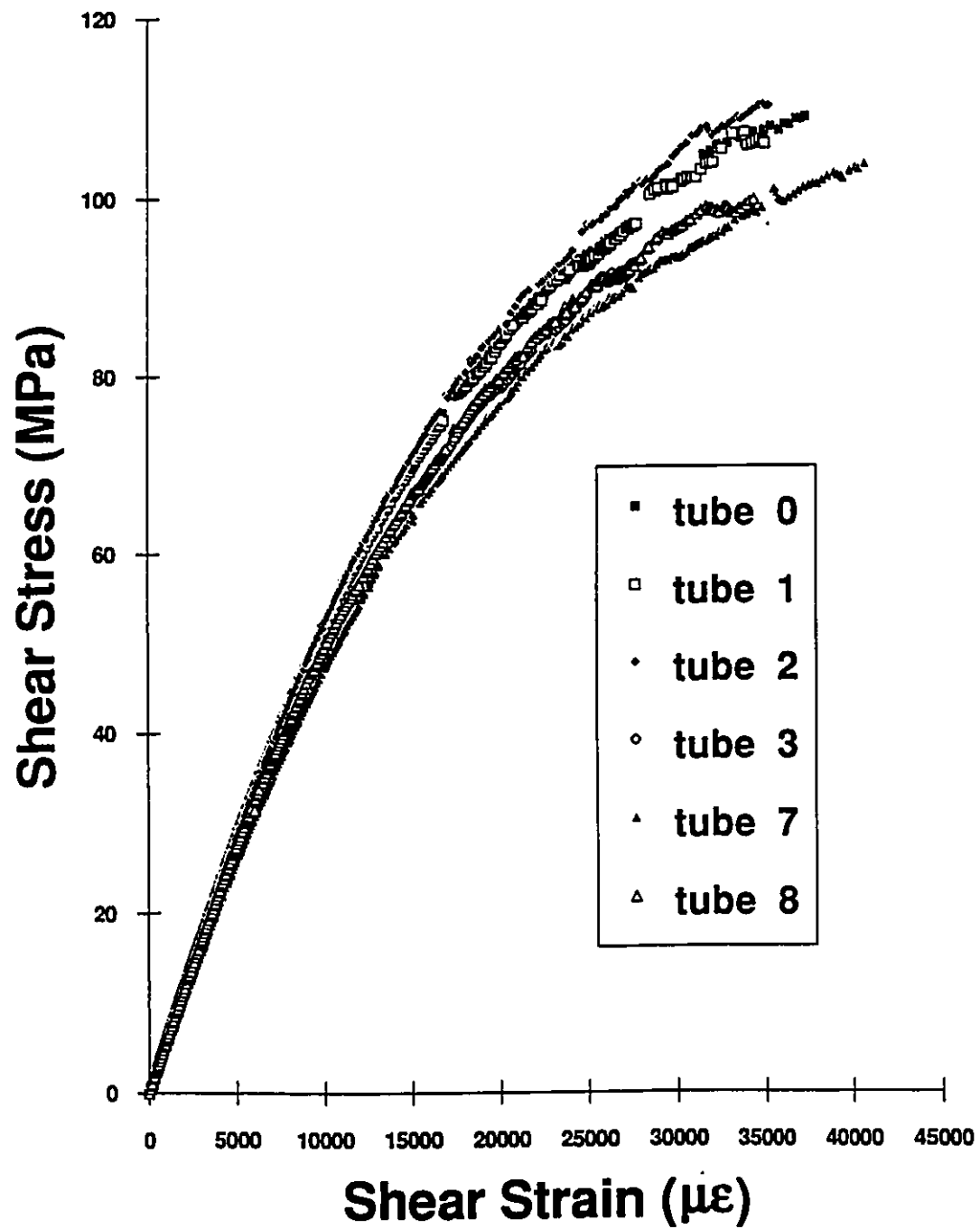


Figure 8.1 Stress-strain curves for the six tubular in-plane shear specimens.

Another important parameter which was examined was the degree to which the shear strains (measured by the three rosette gauges mounted on each in-plane shear specimen) corresponded to one another. Figure 8.2 shows the shear stress-strain response for the three gauges of Tube 7. The three curves are very similar, thus, indicating that there is no change in strain response around the specimen circumference. The shear strains measured by the three gauges of the other five specimens also correspond closely to one another which suggests that only one strain gauge is needed in future experimental studies using tubular in-plane shear specimens.

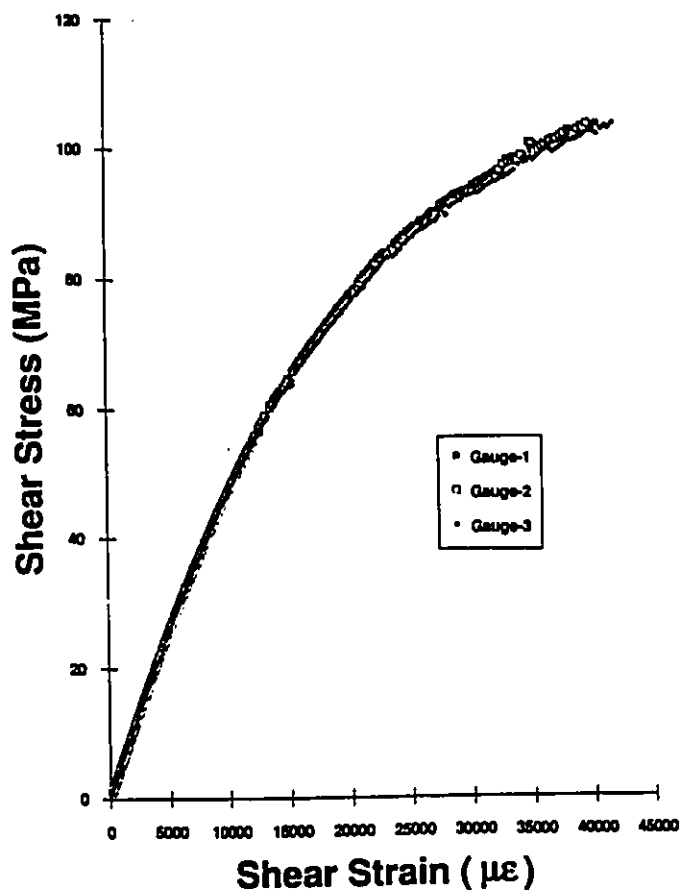


Figure 8.2 Shear stress-strain curves for three circumferential gauges of Tube 7.

The two preliminary tubes, PR 5 and PR 6, were not used for in-plane shear testing because both had thick axial and circumferential seams as a result of the vacuum bagging

technique used during the development stage of the manufacturing process. It was thought that these seams might cause the tubes to fail prematurely. The average shear strength of these two tubes was 86.0 MPa (Table 7.2) whereas the average shear strength of the six in-plane shear specimens was 103.8 MPa (Table 7.3). The two preliminary tubes on average had shear strengths that were 17% lower than the tubes with thin axial (and no circumferential seams). This result highlights the necessity for reducing the axial seam thickness.

Figure 8.3 and Figure 8.4 show typical failed specimens for positive and negative torque loading, respectively. It is worth noting that the two failure geometries were quite different. The positive torque failure demonstrates the tightening of the helix which has coiled up between the two broken ends of the tube. Alternatively, the helix in Figure 8.4 appears to have uncoiled. This corroborates the predictions for positive and negative torque loading discussed in Section 5.3.

8.3 Analysis Of Test Results for Four In-plane Shear Test Methods

8.3.1 In-plane Shear Results

Table 8.1 summarizes the average shear modulus values, average shear strength values and the related coefficients of variation for each of the four tests. The major difference in the results is the high coefficients of variation for the modulus values of the three coupon specimens (8.8%-15.4%) compared to 3.5% for the tubular specimens. The failure values for the four tests are very different with the tubular shear test specimens having the highest strength and the Iosipescu specimens the lowest. The 90° Iosipescu and $\pm 45^\circ$ tensile specimens have high coefficients of variation (20.7%, 24.4% respectively) but the 10° off-axis and tubular shear specimens have relatively low coefficients of variation (8.8%, 5.7% respectively). The modulus values do not vary as much as the strength values.

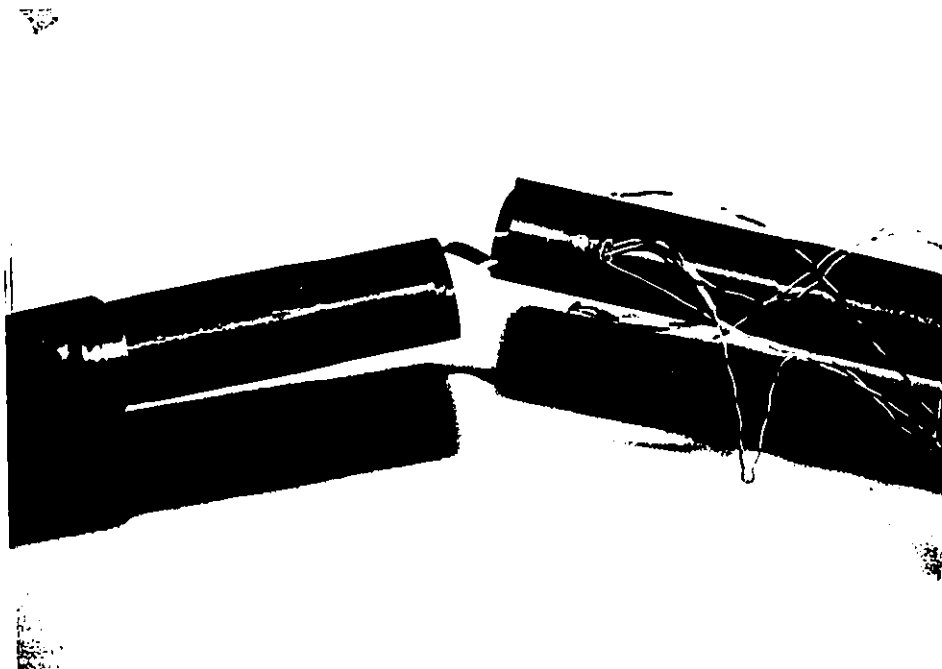


Figure 8.3 Positive torque failure mode (Tube 7).

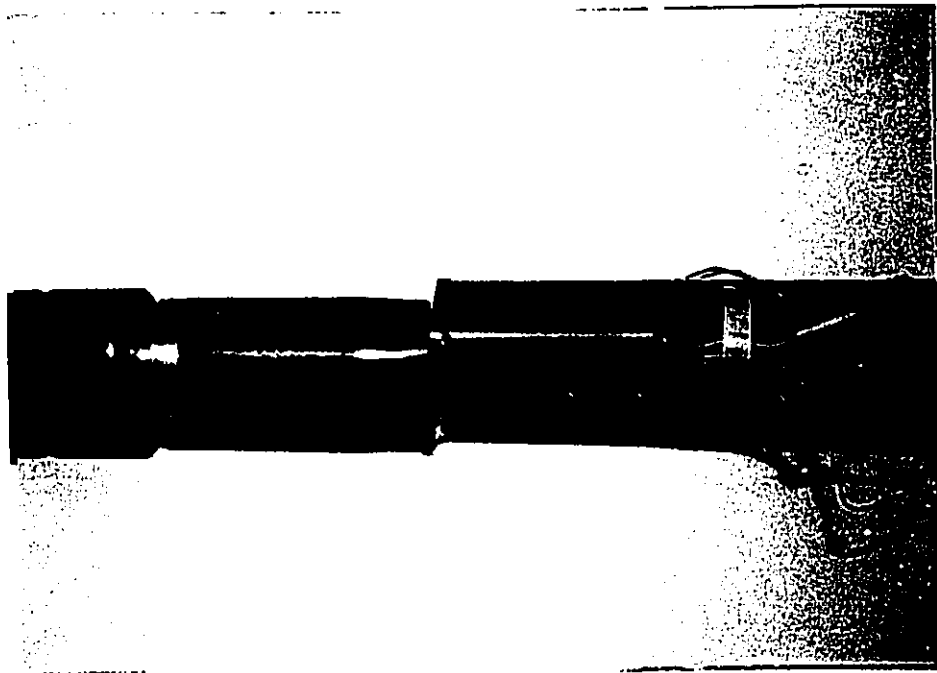


Figure 8.4 Negative torque failure mode (Tube 8).

**Table 8.1 Summary Of In-plane Shear Modulus and Strength Values
For Four Test Methods**

Test Methods	10° Off-Axis	±45° Tensile	90° Iosipescu	Shear Tube
No. Specimens	10	4	4	6
Fibre Volume Fraction, V_f (%)	59.5	61.4	62.7	56.2
Shear Strength (MPa)	76.0	90.0	44.0	103.8
Coefficient of Variation (%)	4.9	24.4	20.7	5.8
Shear Modulus (GPa)	6.0	6.1	4.2	5.5
Coefficient of Variation (%)	8.8	15.4	13.0	3.5

8.3.2 Evaluation Of Shear State

The initial goal for each of the four in-plane shear tests was a stress state which is dominated by shear. Since there are results in the literature that indicate that the three coupon test specimens do not have a pure shear state, it is of interest to compare the stress states of these specimens with that of the tubular specimen using three approaches.

8.3.2.1 Strain State Analysis

The first comparison employs the normalized strain curves which have already been

introduced. Curves were prepared for the $\pm 45^\circ$ tensile test data, the Iosipescu test data and for the tubular data (normalized strain curves for all specimens appear in Appendix E). The 10° off-axis data does not lend itself to this type of analysis. Figures 8.5 and 8.6 are typical normalized strain curves for the two coupon specimens. The $\pm 45^\circ$ tensile specimen has a reasonable level of pure shear with both strain readings within $\pm 10\%$ of 1.0; however, only test results for four of the nine $\pm 45^\circ$ tensile specimens were acceptable. For the 90° Iosipescu specimen testing, alignment of the specimen in the fixture affected the degree of pure shear. It is possible that in the simple tensile test used for the $\pm 45^\circ$ tensile specimen there may be an alignment effect.

As has been mentioned twelve Iosipescu specimens were originally tested. In-plane and out of plane bending can occur unless considerable care is used in installing the Iosipescu specimen in the test fixture. At that time the criterion for acceptance of the results was that all four normalized strain readings had to be within $\pm 10\%$ of 1.0. Since only four specimens passed this criterion, the 90° Iosipescu test requires considerable care to ensure a pure shear state. One would have to test 30 specimens in order to achieve 10 specimens which meet the criterion for acceptance. The tubes and $\pm 45^\circ$ tensile tests received no such attention because these specimens are actually quite simple to test whereas the 90° Iosipescu specimens can easily be improperly positioned in the test apparatus.

The results for the normalized shear values for the tubular specimens are generally much better than those of the $\pm 45^\circ$ specimens, both in spread of the data and number of acceptable specimens. All of the curves fall within the $\pm 10\%$ bounds. This consistency is indicated by the low coefficients of variation shown in Table 8.1. By comparison, the other two tests yield very high coefficients of variation for both shear strength and modulus results. The tubes clearly develop a more consistent shear state and one that is closer to pure shear than either the $\pm 45^\circ$ tensile and Iosipescu tests using normalized strain curves as a basis for comparison. Both the 90° Iosipescu and $\pm 45^\circ$ tensile specimens have a reasonably pure shear state; however, the fact that only four of each were acceptable is an indication that both tests are inconsistent.

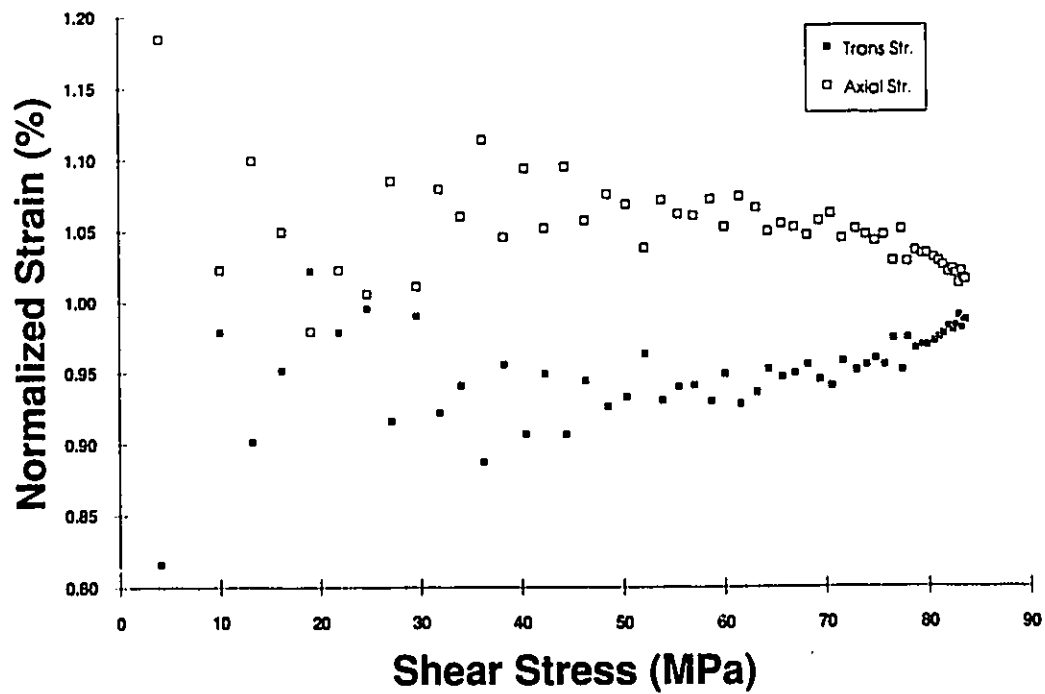


Figure 8.5 Typical normalized strain curves for $\pm 45^\circ$ tensile test.

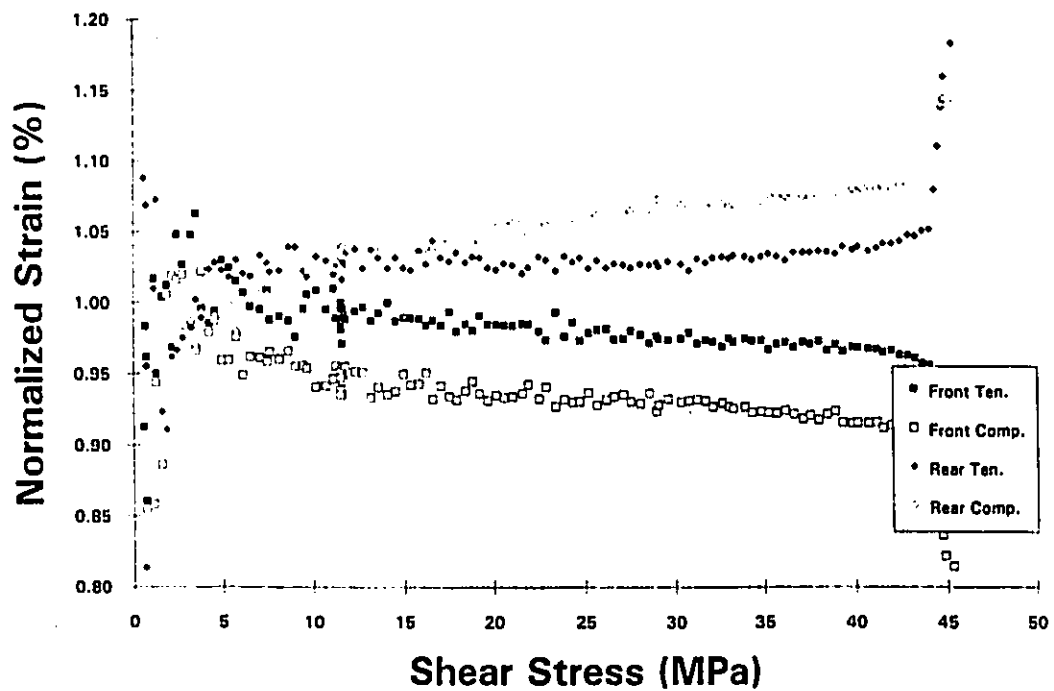


Figure 8.6 Typical normalized strain curves 90° Iosipescu specimen.

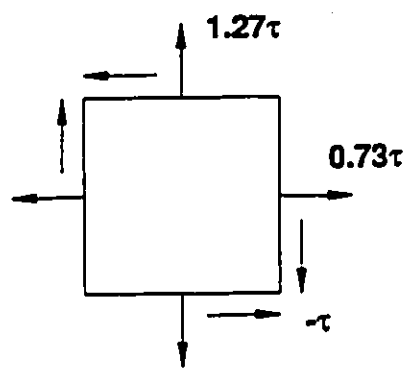
8.3.2.2 Stress State Analysis

A second method is to analyze the laminar shear state by using stress transformation matrices to convert the known stresses in the load axes to the material axes. This type of analysis is possible for the $\pm 45^\circ$ tensile, 10° off-axis, and tubular specimens. The 90° Iosipescu specimen does not lend itself to this type of analysis, but finite element analyses of several authors [6,7,8] have established the stress state for this specimen. Figure 8.7 shows the stress states in the material axes for the four specimens. The σ_1 and σ_2 stress levels have been normalized with respect to the in-plane shear stress, τ_{12} .

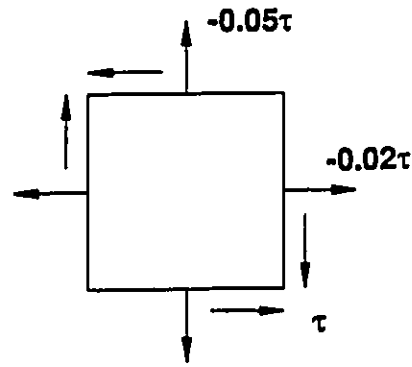
It is clear that neither the $\pm 45^\circ$ tensile specimen nor the 10° off-axis specimen develop a pure shear state due to mechanical loading. In the case of the 10° off-axis specimen the stress in the fibre direction is 5.7 times the shear stress. In both coupon specimens the contribution of the transverse stress to the total stress state is quite low. By contrast, the tube stress state is predominantly shear stress with the other two stress components at 7% of the shear stress level. The 90° Iosipescu specimen actually displays the highest level of pure shear with the longitudinally and transverse stresses reaching only 5% of the shear stress magnitude. This data is, however, produced from finite element analysis which assumes perfect alignment of the specimen in the test fixture. Given the difficulty in achieving the perfect alignment of the 90° Iosipescu specimen in the fixture, it seems likely that the tubular shear specimen is the only specimen which provides a consistently near pure shear state.

8.3.2.3 Microscopic Analysis

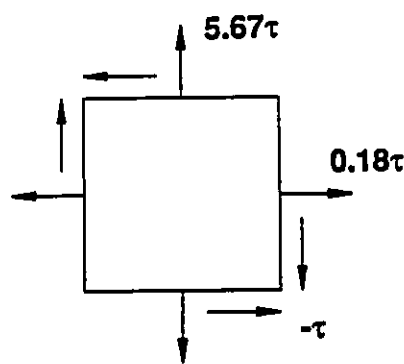
Microscopy is not as qualitative a method in the analysis of stress states as are the previous two approaches; however, trends can often be revealed by observing the specimen failure surfaces. The failure surfaces of all four specimens were analyzed in an attempt to identify cases of pure shear. Only the 90° Iosipescu, 10° off-axis and tubular specimens could be analyzed microscopically. The $\pm 45^\circ$ tensile specimen failure surfaces were too damaged from testing for a microscopic study to be performed.



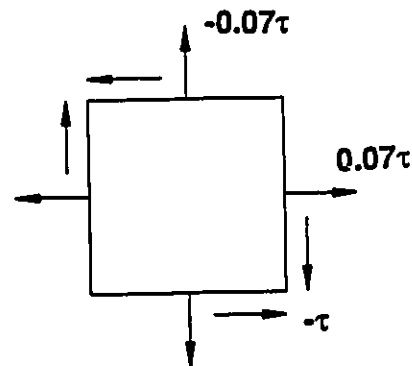
±45° Tensile Specimen



90° Iosipescu Specimen



10° Off-axis Specimen



Tubular Shear Specimen

Figure 8.7 Stress states for the four test methods as a function of the in-plane shear stress, τ_{12} .

Figures 8.8 - 8.10 are photomicrographs of the failure surfaces of the tubular, 10° off-axis, and 90° Iosipescu specimens taken by scanning electron microscope (SEM) at a magnification of 500 times. The tube failure surface is different from the other two in that there appears to be considerable debris (probably resin particles). The 90° Iosipescu specimen appears to have the smoothest failure surface with the most fibre definition which would suggest that a lower failure stress was necessary to fail the specimen. These observations corroborate the failure levels reported in Table 8.1. The higher magnification (1000X) SEM photomicrographs of Figures 8.11 - 8.13 reveal that all of the specimens show evidence of hackling which is associated with shear failure surfaces [5]. The smoothness of the 90° Iosipescu failure surface is more exaggerated in Figure 8.13 than in Figure 8.11.

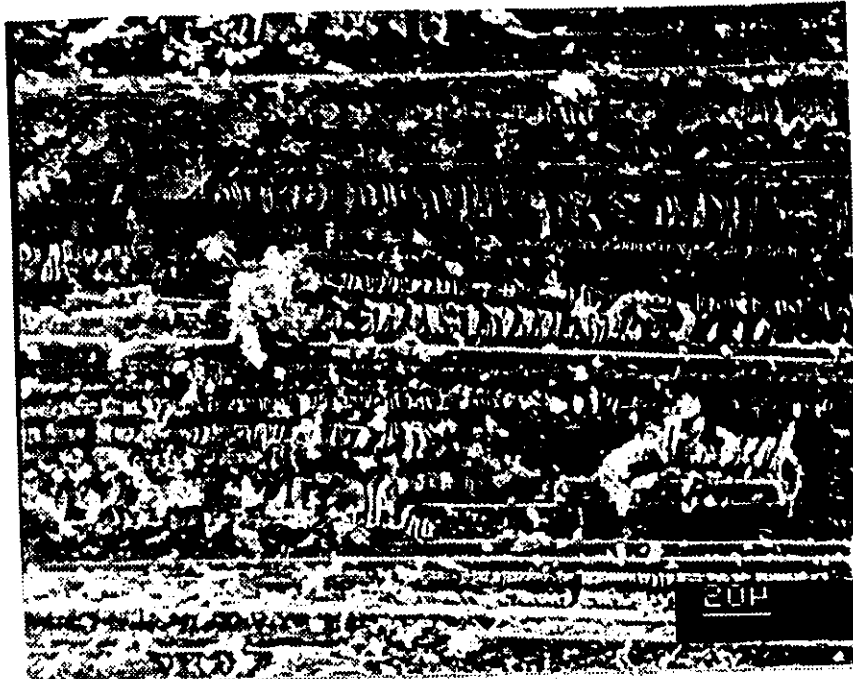


Figure 8.8 Failure surface of a tubular shear specimen (500X).

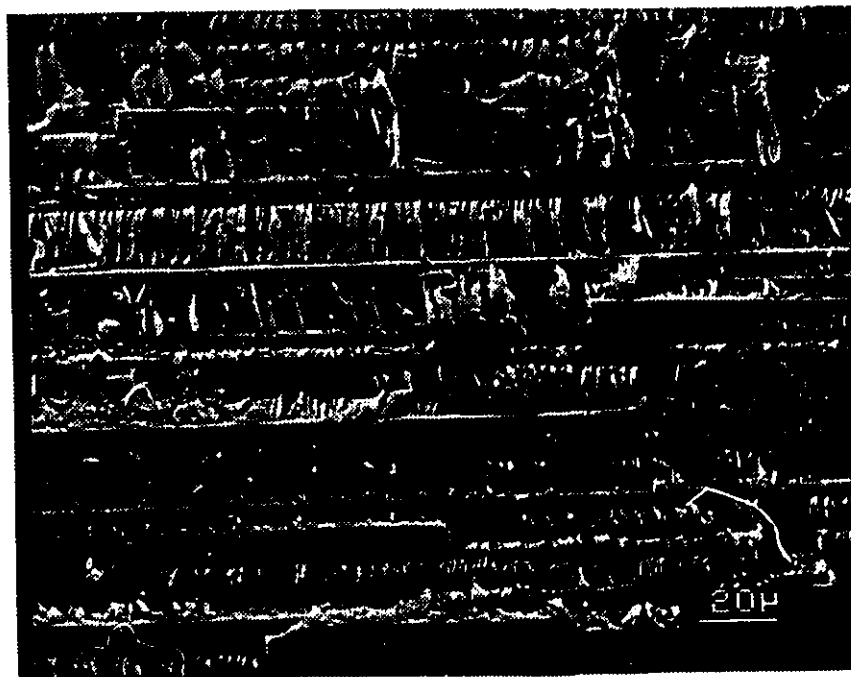


Figure 8.9 Failure surface of a 10° off-axis specimen (500X).

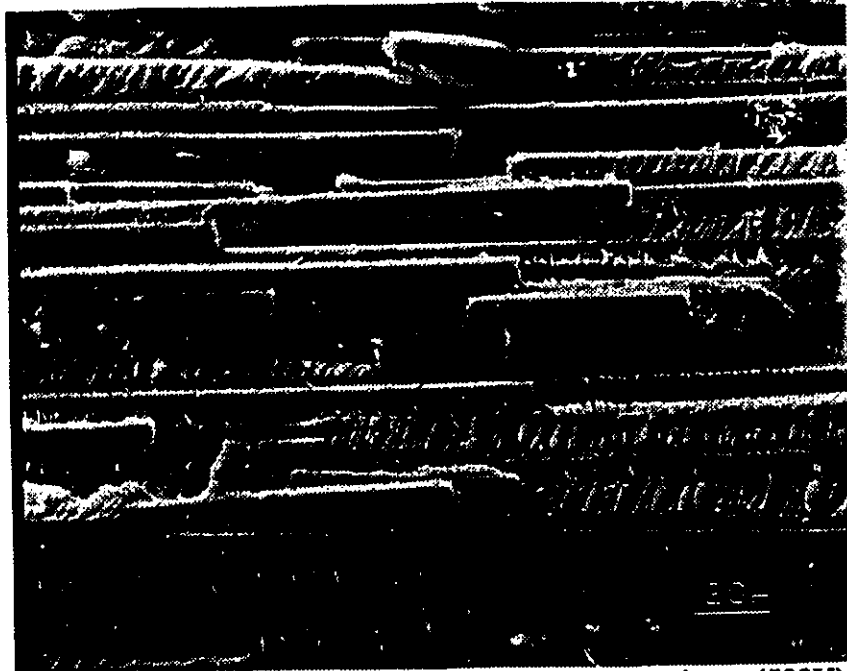


Figure 8.10 Failure surface of 90° Iosipescu specimen (500X).

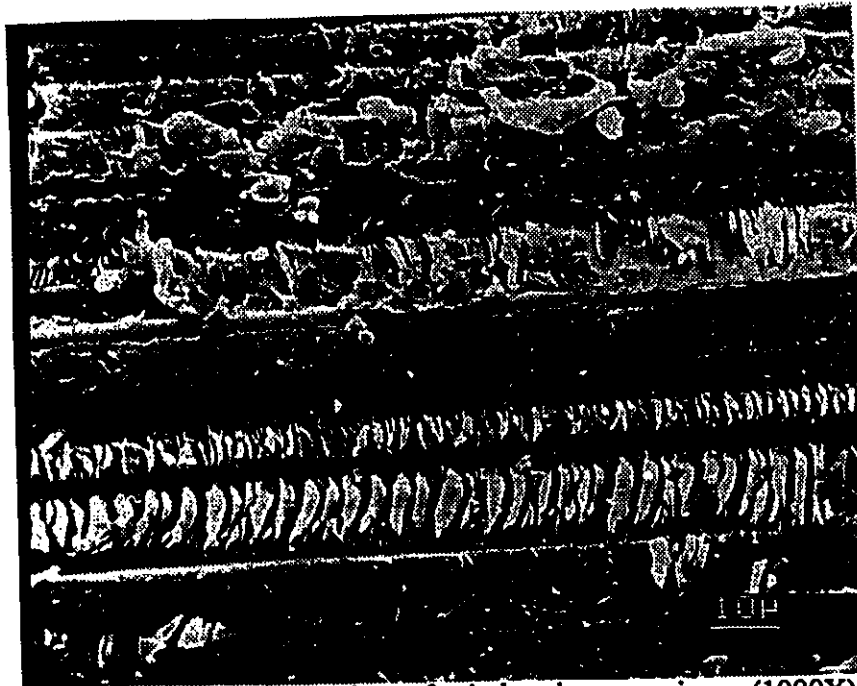


Figure 8.11 Failure surface of tubular shear specimen (1000X).



Figure 8.12 Failure surface of 10° off-axis specimen (1000X).

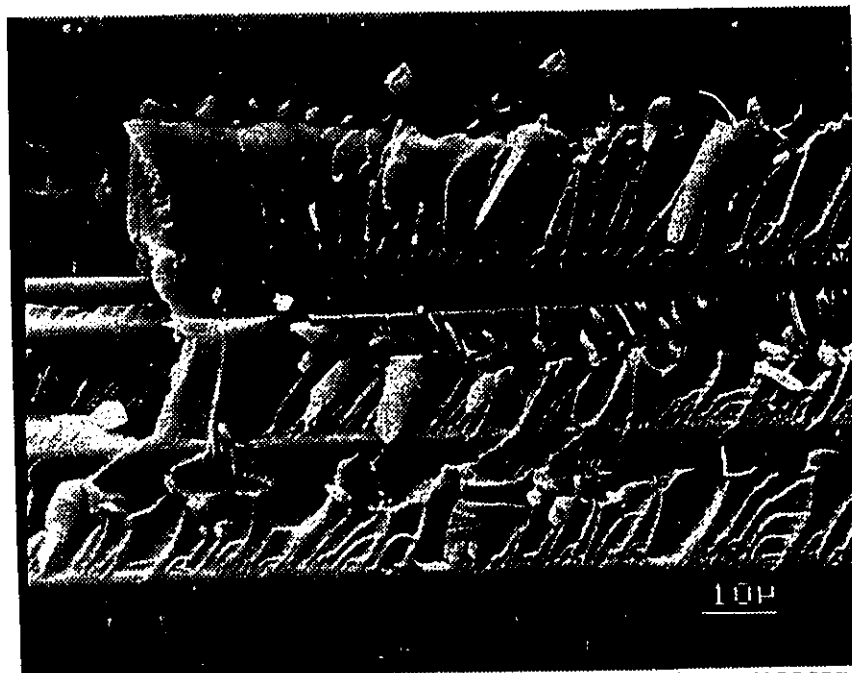


Figure 8.13 Failure surface of 90° Iosipescu specimen (1000X).

Figures 8.14 -8.16 are SEM photomicrographs taken at a magnification of 2500 times. At this magnification the similarities of the 10° off-axis and the tubular specimens are even more apparent. Both have a high degree of hackling with good adhesion between resin and the fibres. The 90° Iosipescu, alternatively, seems to be very smooth with only the regular river patterns indicating a minimal amount of irregularity in the resin surface. Fibres can be seen to run beneath the resin which may indicate good resin/fibre adhesion.

In spite of the normalized strain evidence presented it does not appear from the SEM photographs that the 90° Iosipescu specimen failed in pure shear. Figure 8.17 is a photograph of two failed 90° Iosipescu specimens. One specimen failed in the notched region as expected and one outside of the notched area. Further investigation showed that the failure to the right of the notch occurred directly beneath where the load is introduced to the specimen during testing. This suggests that the region between the notches which is the test section of the specimen may not be the region of highest stress. Stress may be concentrated beneath the load point causing premature failure of the specimen at that point. The smooth failure surface of Figure 8.16 may be due to tensile stresses (from bending) in addition to the shear stresses.

As mentioned, SEM analysis of the $\pm 45^\circ$ tensile specimen was not possible due to the destructive nature of this test (Figure 8.18). Of particular note is the amount of exposed and broken fibre that dominates the failure site. This indicates that there is extensive fibre interaction during failure. It is worth pointing out that despite the evidence in favour of a mixed mode failure the $\pm 45^\circ$ tensile specimens failed at nearly the same shear strength as the tubular specimens, although the ultimate shear strain was significantly greater.



Figure 8.14 Failure surface of tubular shear specimen (2500X).



Figure 8.15 Failure surface of a 10° off-axis specimen (2500X).

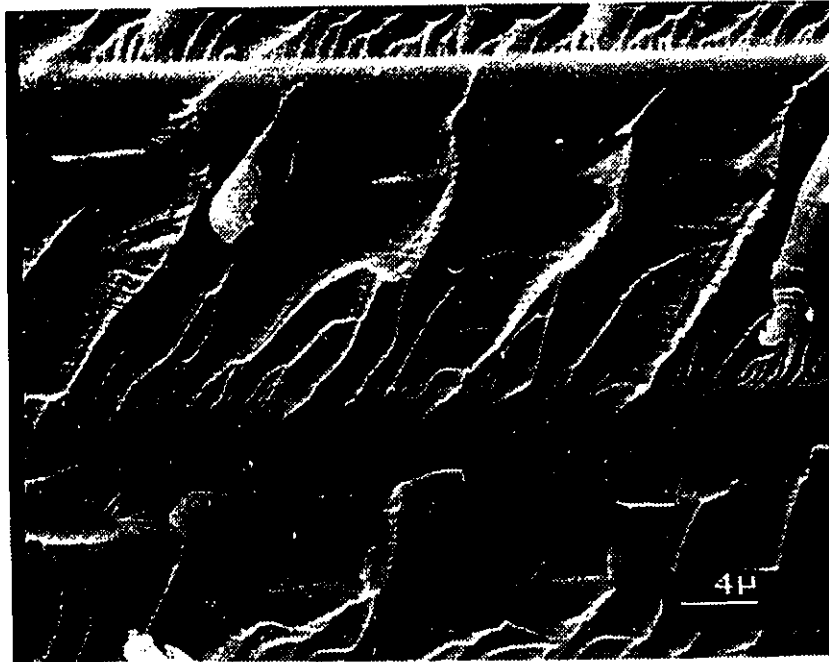


Figure 8.16 Failure surface of a 90° Iosipescu specimen (2500X).

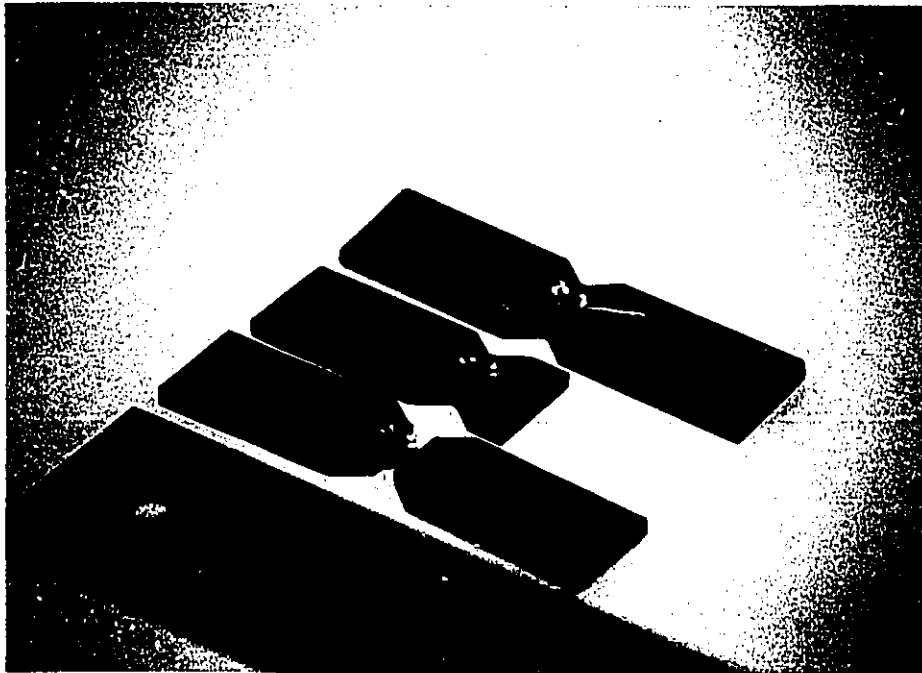


Figure 8.17 Failed 90° Iosipescu specimen.

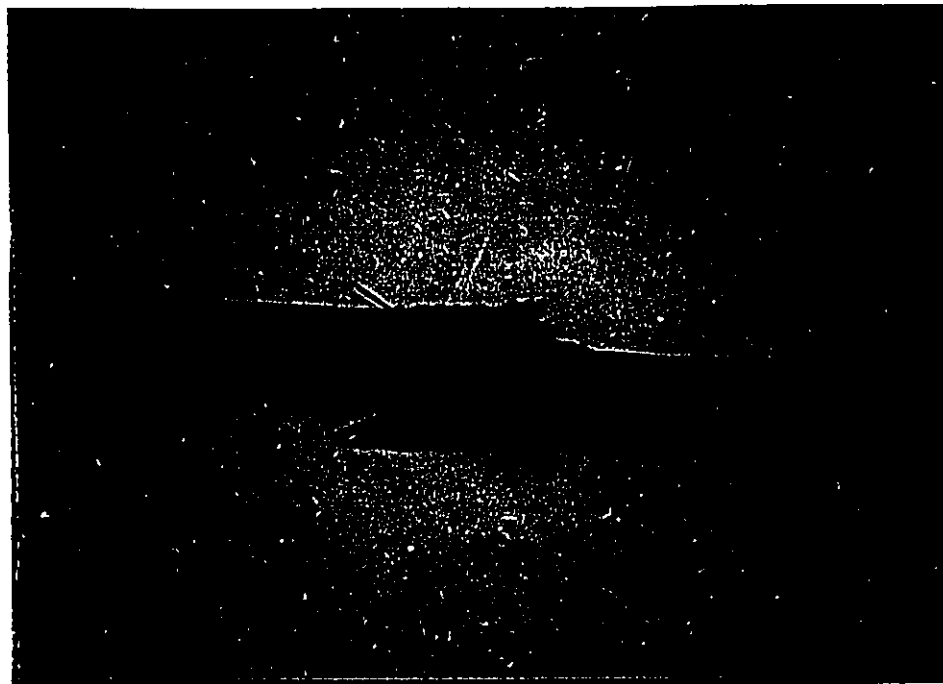


Figure 8.18 Failed $\pm 45^\circ$ tensile specimen.

8.3.3 Summary of Shear State Analysis

It is difficult to say conclusively that one of the test specimens is better than any of the others based on the evidence presented. The tubular in-plane shear test appears to have the highest level of pure shear, although the normalized strain curves indicate that the results for the four 90° Iosipescu specimens are comparable to the tubes. The shear modulus and shear strength values for the tubular test are also more consistent than those of the other tests. It is worth noting that the 10° off-axis specimens had a coefficient of variation for shear strength of 7.0% which is only slightly higher than for the tubular specimens (5.0%). In addition, the 10° off-axis failure surfaces appear to be nearly identical to those of the tubes with the exception of the resin debris which is characteristic of the tube failure. In spite of the many similarities of the four tests there is no evidence which suggests that the tubes did not fail in almost a pure shear state. By contrast, each of the other tests show some evidence of a mixed stress state and/or are not consistent. Circumstantially, at least, it would appear that the tubular in-plane shear test is the only true shear test.

8.4 Selection Of An In-plane Shear Test Method Based On The Shear Sensitivity Of Laminate Tensile Modulus

The evaluation of the shear state of the specimens during testing and the analysis of failure results was sufficient for determining which of the tests studied actually measured the in-plane shear modulus or shear strength of the AS4/3501-6 material. This is not, however, the most important criterion for evaluating which of these shear test methods should be used to generate composite properties. In particular, it has not been established that the shear modulus found using these tests can be used in laminate calculations with any degree of certainty. Lee et. al. have published a series of papers [3,5,12,15] whose intent it was to determine which of the 90° Iosipescu, 10° off-axis and $\pm 45^\circ$ tensile test methods was most suitable for the determination of the composite in-plane shear modulus. In one of these [12], a method was developed which provided a basis for comparing these tests. The selection criterion was based on which method best represented the in-plane shear behaviour of a lamina in a laminate. The laminate used for their study was found to be too insensitive to the differences between the shear moduli found using the three test methods. It was therefore not possible to select the most accurate in-plane shear method. It was recommended in the paper that a laminate which has a tensile modulus that is more dependent on the in-plane shear modulus be used to evaluate the in-plane shear test methods.

Four shear sensitive laminates were determined in a separate but related project [13]. The testing of these has already been discussed. A "shear sensitive" laminate has a tensile modulus, E_{xx} , which is substantially dependent on the in-plane shear behaviour of the laminae. In particular, the in-plane shear modulus would contribute significantly to the tensile modulus of the laminate. The degree to which the laminate tensile modulus changes when G_{12} is varied was defined as the shear sensitivity of the laminate [13]. For this analysis E_{11} , E_{22} and ν_{12} were held constant while G_{12} was varied [13]. The elastic constants for Hercules AS4/3501-6 laminae, which appear in Appendix B, were measured using ASTM 3039-76 [61]. Laminate tensile elastic moduli, E_{xx} , were then calculated using Classical Laminate Plate Theory and shear sensitivity was estimated by calculating the change in E_{xx} for a corresponding change in G_{12} . The resulting

four shear sensitive laminates were $[\pm 55, \pm 50]_{3S}$, $[\pm 35, \pm 40]_{3S}$, $[\pm 65, \pm 50]_{3S}$ and $[\pm 25, \pm 40]_{3S}$.

It is possible, using Classical Laminate Plate Theory, to calculate the tensile modulus for a particular laminate given the laminar material properties. By substituting the G_{12} values found by each of the four shear tests in Classical Laminate Plate Theory (all other material properties held constant) four different theoretical E_{xx} values for each of the four shear sensitive laminates were determined. The four laminates were fabricated and their tensile moduli were found experimentally (Table 7.5).

Each of the specimens types that were tested had different fibre volume fractions. It was therefore necessary to adjust all of the input elastic constants for Classical Laminate Plate Theory (E_{11} , E_{22} , G_{12} , ν_{12}) to be representative of properties for the laminate volume fractions. The longitudinal tensile moduli, E_{11} , were scaled according to a linear relationship between the resin tensile modulus and the tensile modulus of the longitudinal test specimens [2]. The transverse tensile moduli, E_{22} , and in-plane shear moduli, G_{12} , were adjusted according to a ratio predicted from the two fibre volume fractions by Halpin-Tsai relationships [2]. The major Poisson's ratio, ν_{12} , was not modified. The modified elastic constants for all of the laminates are given in Table 8.2. The $\pm 45^\circ$ tensile in-plane shear test specimen can also be considered as a laminate specimen. It has been found that this laminate is, in fact, the most shear sensitive of any combination [13]. The modified constants for the $\pm 45^\circ$ tensile specimen are also in Table 8.2. The laminate tensile modulus values for the $[\pm 45]_{4S}$ laminate were determined using the same method as for the other four laminates. The average tensile modulus was 20.1 GPa with a coefficient of variation of 7.8%.

The modified elastic constants were next used in Classical Laminate Plate Theory calculations to predict laminate tensile modulus values for each shear test. These results are summarized with the experimental tensile modulus values in Table 8.3. The percentage differences between the predicted E_{xx} values for each shear test and the experimental E_{xx} values also appear in Table 8.3.

Table 8.2 Summary of Adjusted Material Properties

Specimen Type	Adjusted Shear Modulus (GPa)				
	$[\pm 45]_{4S}$	$[\pm 55, \pm 50]_{3S}$	$[\pm 35, \pm 40]_{3S}$	$[\pm 65, \pm 50]_{3S}$	$[\pm 25, \pm 40]_{3S}$
10° Off-axis	6.5	6.4	5.5	6.4	6.2
$\pm 45^\circ$ Tensile	6.2	5.9	5.1	6.2	5.7
90° Iosipescu	4.0	3.9	3.4	4.1	3.8
Tubular	6.5	6.2	5.4	6.4	6.0
Other Mechanical Properties	$[\pm 45]_{4S}$	$[\pm 55, \pm 50]_{3S}$	$[\pm 35, \pm 40]_{3S}$	$[\pm 65, \pm 50]_{3S}$	$[\pm 25, \pm 40]_{3S}$
E_{11}	136.9	133.4	124.5	136.1	130.5
E_{22}	7.9	7.4	6.6	7.8	7.2
ν_{12}	0.3	0.3	0.3	0.3	0.3

It is apparent that the $\pm 45^\circ$ tensile shear test most accurately predicts the experimental laminate secant tensile moduli at a shear strain of 5000 $\mu\epsilon$. The 90° Iosipescu is closest to predicting the tensile modulus of the $[\pm 65, \pm 50]_{3S}$ laminate but in all other cases the $\pm 45^\circ$ method is closest and is next closest to the 90° Iosipescu in predicting the $[\pm 65, \pm 50]_{3S}$ laminate. Not surprisingly, the $\pm 45^\circ$ tensile shear test has a very low percentage difference (4.5%) between the experimental and predicted values for predicting its own laminate response. This confirms the methodology used to evaluate the four shear tests (using the tensile response of the four laminates).

Table 8.3 Summary of Predicted and Experimental Tensile Modulus Values

Laminate	Experimental Modulus, E_{xx} (GPa)	Predicted Modulus, E_{xx} (GPa)				Percentage Difference (%) ¹			
		10°	±45°	90° Ios.	Tubes	10°	±45°	90° Ios.	Tubes
[±45] _{4s}	20.1	21.8	21.0	14.4	21.8	8.4	4.5	-28.4	8.4
[±55,±50] _{3s}	13.9	15.8	14.8	10.7	15.4	13.5	6.5	-23.3	10.6
[±35,±40] _{3s}	26.7	29.8	28.3	21.2	29.2	11.6	6.1	-20.6	9.4
[±65,±50] _{3s}	14.0	15.8	15.6	12.6	15.8	12.8	11.4	-9.6	12.8
[±25,±40] _{3s}	44.4	48.9	47.3	40.6	48.3	10.3	6.6	-8.6	8.9

$$1 \frac{(Pred-Exp)}{Exp} \times 100$$

Table 8.4 Summary of Laminate Shear Sensitivity

Laminate	Shear Modulus, G_{12} (GPa)	Most Accurate Shear Test	Laminate Shear Sensitivity (%)
[±45] _{4s}	6.2	±45° tensile	100
[±55,±50] _{3s}	5.9	±45° tensile	88
[±35,±40] _{3s}	5.1	±45° tensile	86
[±65,±50] _{3s}	4.1	90° Iosipescu	69
[±25,±40] _{3s}	5.7	±45° tensile	50

Figures 8.19 - 8.23 are laminate stress-strain curves which summarize graphically the values presented in Table 8.3. Of particular note is that the 10° off-axis shear test predictions are always the farthest above the experimental curve and the 90° Iosipescu prediction are the only curves which are below the experimental curves. The tubular shear test E_{xx} predictions are within 10% above the experimental values for all but one of the laminates with the 10° off-axis test predictions being slightly greater than 10% above the experimental values for all but one laminate.

The laminate shear sensitivity is an important parameter that can be calculated using Classical Laminate Plate Theory. The methodology is a simple one. A laminate tensile modulus was calculated for each specimen with the laminar shear modulus set to zero. Next, a laminate tensile modulus was determined by using the in-plane shear modulus (Table 8.2) which best predicts the laminate tensile modulus in Table 8.3. For example, the corrected shear modulus for the $\pm 45^\circ$ tensile test was used for the $[\pm 45]_{4s}$, $[\pm 55, \pm 50]_{3s}$, $[\pm 35, \pm 40]_{3s}$ and $[\pm 25, \pm 40]_{3s}$ laminates while the 90° Iosipescu corrected shear modulus was used in the case of the $[\pm 65, \pm 50]_{3s}$ laminate. Table 8.4 summarizes these parameters and the shear sensitivity of each laminate.

A trend between the shear sensitivity of the laminates and the laminate tensile modulus predictions can be noted. The shear sensitivity of the $[\pm 45]_{4s}$ laminate is entirely dependent on the value of the in-plane shear modulus. For this laminate the $\pm 45^\circ$ tensile, 10° off-axis and tubular shear test tensile modulus predictions agree well with the experimental laminate tensile modulus. The 90° Iosipescu tensile modulus prediction is almost 30% below the experimental tensile modulus for the $[\pm 45]_{4s}$ laminate. The $\pm 45^\circ$ and tubular shear test results agree closely with the $[\pm 55, \pm 50]_{3s}$ and $[\pm 35, \pm 40]_{3s}$ laminates. As the laminate shear sensitivity decreases, however, the 90° Iosipescu E_{xx} predictions are progressively closer to the experimental E_{xx} values. In fact, as the laminate shear sensitivity decreases, all of the E_{xx} predictions approach the experimental values. Ideally, as the laminate shear sensitivity decreases to very low values the shear results of the four tests should yield predicted E_{xx} values very close together as was the case for Lee et. al. [12]. The trend identified here tends to corroborate this finding.

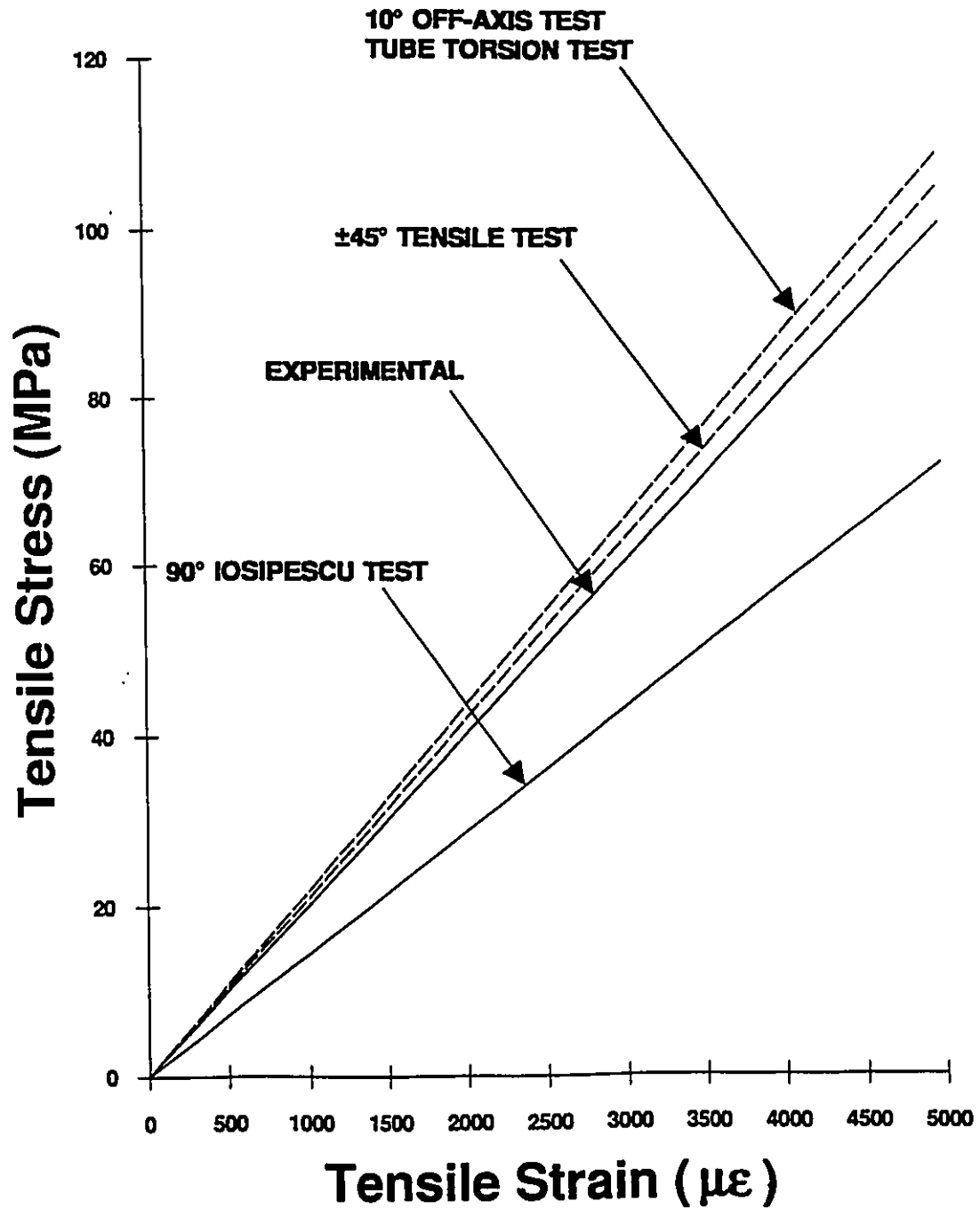


Figure 8.19 Tensile stress-strain curves for predictive and experimental results for $[\pm 45]_{4s}$ laminate.

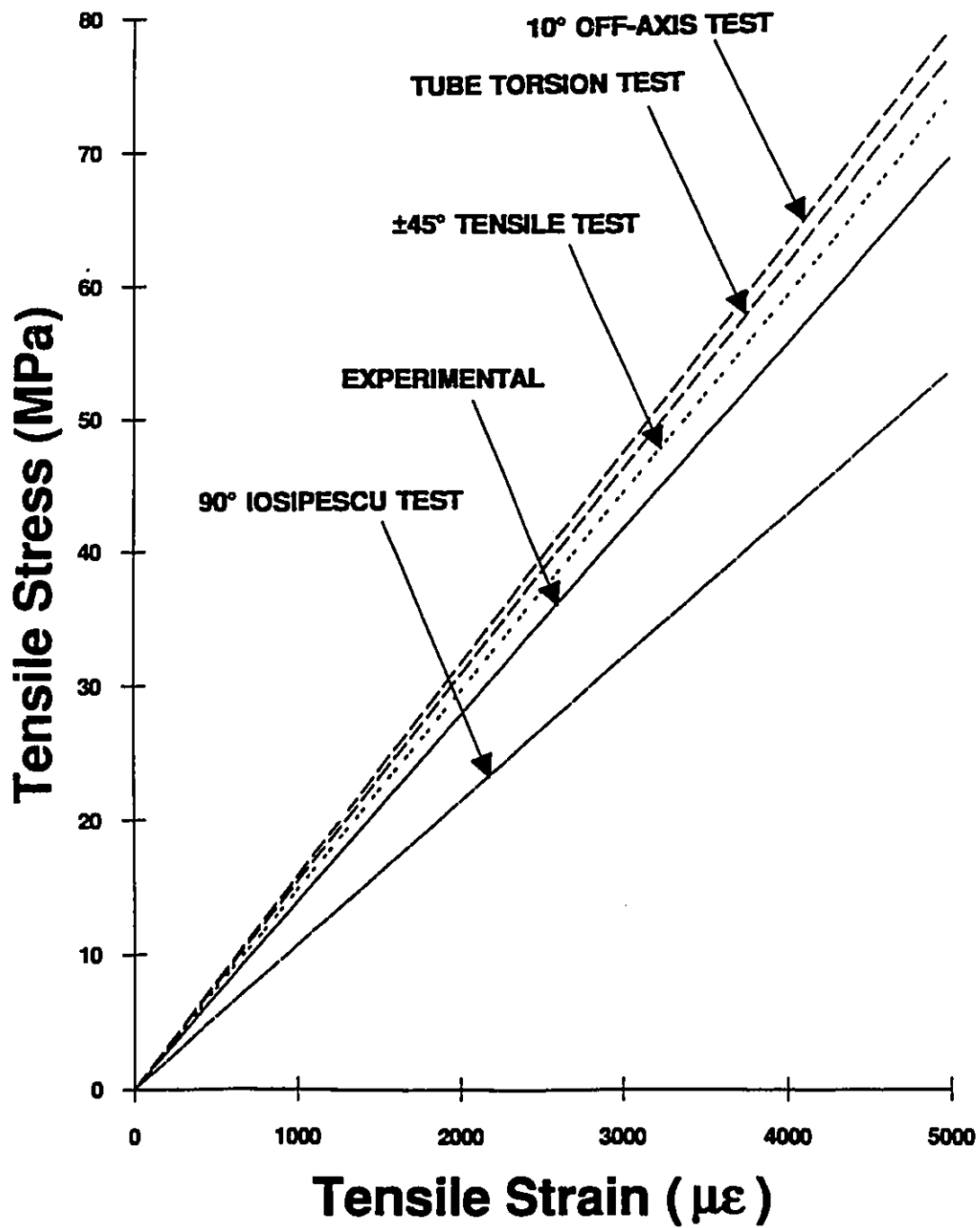


Figure 8.20 Tensile stress-strain curves for predicted and experimental results for $[\pm 55, \pm 50]_{3s}$ laminate.

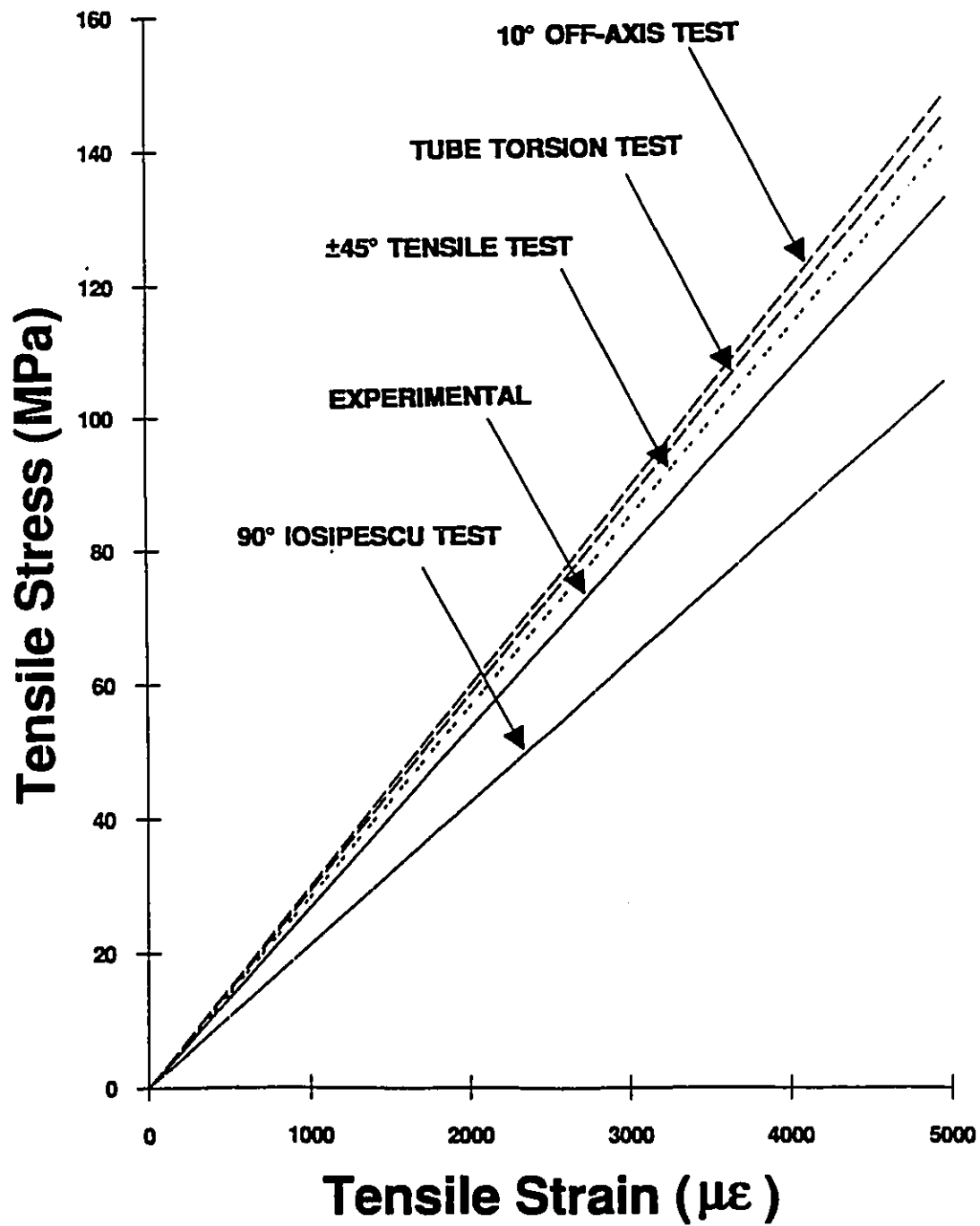


Figure 8.21 Tensile stress-strain curves for predictive and experimental results for $[\pm 35, \pm 40]_{3s}$ laminate.

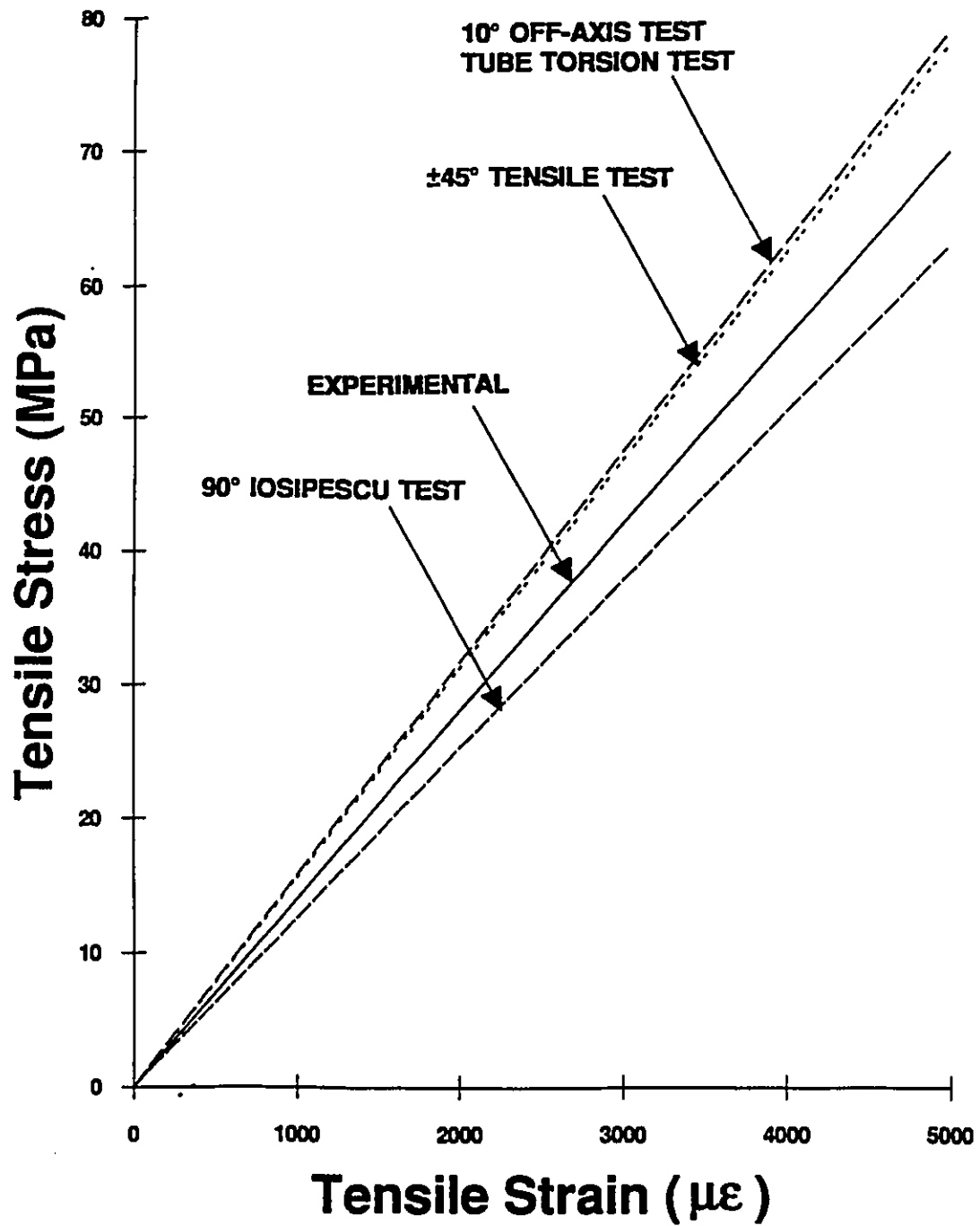


Figure 8.22 Tensile stress-strain curves for predictive and experimental results for $[\pm 65, \pm 50]_{3s}$ laminate.

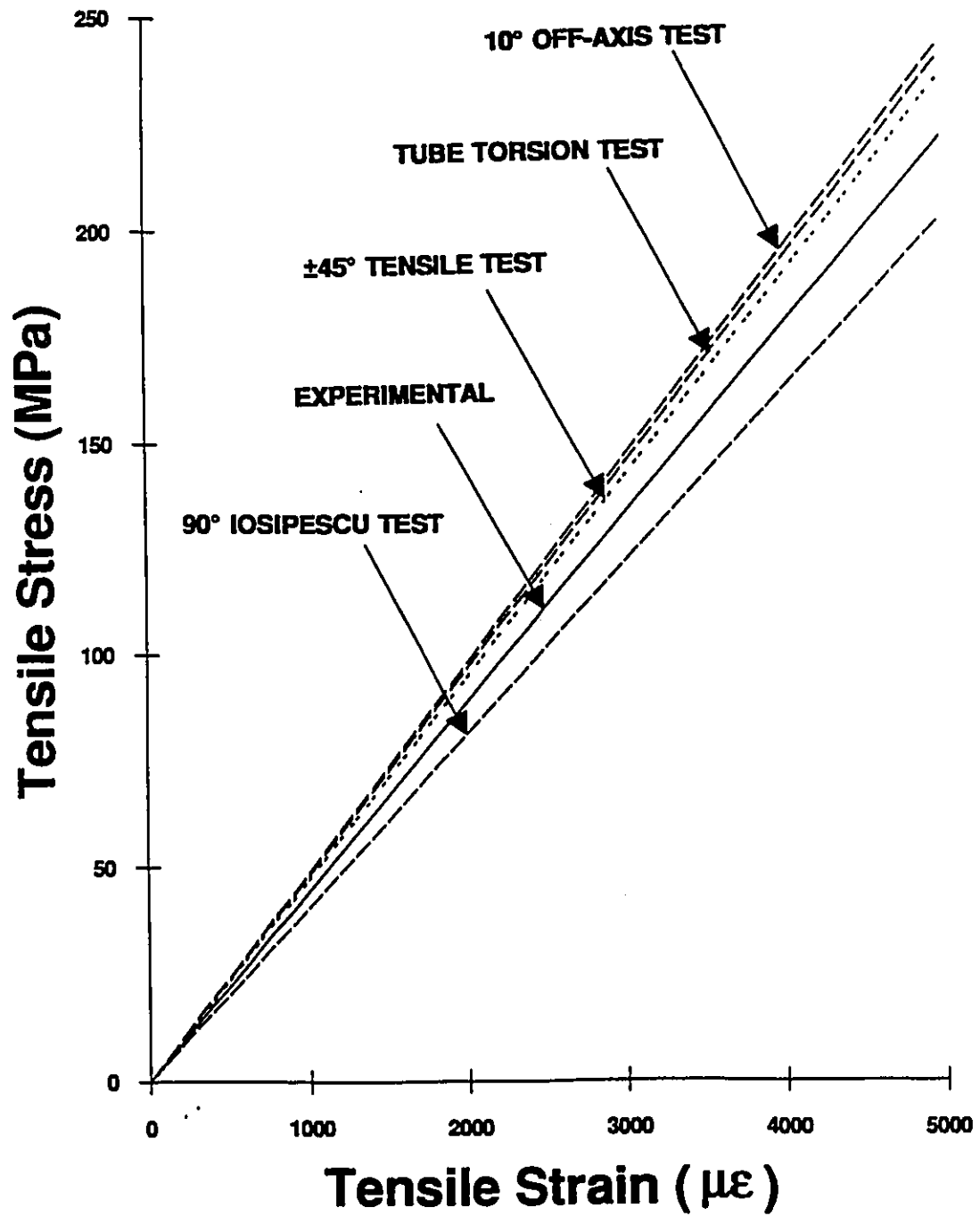


Figure 8.23 Tensile stress-strain curves for predictive and experimental results for $[\pm 25, \pm 40]_{3s}$ laminate.

It has been demonstrated that the tubular in-plane shear specimens produced the most uniform shear state of the four specimens considered. It is therefore difficult to explain why the $\pm 45^\circ$ tensile shear test predictions are, generally, closest to predicting the experimental E_{xx} values and not the tubular shear test predictions. The stress states of the individual laminae in the laminates were examined to determine if any trends existed which might explain this discrepancy. Figure 8.24 presents the stress states of the laminae with fibres oriented in the positive sense within each laminate (laminae with fibres oriented in the negative sense have the same stress magnitudes as those of Figure 8.24 but the shear stress is of opposite sign). All of the stresses are shown as a fraction of the maximum laminate shear stress. In each case there is a large disparity between the longitudinal stresses of the individual laminae. In effect, the in-plane stresses are vastly different from layer to layer. This must result in high interlaminar stresses which might affect the laminate stress state.

There was no trend which developed between the stress states of the laminates and those of the 10° off-axis, 90° Iosipescu and tubular shear tests. Each use unidirectional specimens which do not have interlaminar stresses. Certain layers of the laminates have stress states which resemble those of the $\pm 45^\circ$ lamina. One notable similarity between all the laminates (including the $\pm 45^\circ$ stress state) was that the shear stress changed its sign from layer to layer. In addition, the $\pm 45^\circ$ laminate tensile stress was significantly larger than its transverse stress which is similar to most of the lamina stress states of the laminates.

8.4.1 Summary of Laminate Prediction Results

There is a clear trend apparent from an analysis of the results of Table 8.3. The predicted tensile moduli of the $\pm 45^\circ$ tensile shear test, found using the experimentally determined shear modulus, most closely predict the experimental tensile modulus of the $[\pm 45]_{4S}$, $[\pm 55, \pm 50]_{3S}$, $[\pm 35, \pm 40]_{3S}$ and $[\pm 25, \pm 40]_{3S}$ laminates. The E_{xx} prediction found using G_{12} of the 90° Iosipescu shear test best predicts E_{xx} of the $[\pm 65, \pm 50]_{3S}$ laminate while the $\pm 45^\circ$ tensile shear test predictions were next best in this case. It is also evident that as the shear sensitivity decreases, the E_{xx} predictions for all four test methods converge to the experimental E_{xx} values. Thus for

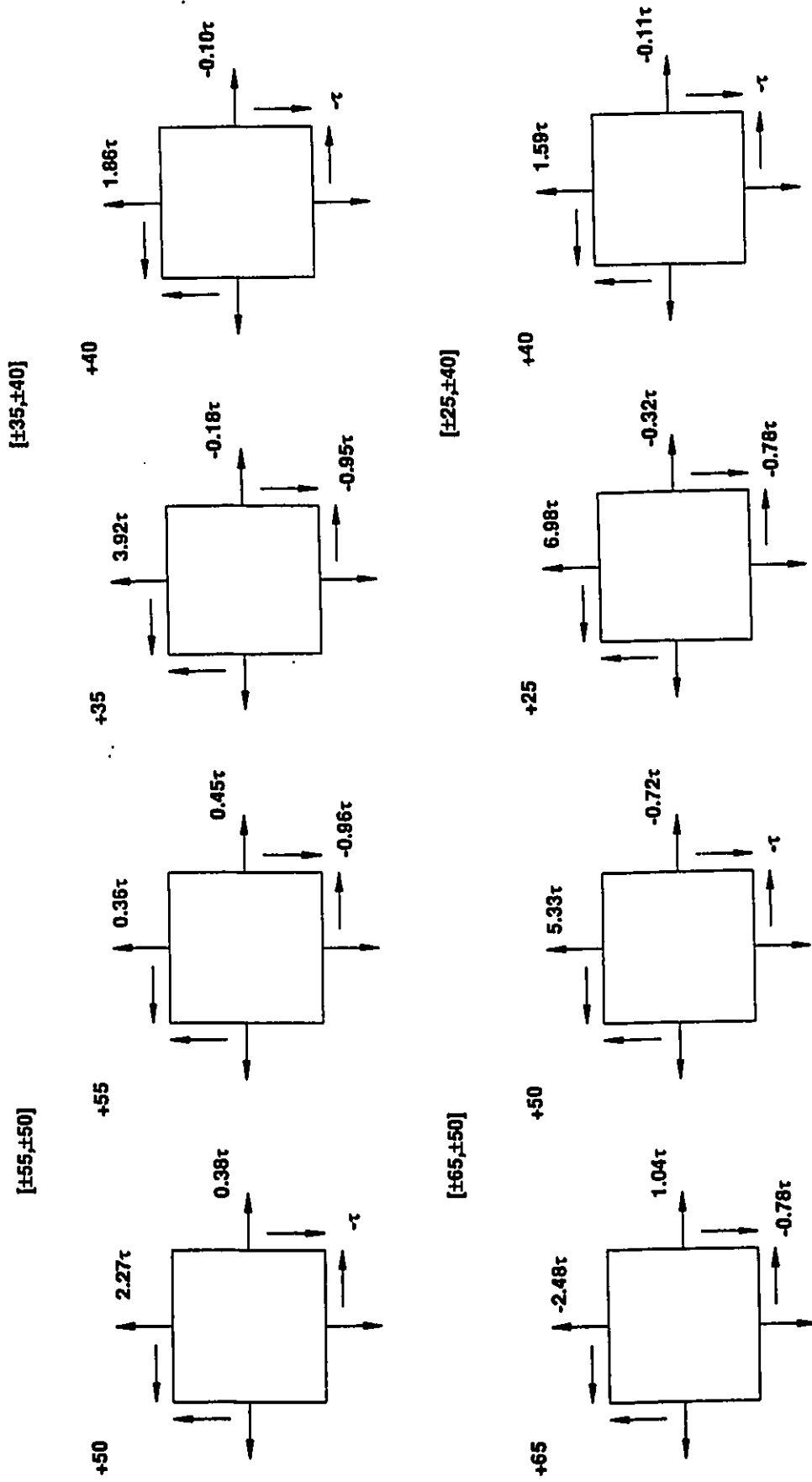


Figure 8.24 Stress states of individual lamina from the four laminates.

laminates that are less shear sensitive than the $[\pm 25, \pm 40]_{3S}$ laminate, any of the four in-plane shear test methods are appropriate.

It was unexpected that the predictions using the G_{12} values for the $\pm 45^\circ$ tensile specimen would be more accurate than those employing the shear tube results. Since it has been shown that the torsion testing of a thin-walled tube results in a much purer shear state as compared to the tensile testing of a $\pm 45^\circ$ specimen, the opposite result was expected. This implies that some other effect particular to laminates is contributing to the stress state in the laminate. The stress state analysis of the laminates suggests that interlaminar stresses could be the reason for this unexpected result.

The final comment is with regards to the practical use of these results. If the design of a shear-sensitive component, fabricated using AS4/3501-6 composite material, requires a highly accurate estimate of laminate modulus, then the $\pm 45^\circ$ tensile in-plane shear test method should be used. Alternatively, if a greater error (10-15%) can be tolerated, then either of the 10° off-axis tensile, $\pm 45^\circ$ tensile or torsion tube shear test methods can be employed.

Chapter 9.0

Conclusions

1. A design procedure was developed which resulted in a successful tubular in-plane shear specimen that did not buckle in torsion and failed in the gauge section.
2. A vacuum bagging technique has been developed which produces a thin-walled composite tube with excellent surface finish and tight geometric tolerances.
3. A tubular specimen end fitting (internal taper) was designed and manufactured that withstood a maximum torque load of 587 Nm.
4. Residual stresses which were found to exist within the tube are increased as the winding tension is increased and when a higher CTE and/or lower stiffness mandrel is used for fabrication.
5. Based on an analysis of normalized shear data, shear stress states and microscopic analysis it was found that the tubular in-plane shear specimen developed the most reliable state of pure shear of the four shear tests studied. In addition, the tubular specimens have shear properties with the lowest coefficients of variation of the four tests studied. These results suggest that the tubular specimen should be used for determining the in-plane shear strength.

6. For shear sensitive AS4/3501-6 laminates the $\pm 45^\circ$ tensile shear test G_{12} results yielded predicted laminate tensile modulus values which most closely corresponded to experimental tensile modulus results. If an accuracy of 10-15% can be tolerated, then either of the 10° off-axis tensile, $\pm 45^\circ$ tensile or tubular shear test methods can be employed.

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Chapter 10

Recommendations

The tapered internal end fitting was developed to satisfy the needs of the tubular in-plane shear specimens. The design is a novel one and should be studied further to determine the optimal taper and the maximum loads that such an end fitting can sustain.

The laminate prediction study was conclusive; however, it is surprising that the shear modulus values determined using the tubular in-plane shear specimens did not most accurately predict laminate response. Evidence was presented which suggested that an interlaminar stress state might affect the tensile modulus predictions. A measure of the interlaminar stresses could possibly be used to correct the tensile modulus predictions. Further study of the potential effects of interlaminar stresses should be pursued.

Although the tubular shear test does not appear at this stage to be the most appropriate for predicting flat laminate elastic response, it may be that tubular in-plane shear properties are required for predicting the elastic response of tubular laminates. Since tubular composite components are becoming more popular, this study should also be undertaken.

Finally, many of the 90° Iosipescu and $\pm 45^\circ$ tensile specimens were not used for the study of in-plane shear properties on the basis that their normalized strain readings were not within $\pm 10\%$ of 1.0. With today's sophisticated data acquisition software it is possible to monitor

normalized strain in real time. This approach might indicate specimen misalignment early in the testing so that it can be corrected. In this way it should be possible to have more specimens which satisfy the normalized strain criterion.

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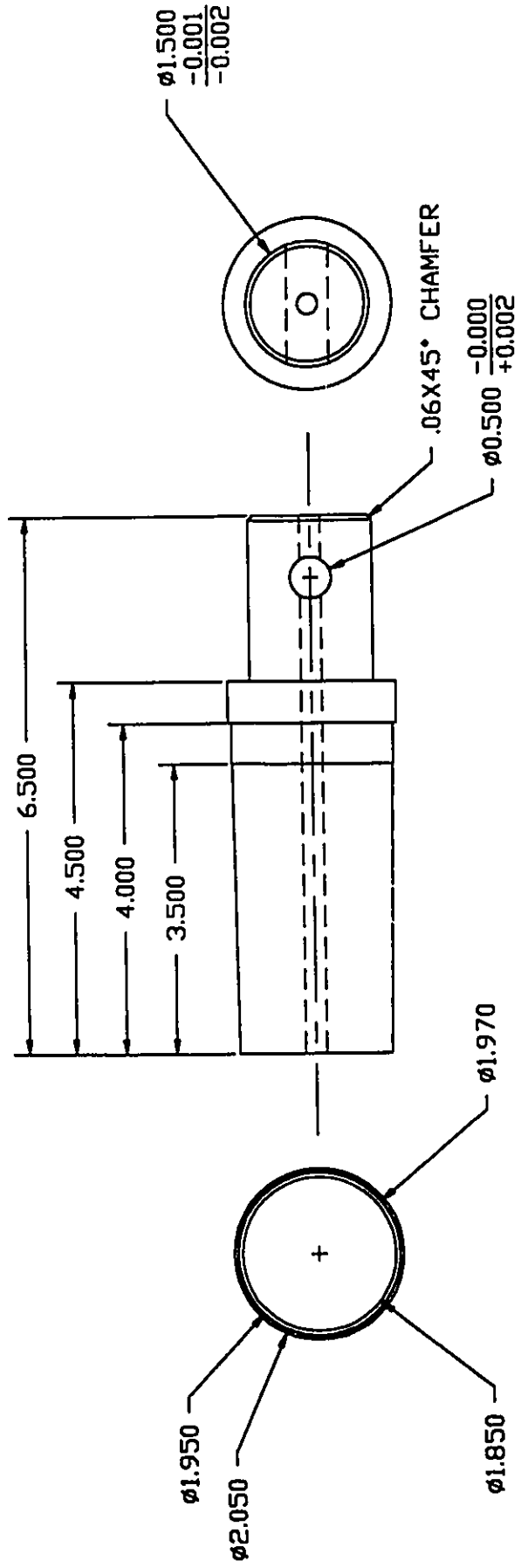
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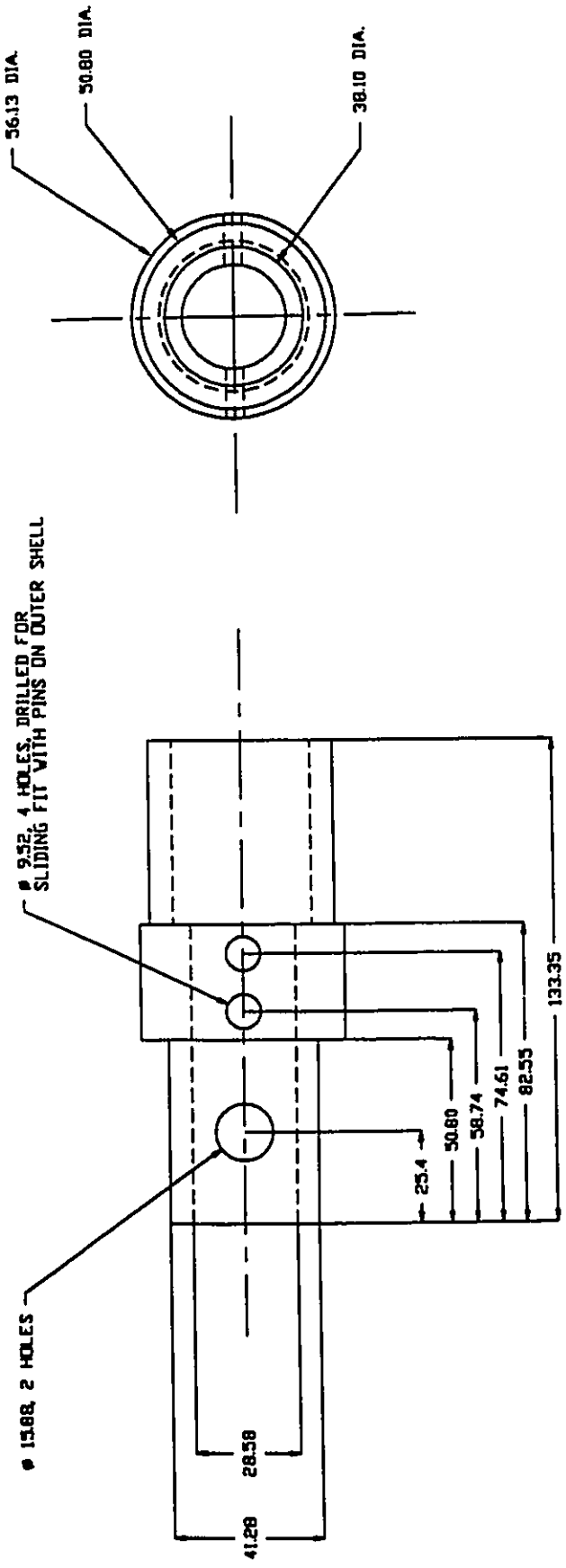
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Appendix A

End Fitting Design Drawings



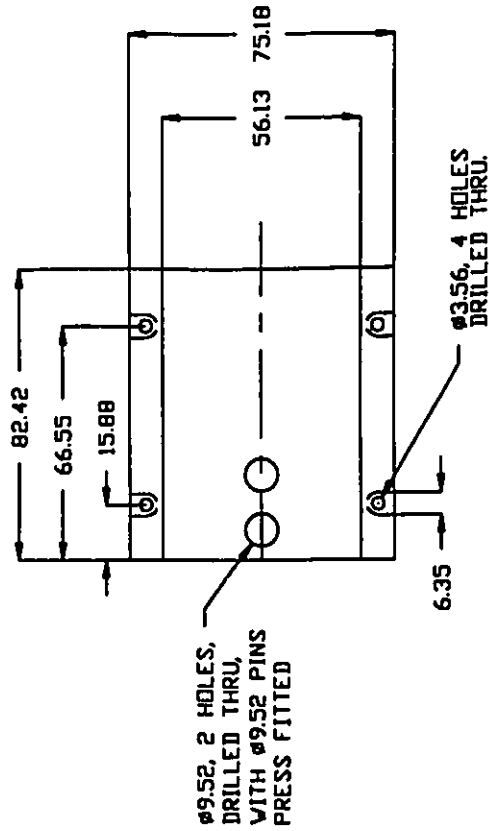
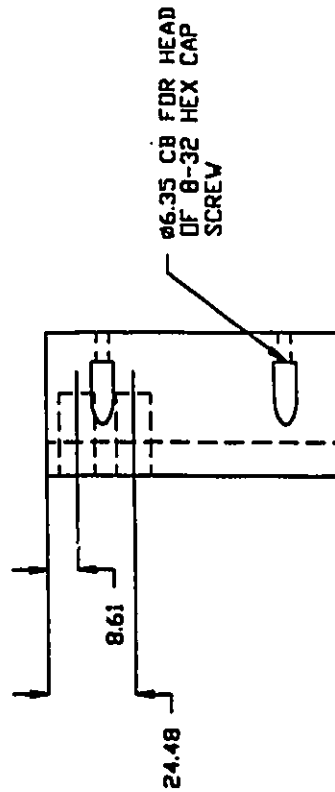
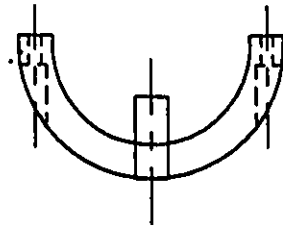
University of Ottawa: Mech. Eng.			
SCALE: NTS	TOLERANCE: ± 0.001 INCHES	MATERIAL: ALUMINUM	
PART NAME: INTERNAL TAPERED ENDFITTING			
DRAWN BY: T. DICKSON			
QUANTITY: 2		PART NO: EFIT1	



University of Ottawa: Mech. Eng.

NOTE:
SOME LINES OMITTED FOR CLARITY

SCALE: NTS	TOLERANCE: ± 0.02 mm	MATERIAL: MILD STEEL
PART NAME: ALIGNMENT PLUG		
DRAWN BY: T. DICKSON		
QUANTITY: 2	PART NO: EFT12-1	



University of Ottawa: Mech. Eng.

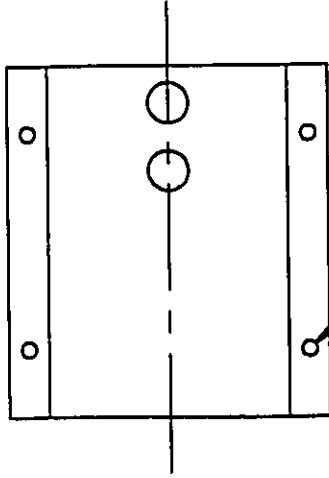
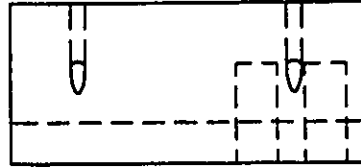
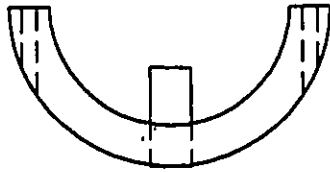
SCALE: NTS TOLERANCE: ± 0.02 mm MATERIAL: MILD STEEL

PART NAME: EXTERNAL SHELL (1 OF 2)

DRAWN BY: T. DICKSON

QUANTITY: 2 PART NO: EF112-2A

NOTE: THE EXTERNAL SHELL IS COMPRISED OF TWO SEPARATE HALVES. THE SECOND HALF IS IN EF112-2B.



4 HOLES DRILLED AND TAPPED
FOR 8-32 HEX HEAD CAP SCREWS

NOTE:
ALL RELEVANT DIMENSIONS INCLUDING HOLE LOCATIONS
APPEAR IN EFIT2-2A. EFIT2-1, EFIT2-2A AND EFIT2-2B
COMPOSE ONE ASSEMBLY. THE 9.52 mm PINS ARE
TO HAVE A SMOOTH SLIDING FIT WITH THE 9.52 mm
HOLES OF THE ALIGNMENT PLUG.

University of Ottawa: Mech. Eng.

SCALE: NTS

TOLERANCE:
 ± 0.02 mm

MATERIAL:
MILD STEEL

PART NAME: EXTERNAL SHELL (2 OF 2)

DRAWN BY: T. DICKSON

QUANTITY: 2

PART NO:
EFIT2-2B

Appendix B

Material Properties For Hercules AS4/3501-6

Summary of Material Property Values For AS4/3501-6

Material Property	
0° Tensile Modulus, E_{11T}	139.7 GPa
0° Tensile Strength, X_T	2142.9 MPa
0° Compression Modulus, E_{11C}	116.5 GPa
0° Compression Strength, X_C	820.5 MPa
90° Tensile Modulus, E_{22T}	8.2 GPa
90° Tensile Strength, Y_T	43.4 MPa
90° Compression Modulus, E_{22C}	Same as 0° Tension
90° Compression Strength, Y_C	206.0 MPa
Poisson's Ratio, ν_{12}	0.3

Appendix C

Manufacturing Summary Table

Specimen	Mandrel	Winding Comments	Utilization	Vacuum Bagging Notes	Comments
PR/R1	3 Pc. Steel	-178N tension	-Manufacturing Study -Residual Str. (RESID_1)	-Standard Bagging technique	-Very poor surface finish -Helical pattern left by shrink tape
PR/R2	3 Pc. Steel	-17.8N tension	-Manufacturing Study -Residual Str. (RESID_2)	-Double shrink tape technique high tension/preheated tape	-Same as above
PR/R3	3 Pc. Al.	-17.8N tension	-Manufacturing Study -Residual Str. (RESID_3)	-Double shrink tape technique, high tension/preheated tape	-Same as above
PR/R4	3 Pc. Al.	-17.8N tension	-Manufacturing Study -Residual Str. (RESID_4)	-Double shrink tape technique, high tension tape	-Same as above
PR_1	1 Pc. Steel	-17.8N tension	-Manufacturing Study	-Single shrink tape technique	-Helical pattern even deeper than for the double shrink tape method
PR_2	1 Pc. Steel	-17.8N tension	-Manufacturing Study	-Single Al. foil layer on gauge section under shrink tape layer	-Foil wrinkles under pressure and vacuum -Leaves poor surface finish
PR_3	1 Pc. Steel	-17.8N tension	-Manufacturing Study	-Double Al. foil layer on gauge section under shrink tape layer	-Same as above
PR_4	1 Pc. Steel	-17.8N tension	-Manufacturing Study -Microscopic Analysis	-203mm long rolled Al. sleeve on gauge section under shrink tape layer	-Deep axial seam from sleeve overlap -Deep circumferential seams at beginning of built up region at end of gauge section -Microscopy shows good consolidation and

PR_5	1 Pc. Steel	-17.8N tension	-Manufacturing Study -Preliminary In-plane Shear Spc.	-203mm long rolled Al. sleeve on gauge section under shrink tape layer	-Edges of sleeve sanded until sharp -Generous axial seem overlap -Resulting seems too deep -Visible surface voids -C-Scan detected no voids in gauge section away from axial seem
PR_6	1 Pc. Steel	-17.8N tension	-Manufacturing Study -In-plane Shear Spc.	-203mm long rolled Al. sleeve on gauge section under shrink tape layer	-Edges of sleeve sanded until sharp -3.2mm axial seem overlap -Holes drilled in sleeve to relieve trapped air -axial and built up end seems too deep -holes have no effect on surface voids -C-Scan detected no voids in gauge section away from axial seem
PR_7	1 Pc. Steel	-17.8N tension	-Manufacturing Study	-203mm long rolled Al. sleeve on gauge section under shrink tape layer	-Butt joint used instead of over-lap, thin film place beneath joint to prevent resin from escaping. -film wrinkled which creased tube surface -Tube accidentally broken during post cure machining operation
TUBE_0	1 Pc. Steel	-17.8N tension	-Manufacturing Study -In-plane Shear Spc.	-216mm long rolled Al. sleeve on gauge section under shrink tape layer with 6.4mm flared ends	-6.4mm flare used to eliminate seem at built up end -butt joint used -C-Scan detected no voids in gauge section away from axial seem -Acceptable quality tube
TUBE_1	1 Pc. Steel	-17.8N tension	-Manufacturing Study -In-plane Shear Spc.	-254mm long rolled Al. sleeve on gauge section under shrink tape layer with 25.4mm flared ends	-25.4mm flare greatly improves transition from gauge section to built up end -C-Scan detected no voids in gauge section away from axial seem -Acceptable quality tube

TUBE_2	1 Pc. Steel	-17.8N tension	-Manufacturing Study -In-plane Shear Spc. -SEM Study of Failure	-254mm long rolled Al. sleeve on gauge section under shrink tape layer with 25.4mm flared ends	-Scarf joint used to give improved axial seem -C-Scan detected no voids in gauge section away from axial seem -Good quality tube
TUBE_3	1 Pc. Steel	-17.8N tension	-Manufacturing Study -In-plane Shear Spc.	-254mm long rolled Al. sleeve on gauge section under shrink tape layer with 25.4mm flared ends	-No change from Tube_2 -C-Scan detected no voids in gauge section away from axial seem -Good quality tube
TUBE_4	1 Pc. Steel	-17.8N tension	-Manufacturing Study	-254mm long rolled Al. sleeve on gauge section under shrink tape layer with 25.4mm flared ends	-No change from Tube_2 and Tube_3 -C-Scan detected 1 large void in gauge section -Tube unacceptable for determination of failure levels
TUBE_5	1 Pc. Steel	-17.8N tension	-Manufacturing Study	-254mm long rolled Al. sleeve on gauge section under shrink tape layer with 25.4mm flared ends	-No change from Tube_2 and Tube_3 -C-Scan detected many small voids in gauge section near one end -Tube unacceptable for determination of failure levels
TUBE_6	1 Pc. Steel	-17.8N tension	-Manufacturing Study -Microscopic Analysis	-254mm long sil. shim stock (.15mm thick) sleeve on gauge section under shrink tape layer with 25.4mm flared ends	-Heat used to pre-shrink the shrink tape -C-Scan detected many small voids in centre of gauge section -Tube unacceptable for determination of failure levels -Used to verify existence of inter- laminar voids -Microscopy also used to verify minimal seem depth

TUBE_7	1 Pc. Steel	-17.8N tension -Hot wound and compacted	-Manufacturing Study -In-Plane Shear Spc.	-254mm long stl. shim stock (.1mm thick) sleeve on gauge section under shrink tape layer with 25.4mm flared ends	-Heat used to pre-shrink the shrink tape -C-Scan detected no voids anywhere in the gauge section -Minimal axial seem -very consistent dimensionally -Excellent quality tube
TUBE_8	1 Pc. Steel	-17.8N tension -Hot wound and compacted	-In-Plane Shear Spc.	-254mm long stl. shim stock (.1mm thick) sleeve on gauge section under shrink tape layer with 25.4mm flared ends	-Heat used to pre-shrink the shrink tape -C-Scan detected no voids anywhere in the gauge section -Minimal axial seem -very consistent dimensionally -Excellent quality tube

Appendix D

Upgrade and Operation of A.T.M. Triaxial Test Machine

D.1 Upgrade of A.T.M. Testing Machine

In its original configuration the A.T.M. triaxial hydraulic test machine could combine torsion, bending and axial (compressive or tensile) loading for rotational or fatigue testing. The maximum torque load possible was 170 Nm while the axial capacity was 130 KN. The bending load, which is measured by the angle that the specimen makes with the load axis, could be up to 15°. The rotational testing range was between 30 and 1500 rpm and the maximum fatigue load was 10 Hz. A schematic of the original configuration appears in Figure D.1.

D.1.1 Lower Assembly Of Original Configuration

The lower hydraulic torque motor serves as the means by which a specimen is rotated at a specific speed (rpm). The rotations per minute are monitored using a gear attached to the lower part of the hydraulic torque motor. One tooth has a magnetic signature which is registered by a magnetic pick-up (not shown in Figure D.1). The torque motor is attached to a bearing housing which in turn supports the specimen grips. The grips are a split collar arrangement which must be tightened using hex-head cap screws after the specimen is positioned. The motor is suspended from an aluminum base plate which is mounted on four 50.8 mm diameter stainless steel posts which are attached at the base plate corners. These posts sit on and are attached to a second plate which is part of the bending control assembly.

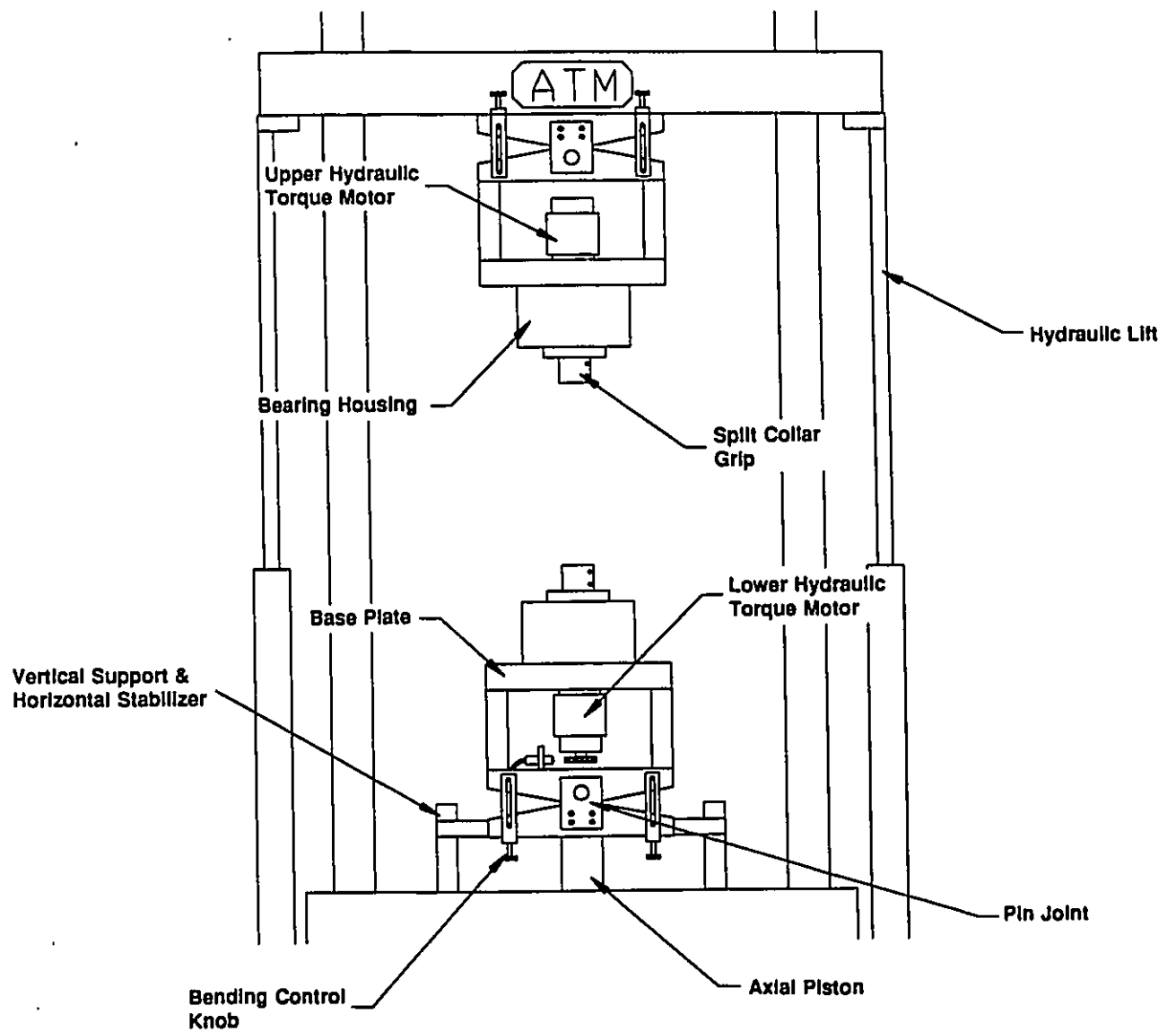


Figure D.1 Original ATM triaxial test machine configuration.

The bending control assembly consists of two aluminum plates, two pin joints, two manual bending controllers and a precision level. The upper plate (which supports the motor housing) has a triangular shaped bottom and sits on an identical plate such that the two triangular surfaces are in contact. The two plates are joined together by two pin joints, one in the front and one in the rear. A pin housing is attached rigidly to the centre of the lower plate. The pin which rotates freely in the housing connects the pin housing to the upper plate. This rotational freedom gives the A.T.M. machine its bending capability. Two manual bending controllers are mounted in the front. By turning the knobs on the two controllers bending can be imposed on the specimen. The degree of bending is measured using a precision level mounted on the lower torque motor housing (not shown in Figure D.1).

The axial load capability is supplied by the lower assembly which is mounted on an axial piston, which is attached to an hydraulic actuator. The actuator has two control modes: load control and displacement control. Four 50.8 mm diameter stainless steel horizontal stabilizers are attached to the lower bending plate; two on either side of the plate. Each stabilizing shaft is mounted on either side of a stainless steel vertical support. This system ensures that the lower assembly can support torque loads without twisting.

D.1.2 Upper Assembly of Original Configuration

The upper assembly is suspended from the horizontal crosshead. It is identical to the lower assembly with two exceptions. The upper torque motor is used to resist the action of the lower torque motor thus generating torque loading of the specimen. Secondly, the entire assembly is rigidly attached to the crosshead which resists any torque in the system. The crosshead moves up and down by means of hydraulic actuators.

D.1.2 New Configuration

For this study the original configuration had to be upgraded for torque loads ranging from 560 Nm to 1700 Nm. To accomplish this extensive modifications were made to the lower

assembly. Furthermore, the original upper assembly was removed in favour of a new system which to support static torque testing. During pure torque testing the specimen can not experience bending loads. With this in mind the existing grips were replaced with universal joints which will ensure that any bending that occurs does not load the specimen. A schematic of the new configuration appears in Figure D.2.

D.1.2.1 Modified Lower Assembly

The most significant change required by the high load requirements was the addition of a torque multiplier (gearbox) to the existing lower torque motor. The motor, by itself, can generate a maximum of 560 Nm of torque while with the 10:1 torque multiplication supplied by the gearbox a maximum of 1700 Nm was possible. The motor and gearbox are, however, now attached permanently which means that when necessary another torque motor will be needed to reassemble the machines original configuration.

The addition of the gearbox to the existing torque motor changed the housing requirements for the motor assembly. As a result, the posts which supported the motor support plate had to be replaced with longer ones. The increase in torque exceeded the torque load carrying capacity of the existing lower assembly supports. Additional stabilizers were therefore attached to the upper surface of the motor support plate. The torque is resisted by the crosshead posts. Finally, the bearing housing and specimen grips were deemed inappropriate for static torque testing and thus they were removed. In their place universal joints were attached directly to the torque motor output shaft via a coupling fixture (see end of Appendix D for individual drawings). The universal joints reduce the possibility of bending loads being transmitted to the specimen.

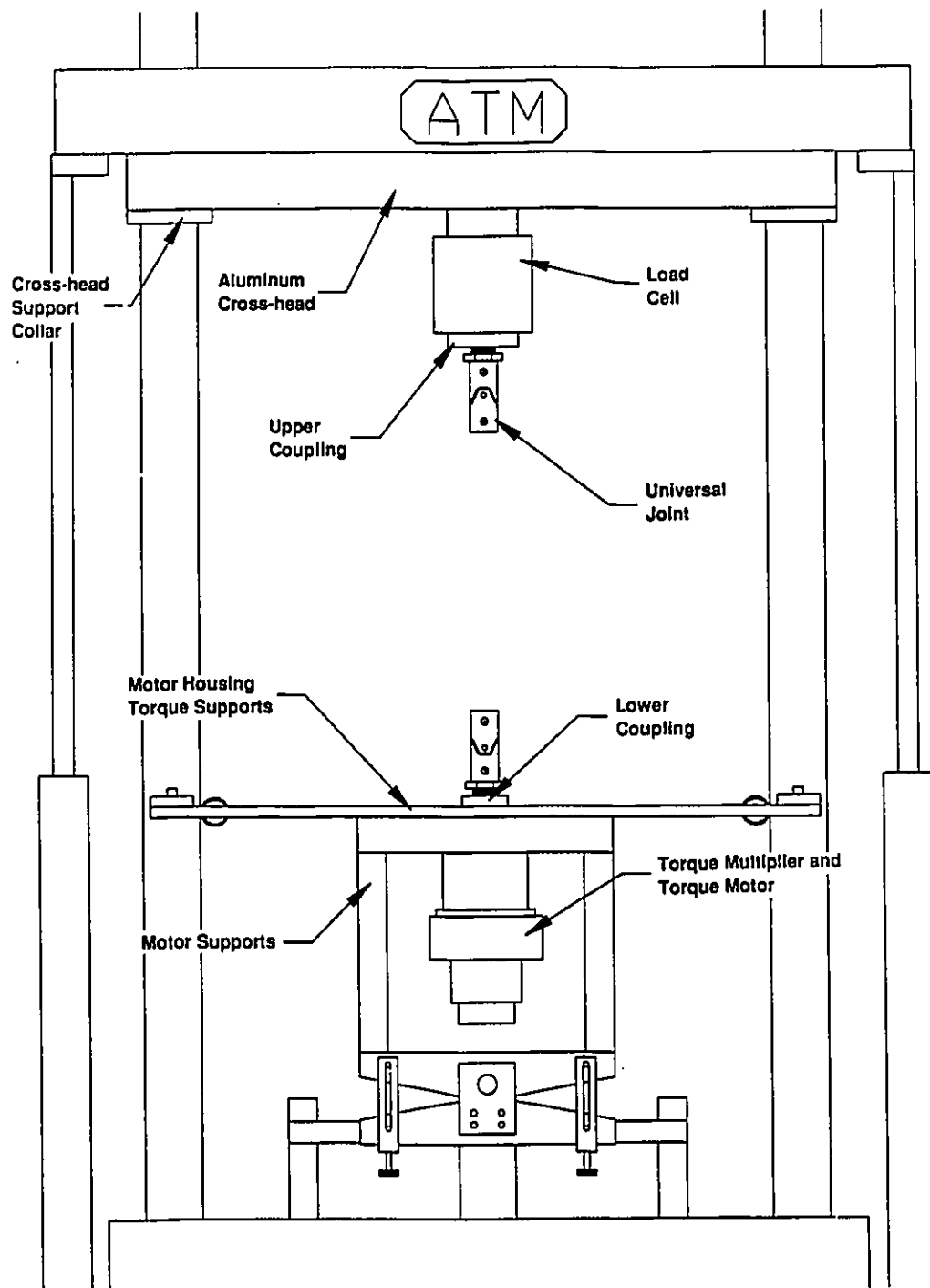


Figure D.2 Modified ATM machine.

D.1.2.2 Modified Upper Assembly

As mentioned the tests to be conducted were static torque tests. This required that either the upper or lower assembly be fixed rigidly. Since the existing upper assembly was free to rotate and could not easily be fixed and reinforced to support the new loading conditions it was removed entirely. At first it was thought that the existing crosshead could be used as the fixed support, however, too many hydraulic fittings and connections would have to be removed. Instead, a new aluminum crosshead was installed which could withstand torque loads of 1700 Nm. Aluminum collars which were attached to the existing crosshead posts were used to hold the new crosshead in place. Finally, a new torque load cell was installed on this crosshead along with a universal joint coupling and a universal joint to hold the specimen. Drawings of the new fixtures etc. fabricated at I.A.R. appear at the end of Appendix D.

D.2 Use Of The ATM Machine and Testing Procedures

Since the ATM testing machine was not originally designed to support static testing of any kind, procedures for conducting this type of testing were not established. For this reason considerable testing was done using a reusable 38 mm diameter aluminum rod machined to act as a specimen. Next, two preliminary composite tubes (PR 5 and PR 6) were used to fine tune the procedure developed using the aluminum specimen. Before dealing with the particulars of the testing, however, important machine features will be explained and any relevant definitions provided.

D.2.1 Important Machine Features and Definitions

D.2.1.1 Calibration Check Switch and Dial

The calibration check switch is used to display and change the calibration voltage. The value of this voltage is determined during the calibration of the load cell prior to certification of the machine for use in testing. The torque and axial control systems each have a calibration

check switch and vernier dial. To check this value the hydraulic oil supply must be turned off and the voltage display must be turned to the torque setting (setting no. 4). If the supply is on when the calibration switch is on the calibration voltage will be translated into a signal in the system which is interpreted as a command. This command becomes a torque load which could either damage a specimen or the machine itself. The calibration voltage is changed using the vernier dial which is beneath the calibration check switch. This value should not be changed unless the load cell is being calibrated.

D.2.1.2 Torque Gain Switch

The gain switch dictates the amplification of the load cell signal. A vernier dial is supplied so that this value can be changed, however, changing this value is not recommended since its value was also determined during calibration of the load cell. The torque and axial control systems each have a gain setting. Increasing the amplification of the load cell signal, generally, would increase the error in the system and will make the system respond more quickly to changes in load. When there is no specimen loading the system the error becomes the only feedback signal. If high, this feedback can cause the system to be unstable when positioning the grips manually in the load control mode. The manufacturer or qualified technician should be consulted prior to the adjustment of the system gain settings.

D.2.1.3 Load Cell Zero Dial

The load cell zero vernier dial is used to adjust the voltage reading of the load cell when there is no load on the system. It should not be necessary to change this value but occasionally if the systems sits unused for long periods of time it might become necessary to change this value. It should be noted however that the voltage output of the load cell can never be zero. There is a small error (approx. 0.0049 Volts) which will always appear even when there is no specimen being loaded.

D.2.1.3 System Control Panel

There are two control panels on the A.T.M. triaxial test machine. The upper panel is used to switch on the crosshead pressure and the torque pressure while the lower panel is used to turn on the pressure for the axial actuator. The power on/off switch, hydraulic off switch and the high and low pressure on switches are on the system control panel. In its present configuration the low pressure switch is the only one which switches on the hydraulic pressure for both panels. Status indicator lights are also on the control panels to indicate the type of pressure in use and any errors which might occur.

D.2.1.4 Voltage Display

The voltage display is used to display the voltage of the load cell during specimen loading. Only one signal can be displayed at one time so that a dial is supplied to switch from one load case to the next. Presently channel one is for the axial load and channel four is for the torque load. Channels two and three were not used for this study but indicate axial position and output shaft velocity.

D.2.1.6 Set Point Vernier Dial

The set point vernier dial is used to manually manipulate the position of and load on the specimen at a given time. It is also used to position the lower universal joint during the mounting of the specimen in the machine. The torque set point manually controls the flow of oil through the torque motor which rotates the universal joint in either the counter-clockwise or clockwise directions. The axial set point controls the flow of oil to the axial actuator which in turn moves the entire lower assembly up and down.

D.2.1.7 System Control Modes

On most modern test machines both displacement and load control features are available

for use during testing. In the case of the A.T.M. machine the axial actuator can be controlled in either load or displacement mode. The torque motor, however, is only controllable in load control. A channel switch is used to move from displacement to control mode but it is essential that the hydraulic pressure be turned off when moving from displacement control to load control. Otherwise a technique must be developed to electronically zero the displacement control after any positioning of the axial actuator has been completed. If one simply switches from displacement control to load control without zeroing, the actuator will not maintain its position but will jump to a new position dictated by the present value of the set point vernier dial. No such procedures were developed for this study due to a lack of time. It should be noted that positioning of the lower assembly is much easier with displacement control because there is always feedback in the system supplied by a L.V.D.T., however, in load control the load cell provides the feedback. When the specimen is not mounted at both ends there is no feedback and the system is in open loop control which can be unstable.

D.2.1.8 Span Vernier Dial

The span vernier dial dictates what percentage of the system capacity will be used for testing using the function generator (see below). The capacity of the machine and the capacity of the load cell are linked and the maximum calibrated voltage of the load cell is presently ± 7.5 Volts. Therefore setting the torque span vernier at 3.0 (30% of full scale) will yield a maximum specimen torque load of 170 Nm.

D.2.1.8 Function Generator

The function generator provides the test machine with the capacity to load a specimen automatically using one of eight predefined load cases. The Run/Hold switch acts to turn the generator on and off while the reset button is used to bring the load to zero if the Run/Hold switch is moved from run to hold. To set the function generator the Run/Hold switch must be in the Hold position. The loading case and loading rates are then selected using the appropriate dials. The span vernier is then set to the appropriate maximum load and the reset button on the function

generator is pushed. This fills the function generator command buffer with the new commands replacing ones that might already be there. To load the specimen one moves the Run/Hold switch to the Run position.

D.2.1.9 Out Of Limit Settings

The out of limit settings are used to automatically turn off the hydraulic supply when the specimen is broken or the load exceeds specified settings for some reason. Both maximum and minimum settings are available with both positive and negative loading voltages. A vernier dial determines the voltage value in each case while a +/- switch determines the sign of the maximum or minimum value. A switch is also supplied to render the hydraulic shut off inactive as necessary. Indicator lights will come on in either case to signal when the system has exceeded the limit settings. To re-enable the system after the hydraulics have been automatically turned off the function generator must first be set to the Hold position and the function generator reset button must be pushed. If the function generator is not in use the load must be manually set to zero using the set point vernier dial. Secondly, the out of limit reset button must be pushed.

D.2.1.10 Pressure Reducing Valve

The maximum operating pressure capacity of the torque motors supplied with the A.T.M. test machine is 17225 KPa which is 3445 KPa below the line pressure of the hydraulic supply in the M-3 building at NRC. To ensure that the correct pressure was maintained in the test machine system a variable pressure reducing valve was installed between the building's supply line and the A.T.M. system manifold. This valve gives the A.T.M. some added flexibility in that it can operate at any pressure.

D.2.2 Test Procedures

The following is a description of relevant aspects of using the A.T.M. machine. A step by step description of the procedure used for testing appears at the end of Appendix D.

D.2.2.1 Mounting Of The Specimen

Prior to using the A.T.M. machine, the hydraulic supply must be on for at least 60 minutes with a strong reusable specimen mounted in it. The system operates differently when the oil is cold than when it is warm. In particular the torque motor will initially rotate at a relatively high velocity. The torque set point must be used to manually stop this rotation, but this changes the zero setting of the torque control. The function generator will see the new setting as a load and when the function generator reset button is pushed any specimen in the machine will experience a sudden load. To eliminate this possibility from happening mount a strong aluminum or steel specimen in the machine after the set point has been used to stop the torque motor rotation. Once this is accomplished move the set point back to its original zero location (5.0 on the set point vernier dial). As the oil in the system warms up, the load on the reusable specimen will dissipate to zero after approximately 60 minutes. Below is a suggested procedure for mounting specimens into the A.T.M. triaxial test machine.

Before attempting to mount the specimen in the test device the crosshead is raised so that ample room is available between the upper and lower universal joints. To do this the two crossheads must be connected (clamping was sufficient in this case) and then the hydraulic unlock valve on the crosshead control is opened. Next the up valve is opened slowly while the two crossheads rise together. When this is done both valves are closed. The upper end fitting of the specimen is then positioned inside the universal joint so that the hole of the end fitting lines up with the hole of the universal joint. A 12.7 mm diameter pin is then pushed through the lined up holes to secure the specimen in place.

Two steps are required to place the lower end fitting of the specimen in the lower universal joint. Firstly, the crosshead is lowered by opening the unlock valve and then the lower valve of the cross head control. The crosshead is lowered slowly until the end fitting enters the lower universal joint. (NOTE: Great care should be taken while doing this procedure so that as the end fitting enters the universal joint it does not catch on the internal bore. The end fitting

should be oiled to facilitate easy introduction of the end fitting into the universal joint.) Once, approximately, 15 mm of the end fitting is inside the universal joint the crosshead should be stopped and then the unlock valve should be closed.

At this point the strain gauge leads should be connected to the data acquisition system. This can be done after the specimen is fully mounted, however, if there are problems on any of the systems channels it could take considerable time to track down the problem increasing the risk of accidentally loading the specimen prior to testing. In addition the leads can be secured so that they are out of the way while the specimen mounting is completed.

The second step employs the torque motor and axial actuator in load control. Displacement control is more steady but as has been mentioned there is no displacement control for control of the torque motor and procedures were not developed to use the axial displacement control. Of further note is that the variable pressure reducing valve was set at 10335 KPa rather than the maximum 17225 KPa. The response of the machine is more sluggish at the lower pressure under manual control but it is easier to position the lower universal joint when the actuators do not respond to small changes in the set point vernier position. In addition the specimens are not expected to require more than 8268 KPa for failure so 10335 KPa was a sufficient pressure for the entire test cycle. By lowering and raising the lower assembly with the axial actuator and simultaneously rotating the universal joint with the torque motor the universal joint and specimen end fitting holes were lined up and the pin inserted. It is important that prior to securing the lower end fitting in place that the function generator reset button is pushed if the function generator is to be used for loading. This resets the position of zero load of the function generator.

D.2.2.3 Testing Specimens

Once the specimen is mounted the set point should not be touched again except if the system has preloaded the specimen in some way (this is unlikely but possible). To check this simply turn the voltage display to the axial position and then the torque position. If there is a

slight voltage on either axis carefully turn the set point vernier until the zero value is reached. Since both axial and torque actuators are in load control there should be no need to worry about the set point again.

After the data acquisition system is prepared the specimen is ready for load introduction. The function generator load setting was turned to number three which is a trapezoidal wave form. No ramp loading exists as the machine was originally designed for dynamic fatigue testing. The trapezoidal wave form is divided by the function generator into four separate segments. The first ramp, the first dwell period, the second ramp and the second dwell period. The operator controls the duration of each of these segments so that if the first ramp is given a long enough duration it can be used as a static loading ramp. Since the specimens failed prior to the end of the first loading ramp the other settings were set at arbitrary non zero values (the function generator must see non zero values for the other three segments or it does not function predictably). The duration of the first ramp was set at 1000 secs.

The torque span vernier was then set at 3.0 for a maximum loading of 2.7 Volts or 510 KPa. After this the function generator reset button is pushed so that the new commands are entered in the command buffer. Finally the Run/Hold switch was placed in the Run position and the specimen testing began. This procedure was repeated for all of the in-plane shear specimens tested.

Procedures For Operation Of The A.T.M. Triaxial Test Machine

General Operating Procedure

Starting The A.T.M. Machine

1. Before activating the A.T.M. equipment make sure the main hydraulic high pressure pump is turned on.
2. Push the main hydraulic manifold manual actuator to the on position.
3. Check the variable hydraulic pressure reducing valve to ensure that the pressure supply to the equipment is at the desired level.
4. Make sure that all set point vernier dials are at their respective zero positions (5.0 on all dials).
5. Make sure that the function run/hold toggle switch is in the hold position.
6. Using the cal. check switches, ensure that the load cell calibration constants are at the correct values (8.721 Volts for the torque portion of the load cell).
7. To engage the hydraulic power supply (H.P.S.) depress the low pressure button on the upper H.P.S. control panel. This, however only engages the manifold which supplies the torque motor and crosshead. To engage the manifold which supplies the axial actuator depress the low pressure button on the lower H.P.S. control panel.

Installation Of Specimen

1. Loosen all crosshead bolts.
2. Ensure that the static crosshead is rigidly fixed to the original cross head.
3. Loosen and lower static crosshead support collars.

4. Close the Hydraulic "Lock" valve situated on the crosshead control and open the "Unlock" valve.
5. Open the "Raise" valve and raise the crosshead so that there is sufficient room between the two universal joints to comfortably install the specimen.
6. Close the "Raise" and "Unlock" valves.
7. Slide the upper specimen end fitting into the upper universal joint and align the universal joint and end fitting pin holes.
8. Slide the load transmission pin through the universal joint and end fitting.

For the next portion of the installation procedure two people will be needed. One person will operate the controls (operator) while the other monitors (monitor) the installation of the lower specimen end fitting into the lower universal joint. The person monitoring the specimen must actually guide the specimen into the universal joint.

9. Lightly lubricate the lower universal joint and end fitting.
10. Open the "Unlock" valve and then the "Lower" valve and slowly begin to lower the two crossheads (operator).
11. Simultaneous with 9., hold the universal joint in a vertical position (monitor).
12. As the specimen is lowered the monitor must guide it into the universal joint. Jiggling the universal joint is often helpful when introducing the end fitting.

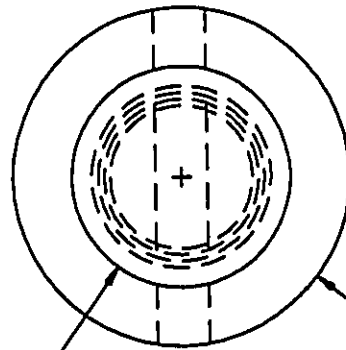
Caution: Great care must be taken during the above operation so that the end fitting does not become jammed on the universal joint. The operator must be ready to stop the crosshead at any time by closing the "lower" valve.

13. Once the end fitting is approximately halfway into the universal joint, close the "Lower" valve.

Warm Up Procedure

The operation of the A.T.M. machine differs depending on the temperature of oil. When the oil is cold and the H.P.S. is turned on the torque motor will be rotating somewhat. At least sixty minutes is required to warm up the oil so that this transient response dissipates. Sometimes, however, due to some flaw in the machines static mode design the torque motor continues to rotate. To ensure that the machine settles down completely a dummy specimen, preferably aluminum or steel should be installed in the machine.

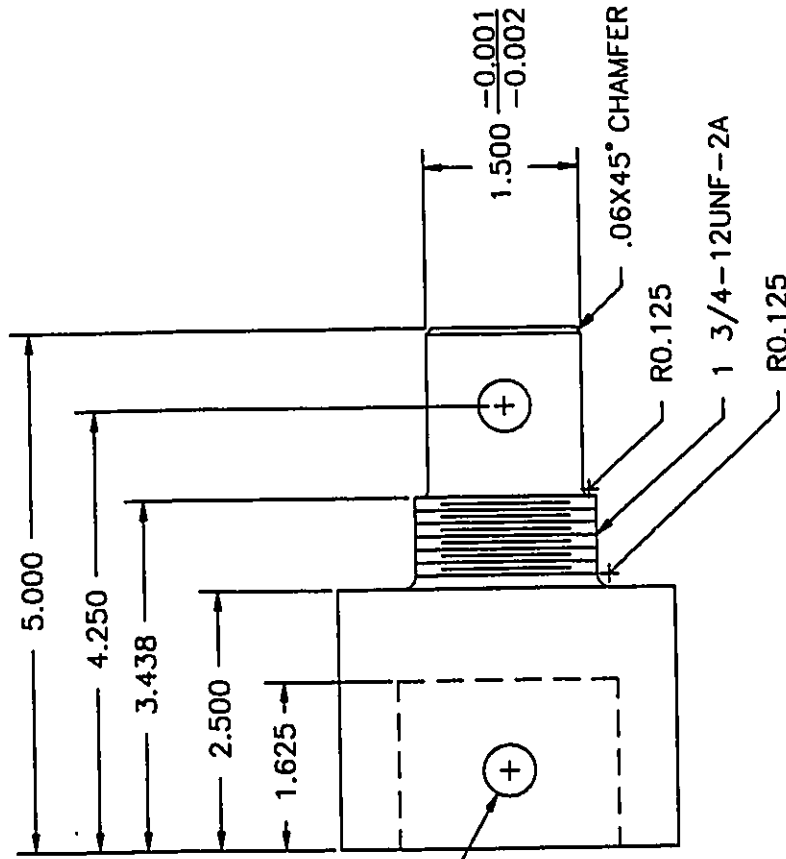
**Fixture Drawings For A.T.M.
Upgrade**



Ø 2.125

Ø 3.250

Ø 0.500
0.501
3 HOLES
DRILLED THRU



University of Ottawa: Mech. Eng.

SCALE: NTS TOLERANCE: ±0.002 MATERIAL: AISI 4340 OR EQ.

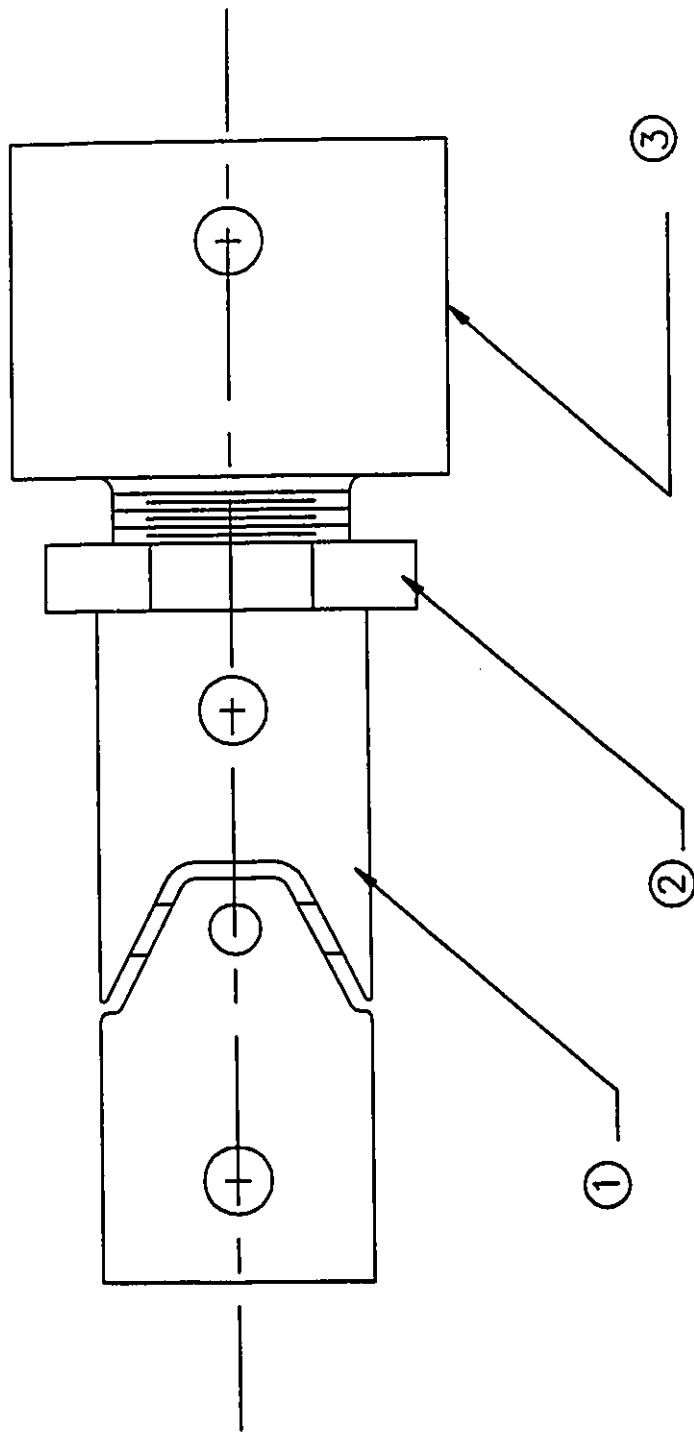
PART NAME: LOWER COUPLING

DRAWN BY: T. DICKSON

QUANTITY: 1 PART NO: ATM-1

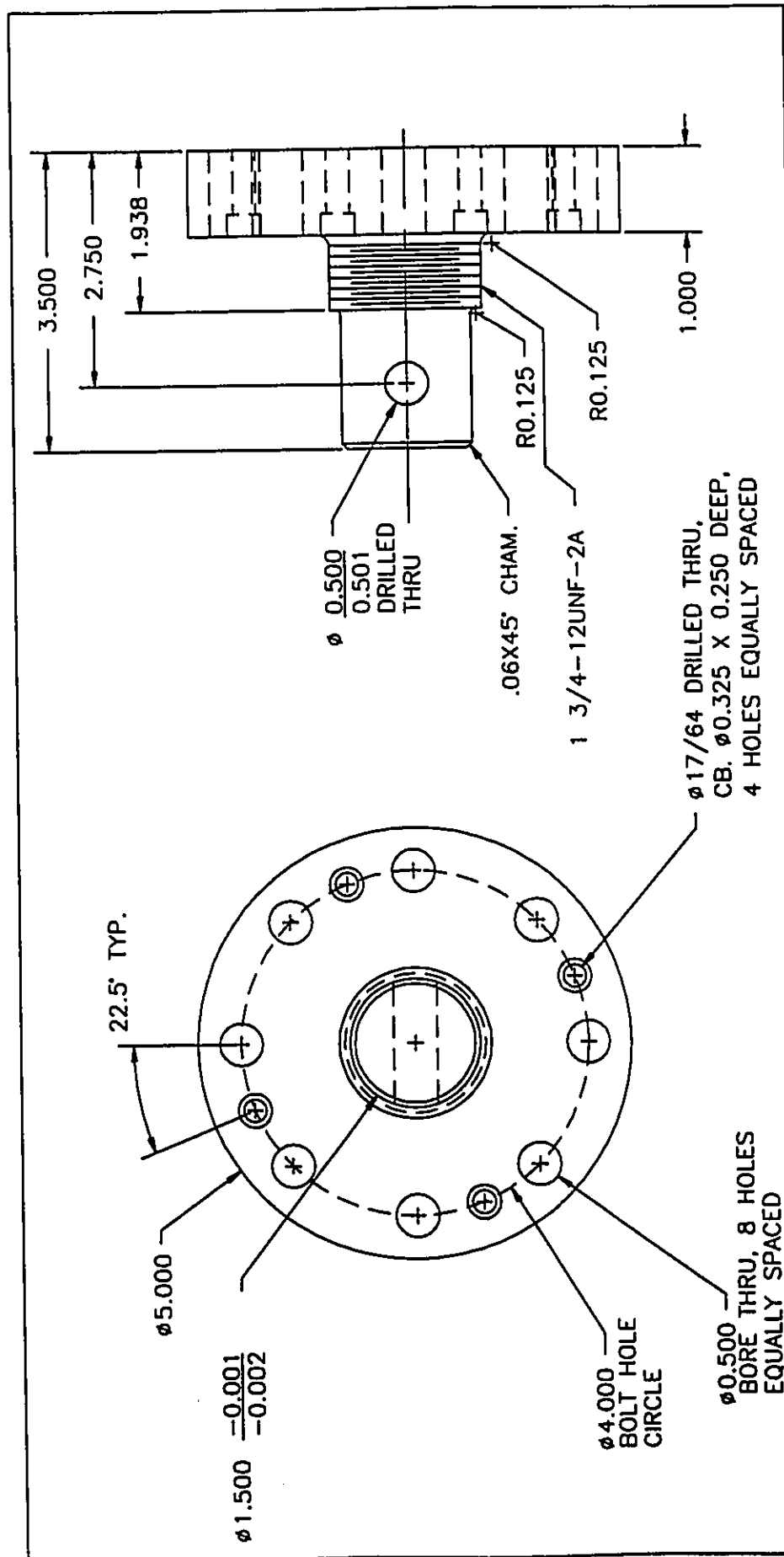
NOTES:

HARDEN TO 40-42 Rc.
SOME LINES OMITTED IN THE
INTEREST OF CLARITY.



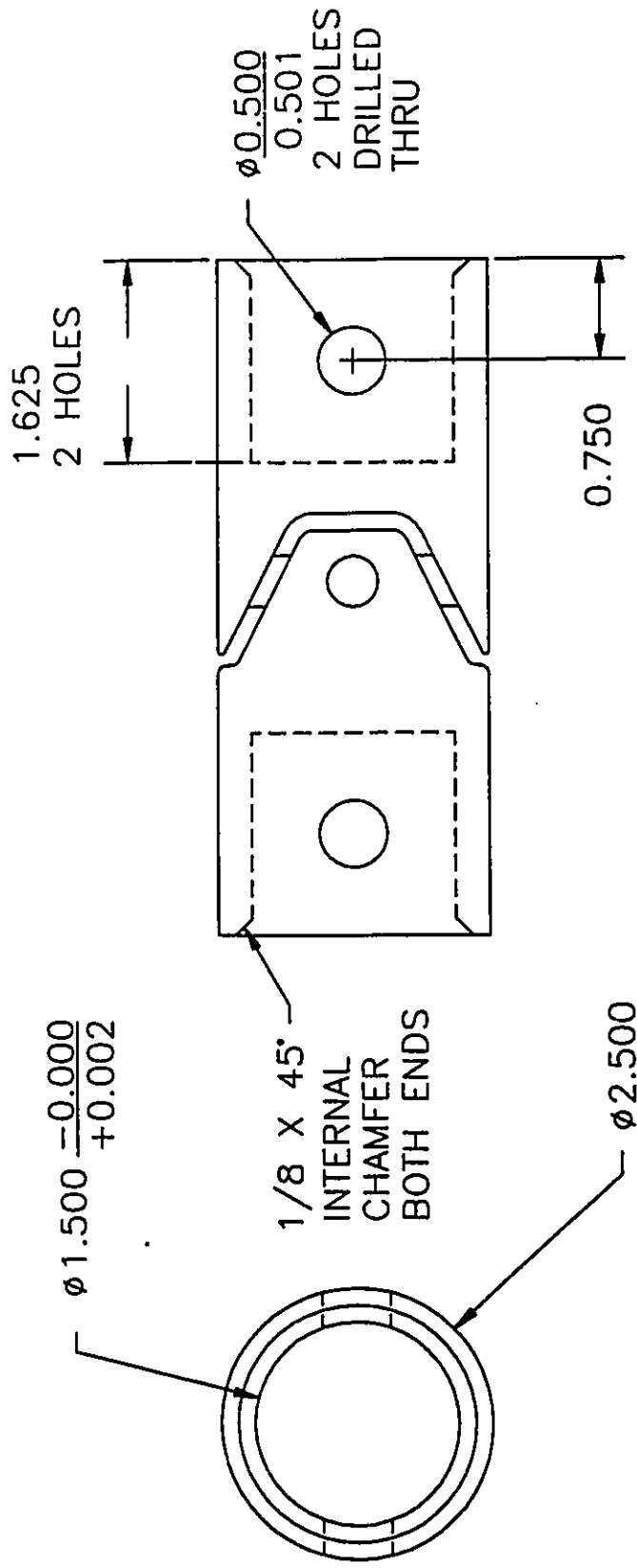
LIST OF PARTS	
1	UNIVERSAL JOINT
2	COUPLING LOCK-NUT
3	LOWER COUPLING

University of Ottawa: Mech. Eng.	
SCALE:	TOLERANCE:
PART NAME:	
DRAWN BY:	
QUANTITY:	PART NO:



University of Ottawa: Mech. Eng.			
SCALE: NTS	TOLERANCE: ± 0.002	MATERIAL: AISI 4340 OR EQ.	
PART NAME: UPPER COUPLING			
DRAWN BY: T. DICKSON			
	QUANTITY: 1	PART NO:	ATM-2

NOTES:
 HARDEN TO 40-42 Rc.
 SOME LINES OMITTED IN THE
 INTEREST OF CLARITY.



NOTES:

1. EFFECT CHANGES DETAILED TO EXISTING UNIVERSAL JOINTS.
2. LINES HAVE BEEN OMITTED IN THE INTEREST OF CLARITY.

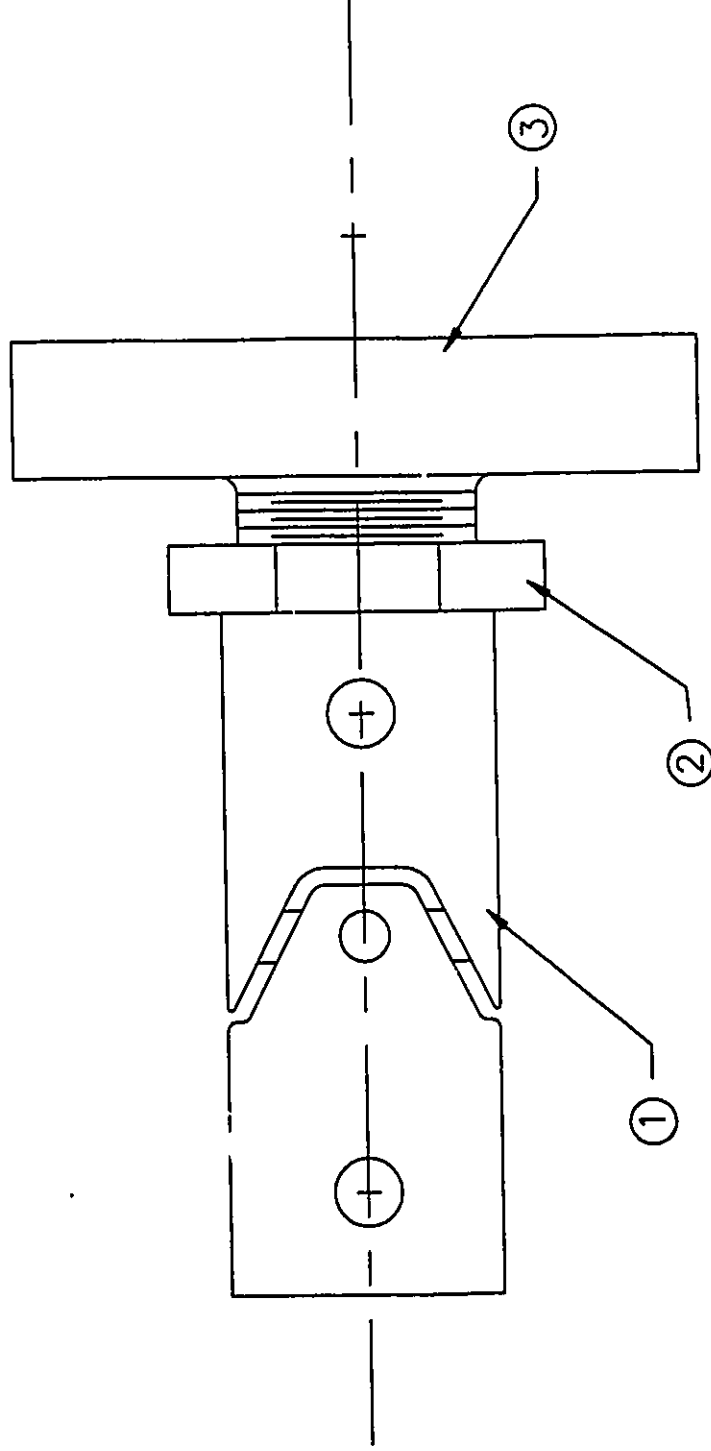
University of Ottawa: Mech. Eng.

SCALE: NTS TOLERANCE: ± 0.002 MATERIAL:

PART NAME: UNIVERSAL JOINT

DRAWN BY: T. DICKSON

QUANTITY: 2 PART NO: ATM-3



LIST OF PARTS	
1	UNIVERSAL JOINT
2	COUPLING LOCK-NUT
3	UPPER COUPLING

University of Ottawa: Mech. Eng.

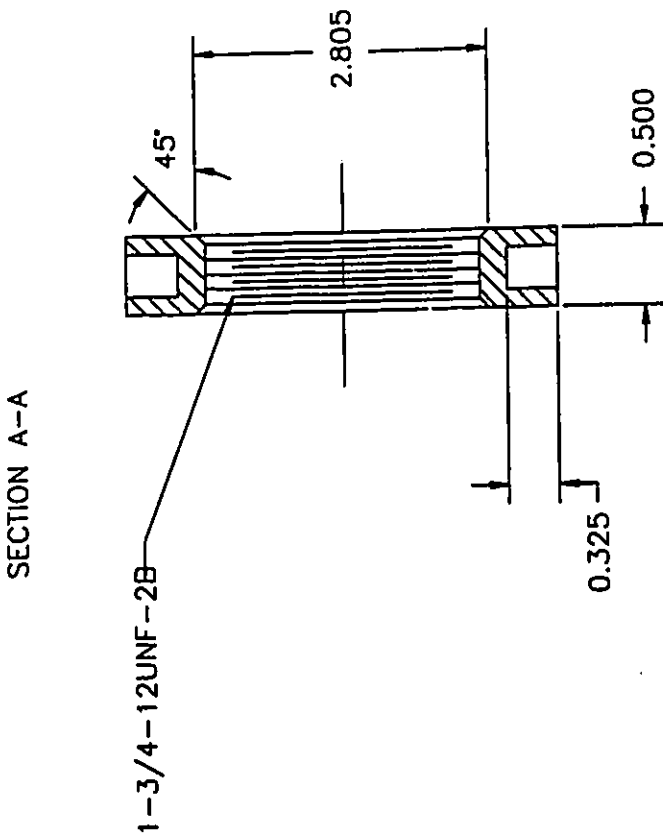
SCALE: TOLERANCE: MATERIAL:

PART NAME:

DRAWN BY:

QUANTITY: PART NO:

SECTION A-A

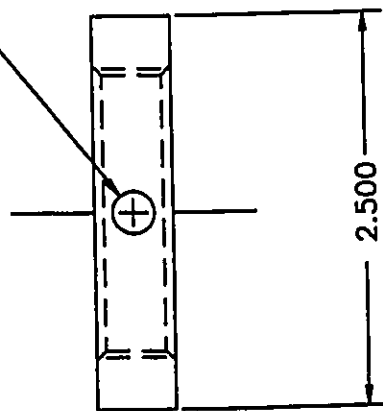
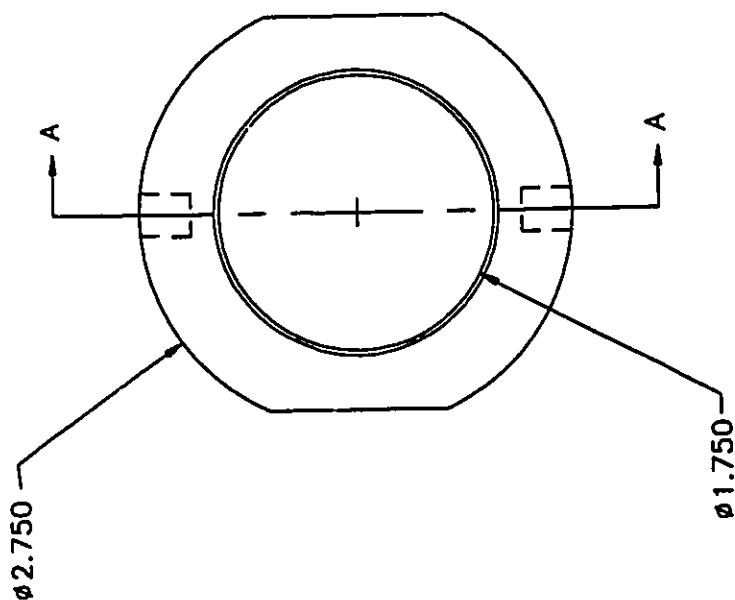


Ø0.270 2 HOLES

NOTE:

LINES HAVE BEEN OMITTED IN THE
INTEREST OF CLARITY.

HARDEN TO 30-32 Rc



University of Ottawa: Mech. Eng.

SCALE: NTS

TOLERANCE:
±0.002

MATERIAL:
AISI 4340 OR EQ.

PART NAME: ATM COUPLING LOCK-NUT

DRAWN BY: T. DICKSON

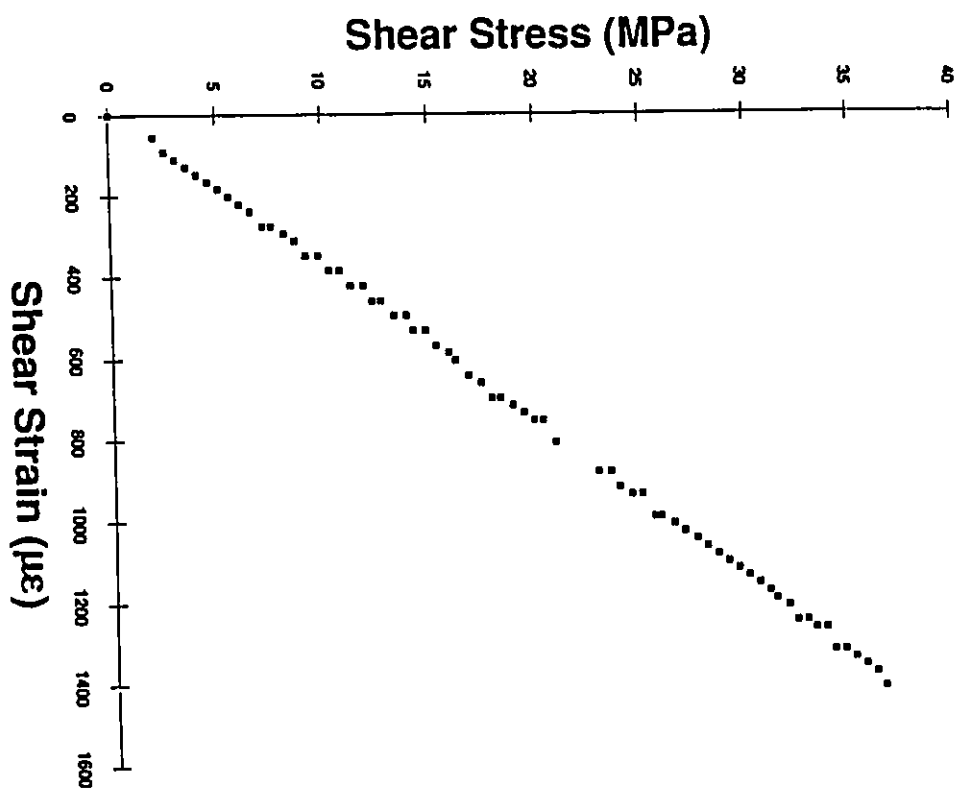
QUANTITY: 4

PART NO: ATM-4

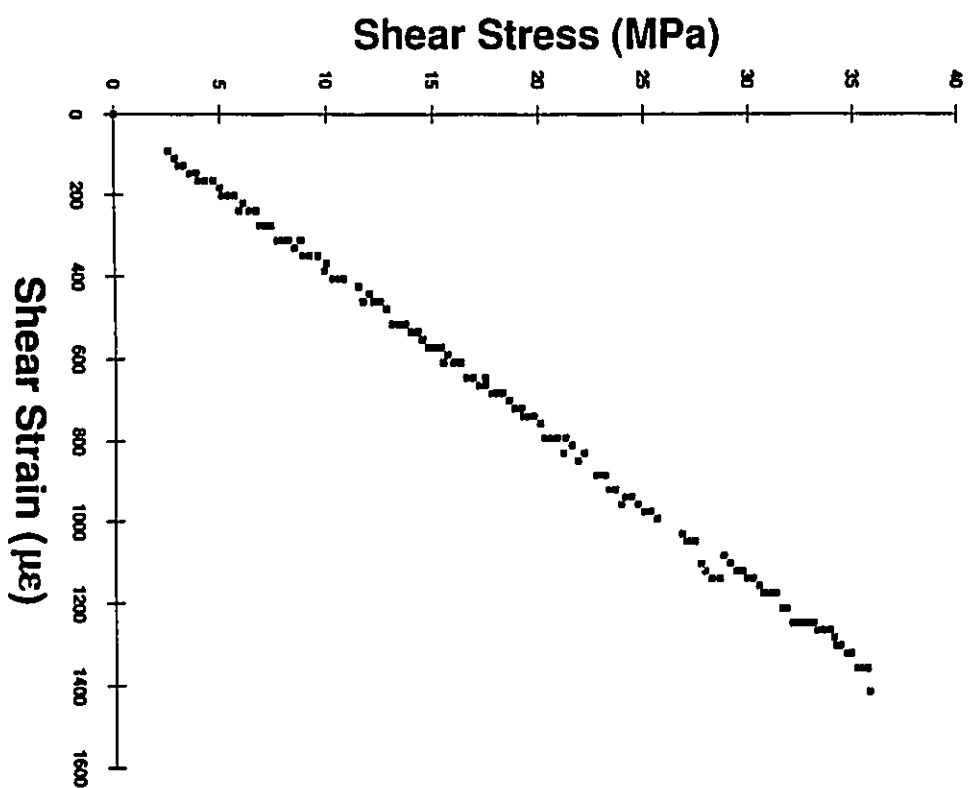
Appendix E

Additional Test Results

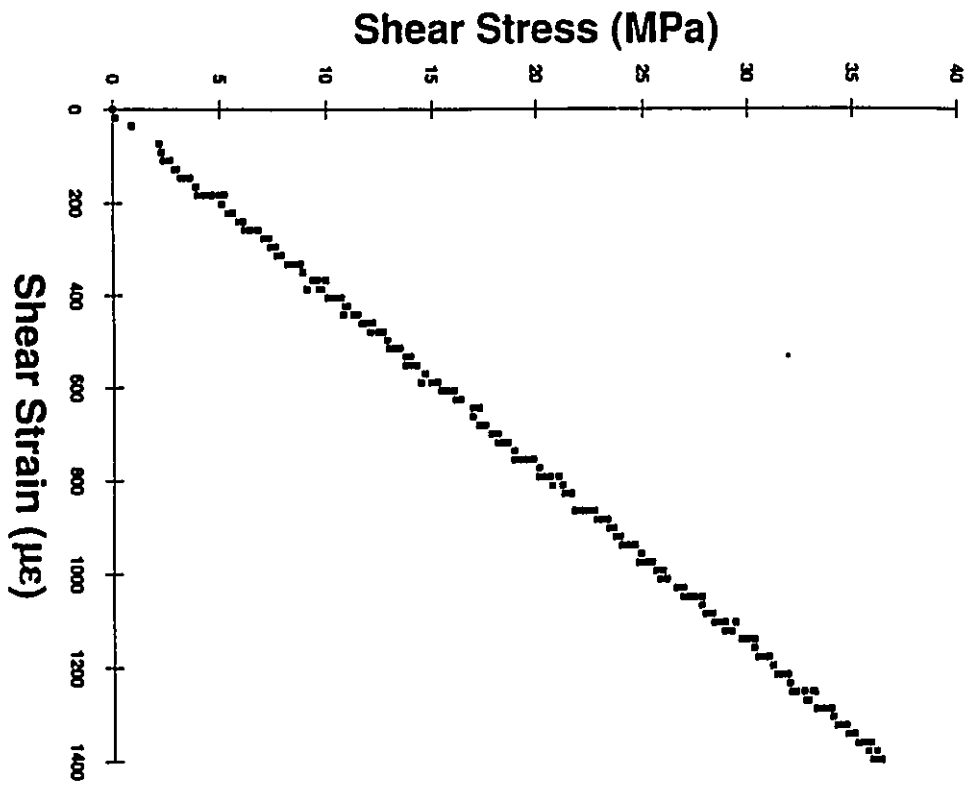
Stress-Strain Curves For Aluminum Shaft



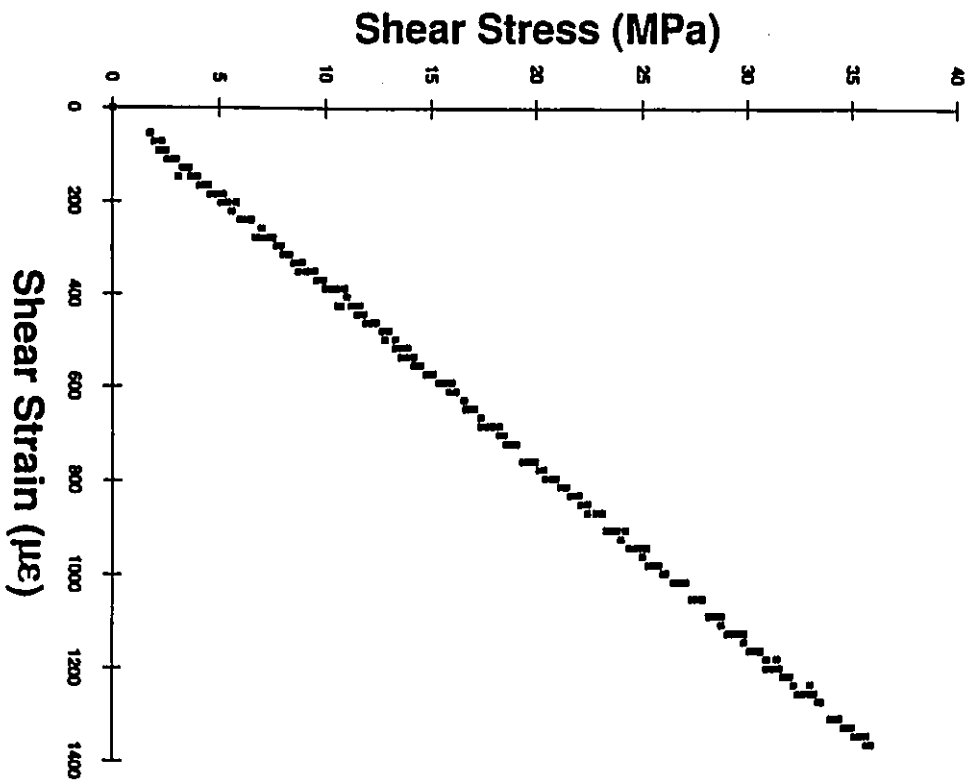
Stress Strain Diagram for Aluminum Spec. Trial 1



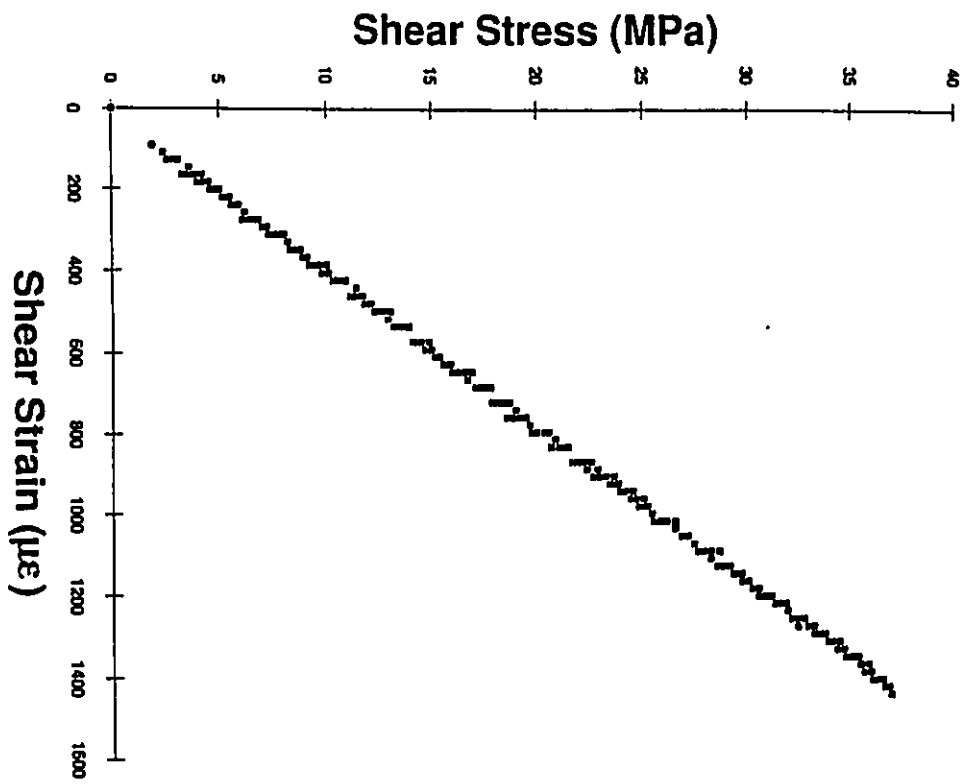
Stress Strain Diagram for Aluminum Spec. Trial 2



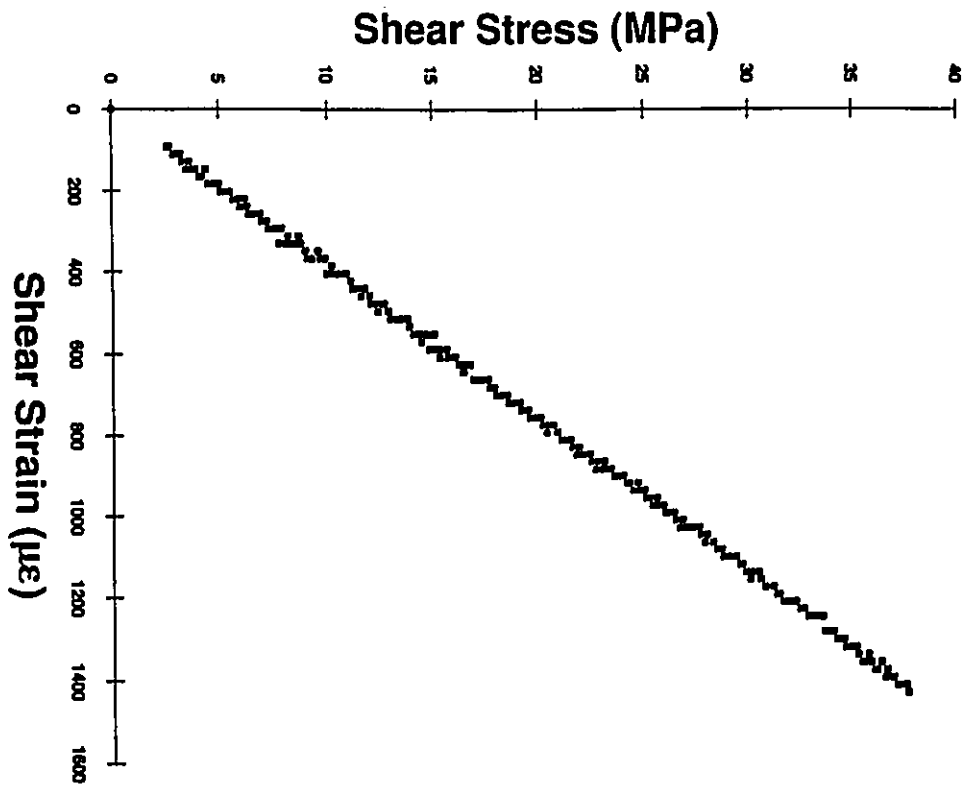
Stress Strain Diagram for Aluminum Spec. Incl 3



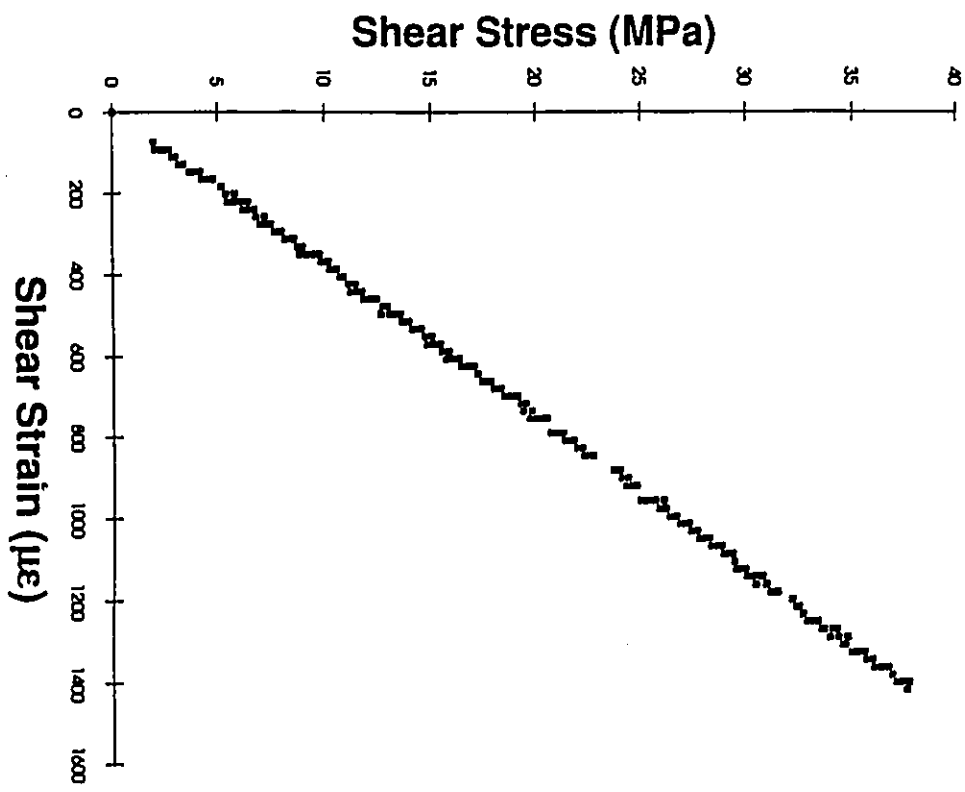
Stress Strain Diagram for Aluminum Spec. Incl 4



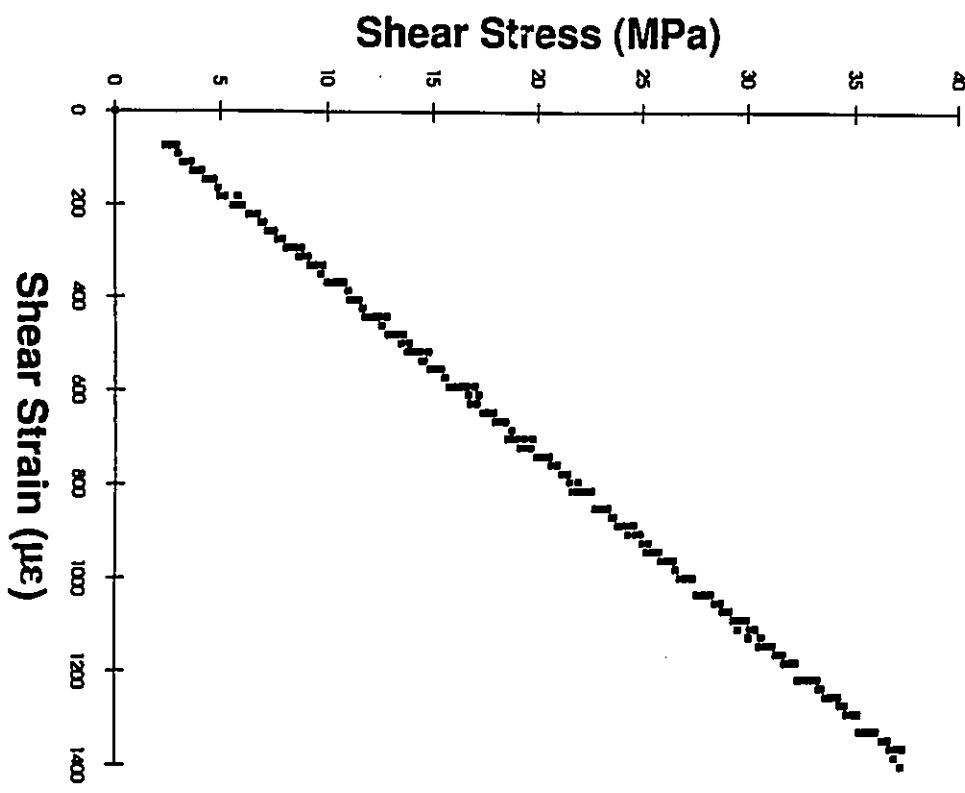
Stress Strain Diagram For Aluminum Spec. Trial 5



Stress Strain Diagram For Aluminum Spec. Trial 6

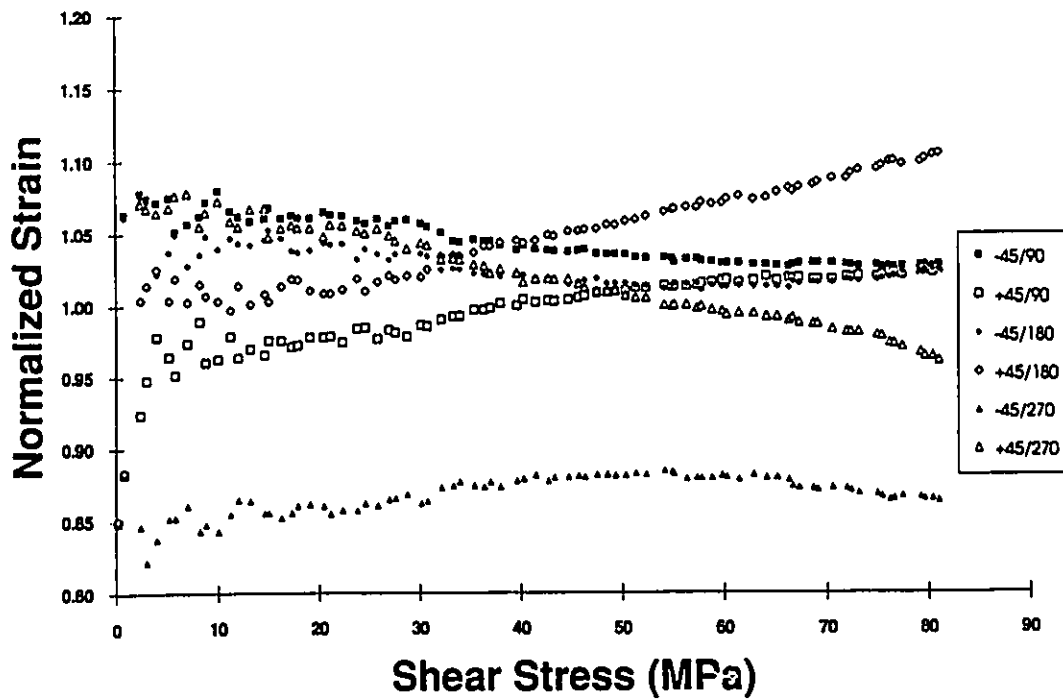


Stress Strain Diagram For Aluminum Spec. Trial 8 (Neg. Torque)

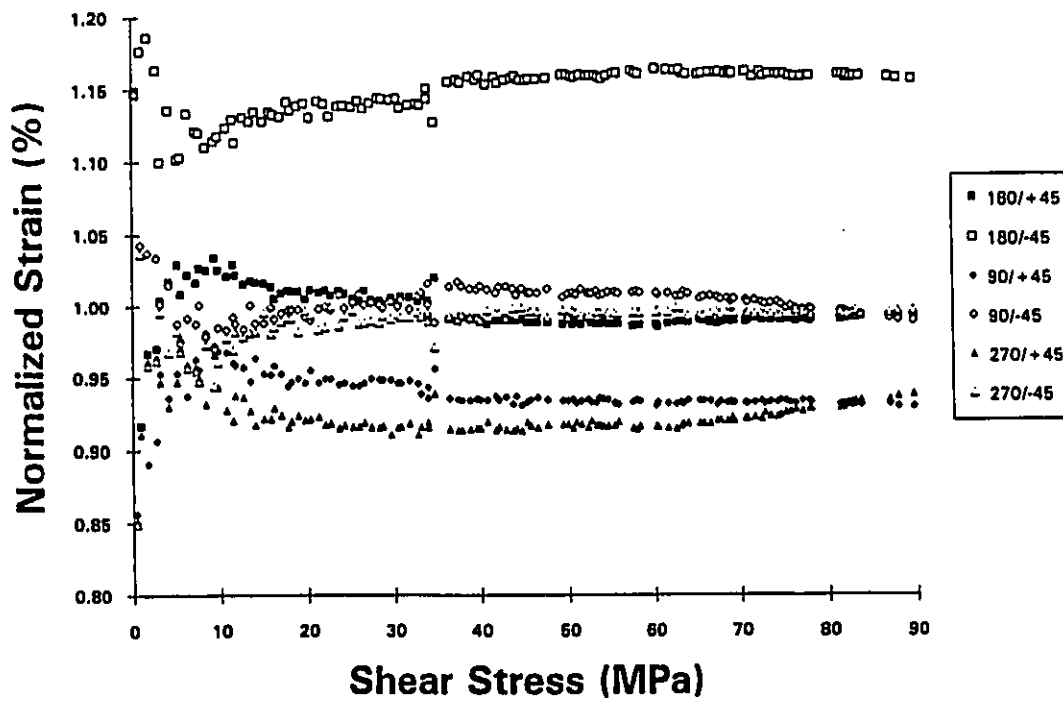


Stress Strain Diagram For Aluminum Spec. Trial 9 (Neg. Torque)

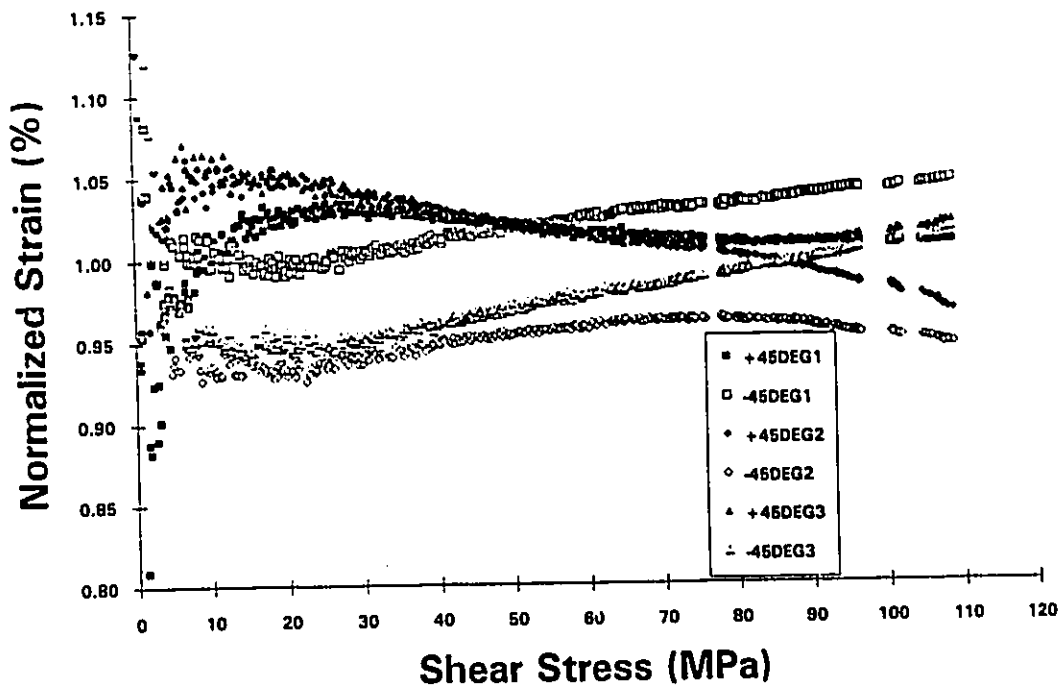
Normalized Strain Curves for Tubular In-plane Shear Specimens



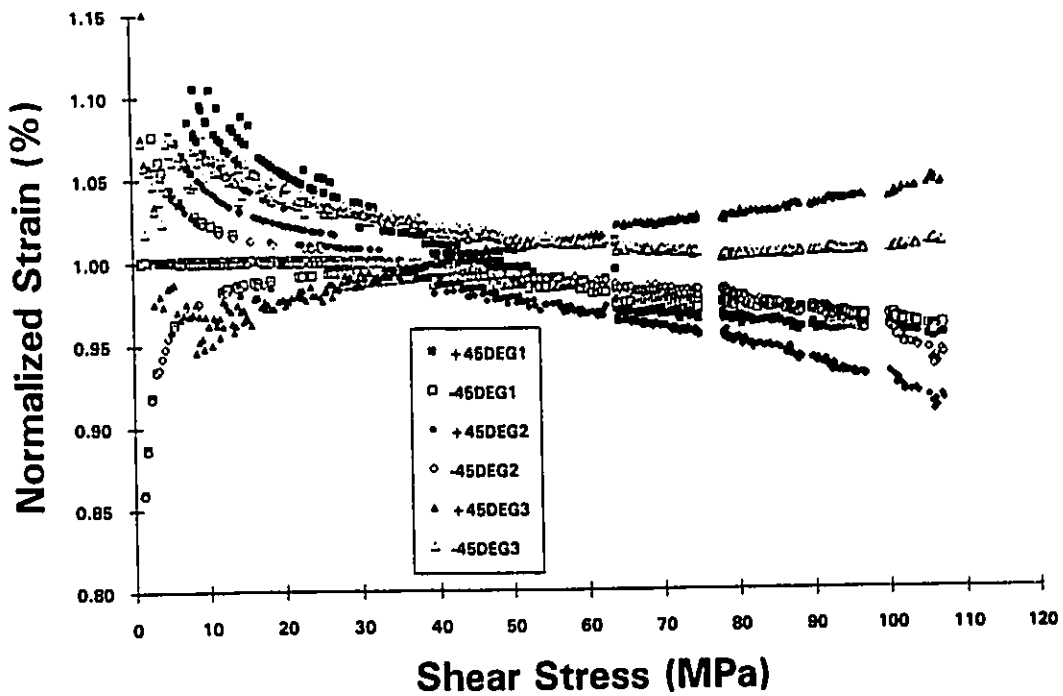
Normalized Strain Curves For Tube PR 5



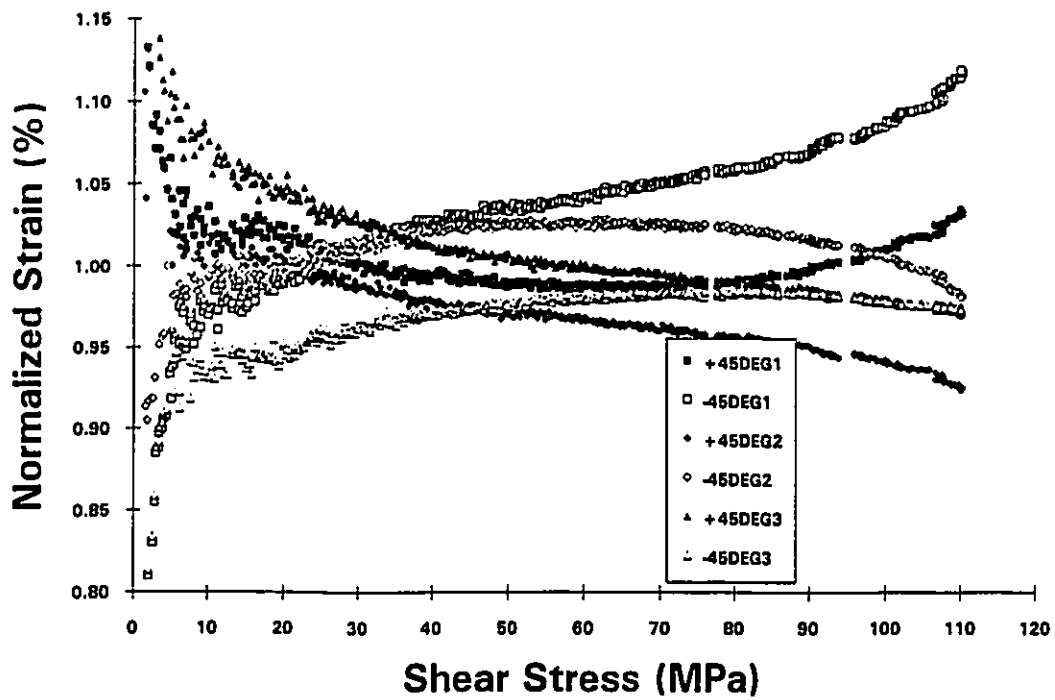
Normalized Strain Curves For Tube PR 6



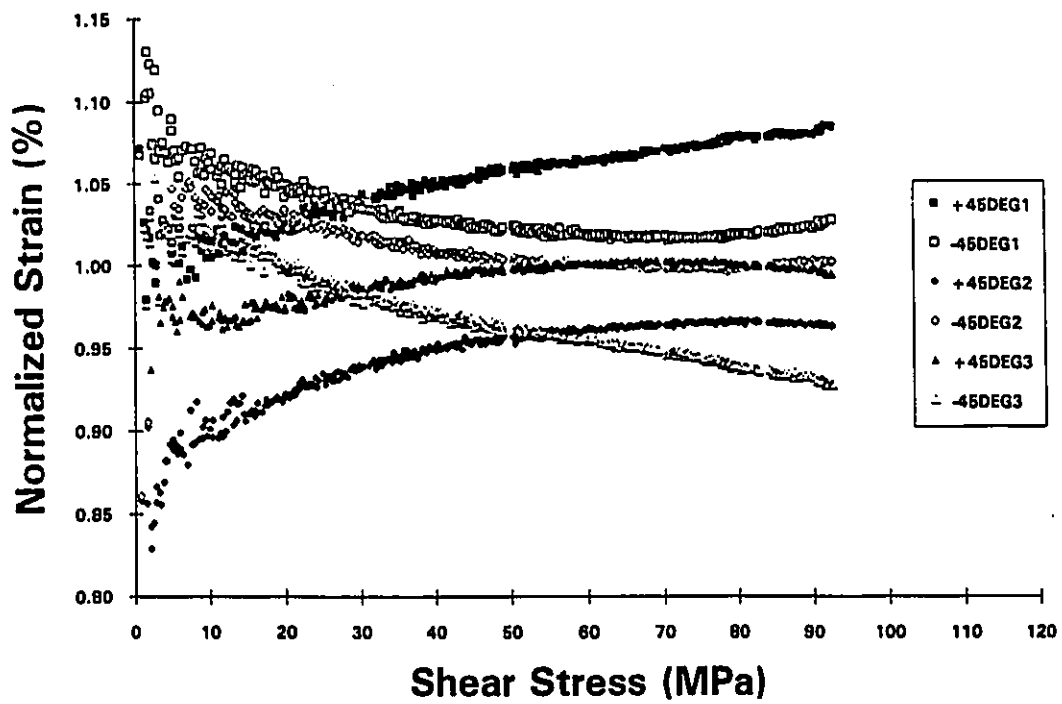
Normalized Strain Diagram For Tube_0



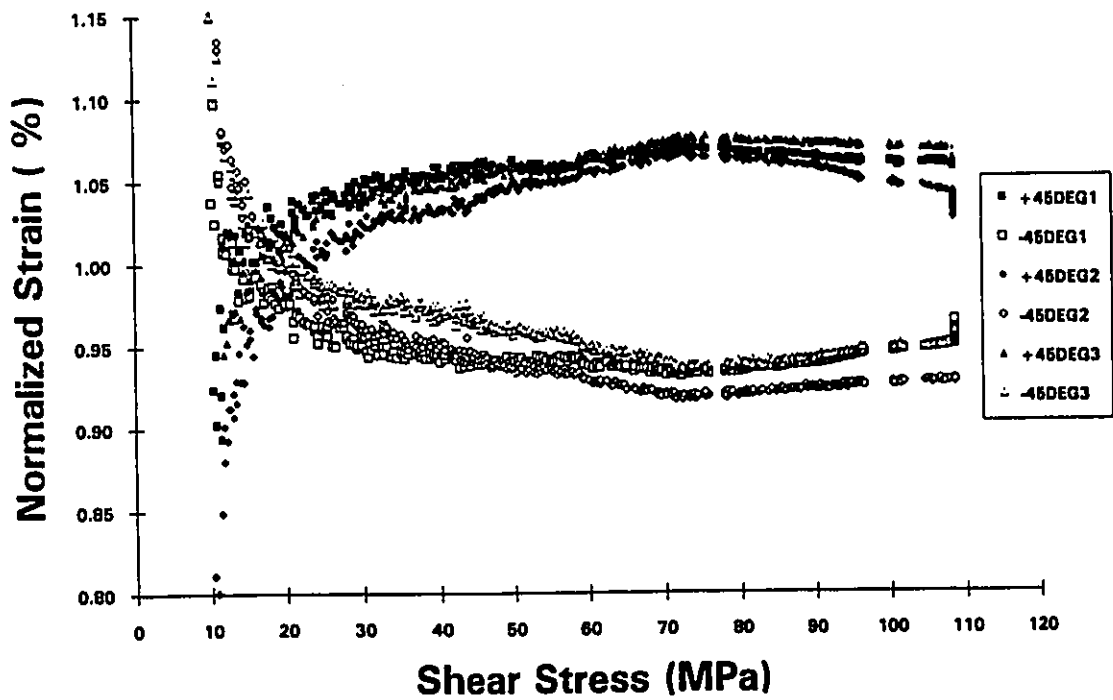
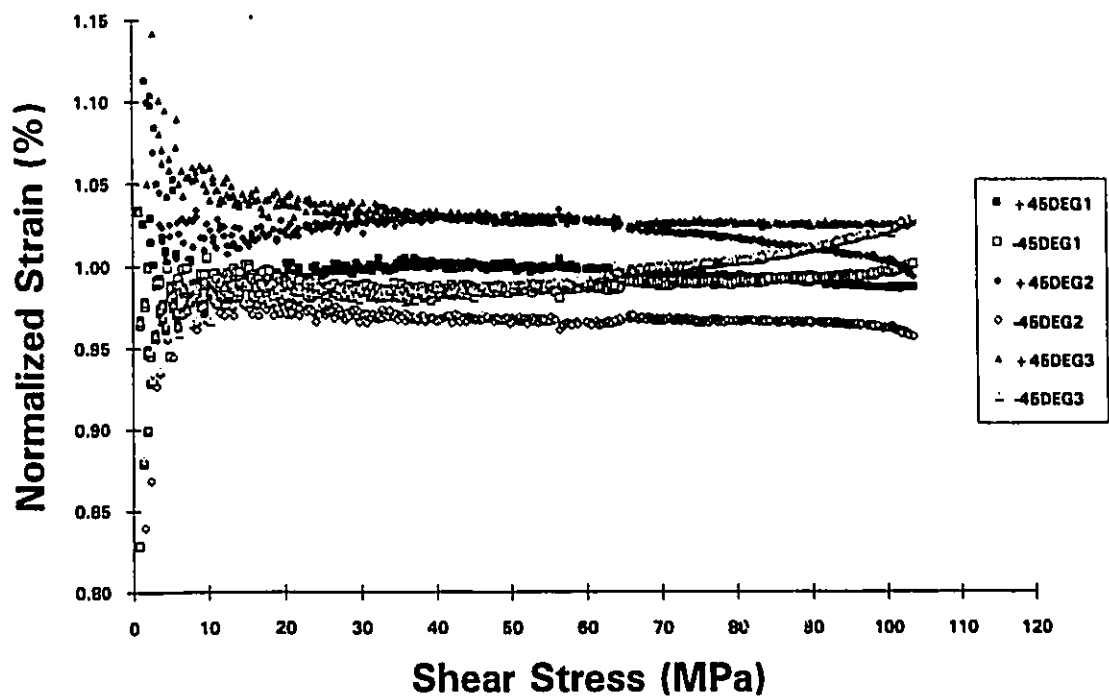
Normalized Strain Curve For Tube_1



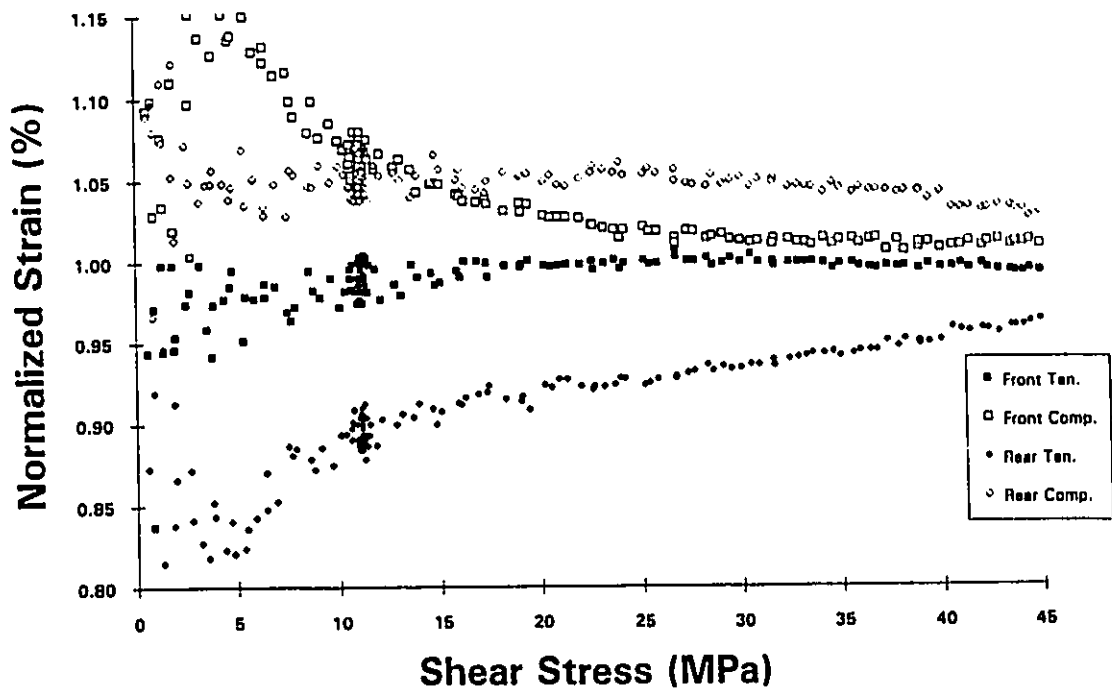
Normalized Strain Curve For Tube_2



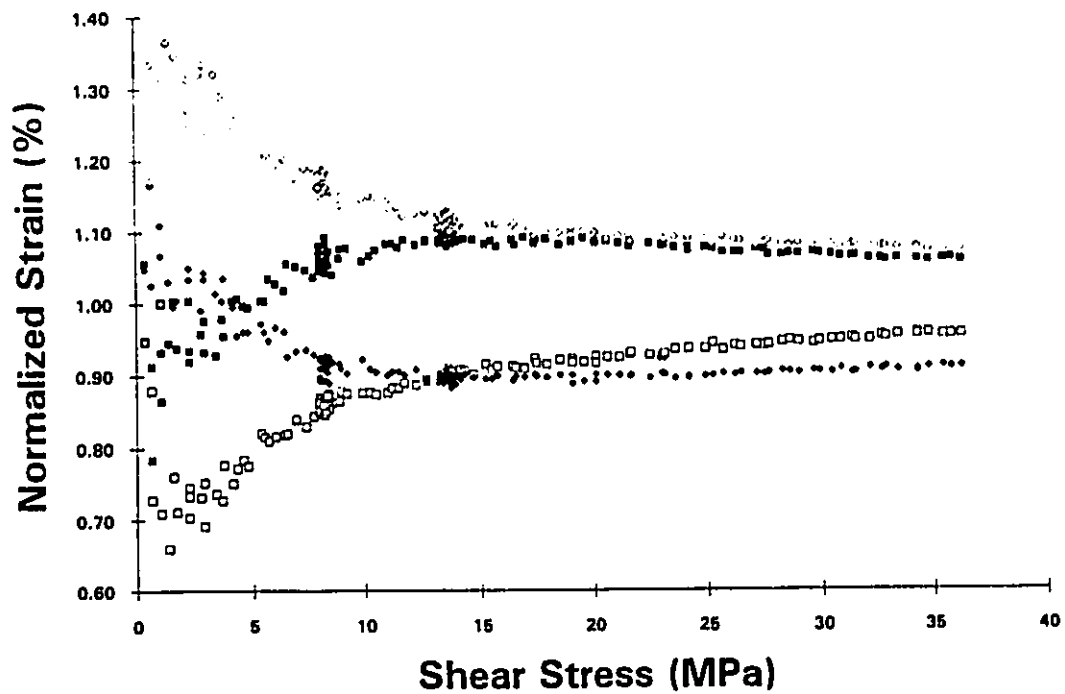
Normalized Strain Curve For Tube_3



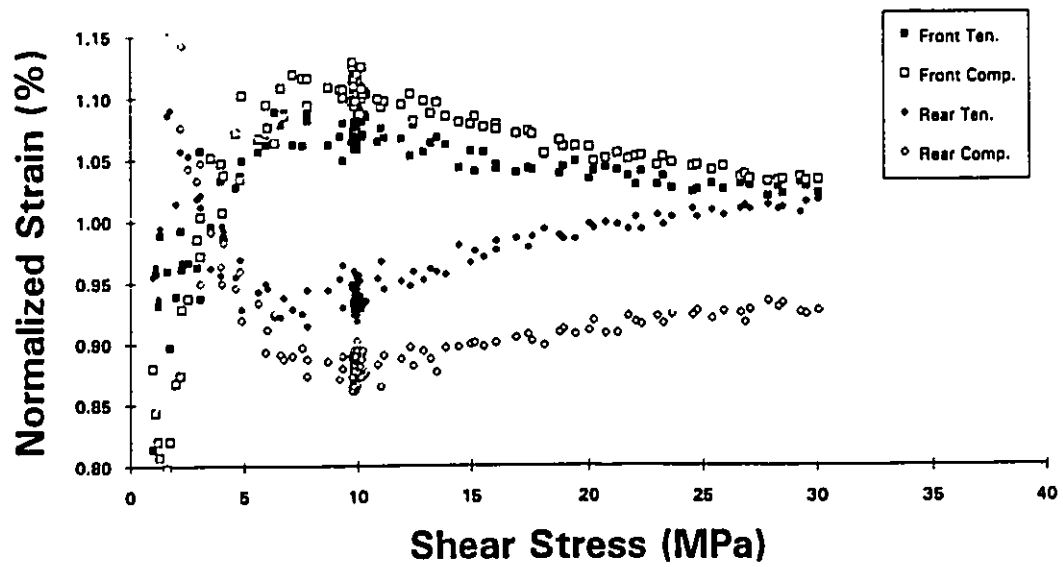
Normalized Strain Curves for 90° Iosipescu Shear Test



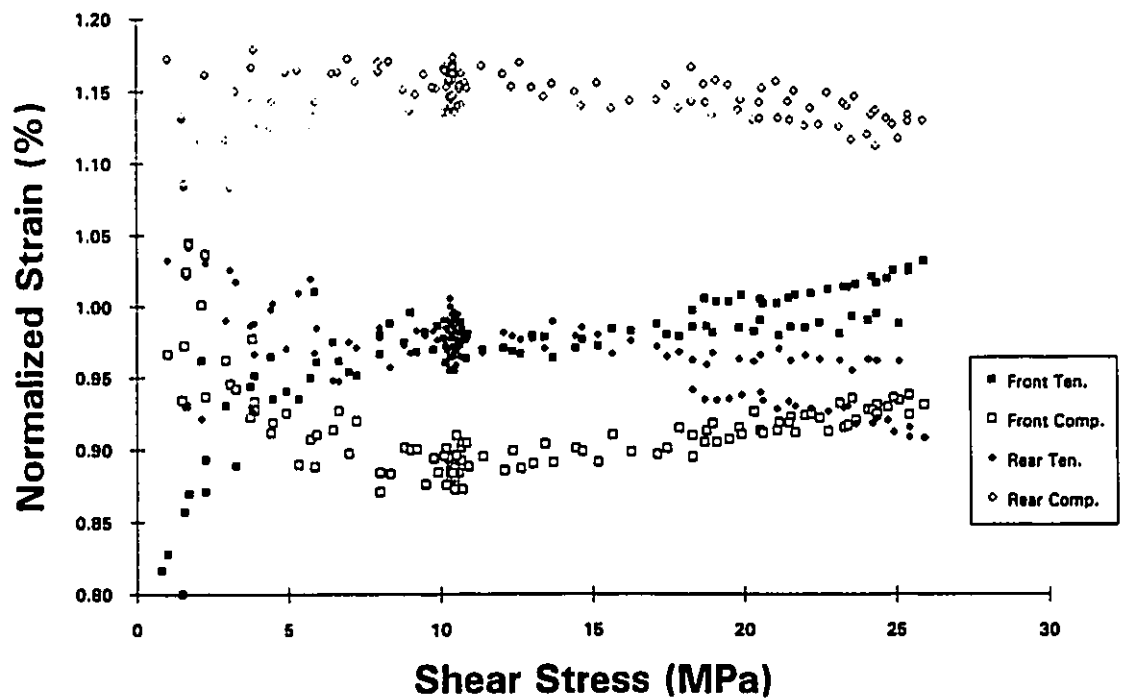
Normalized Strain Curve For Shear3_0



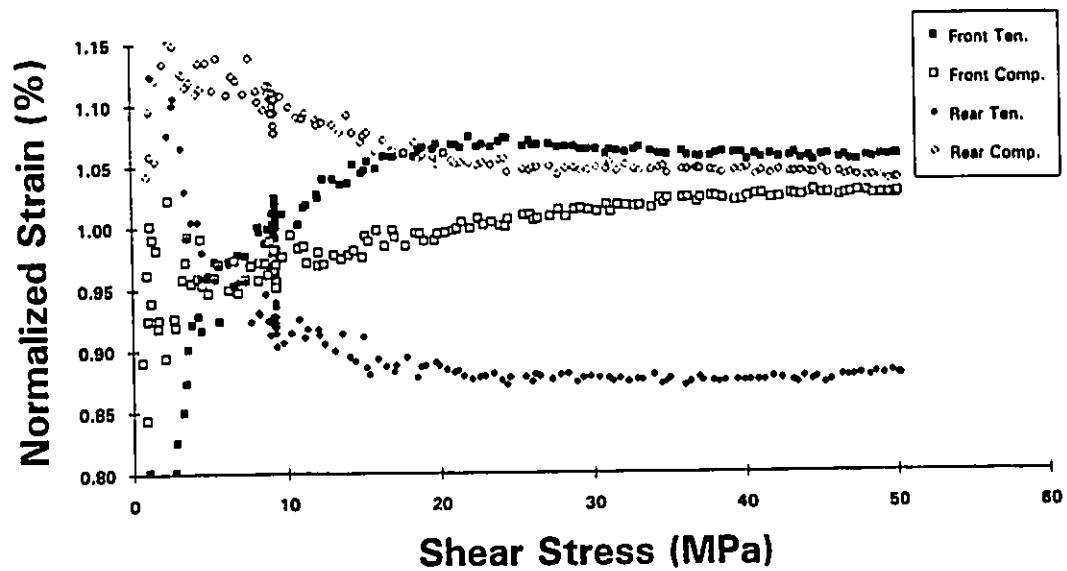
Normalized Strain Curves For Shear3_1



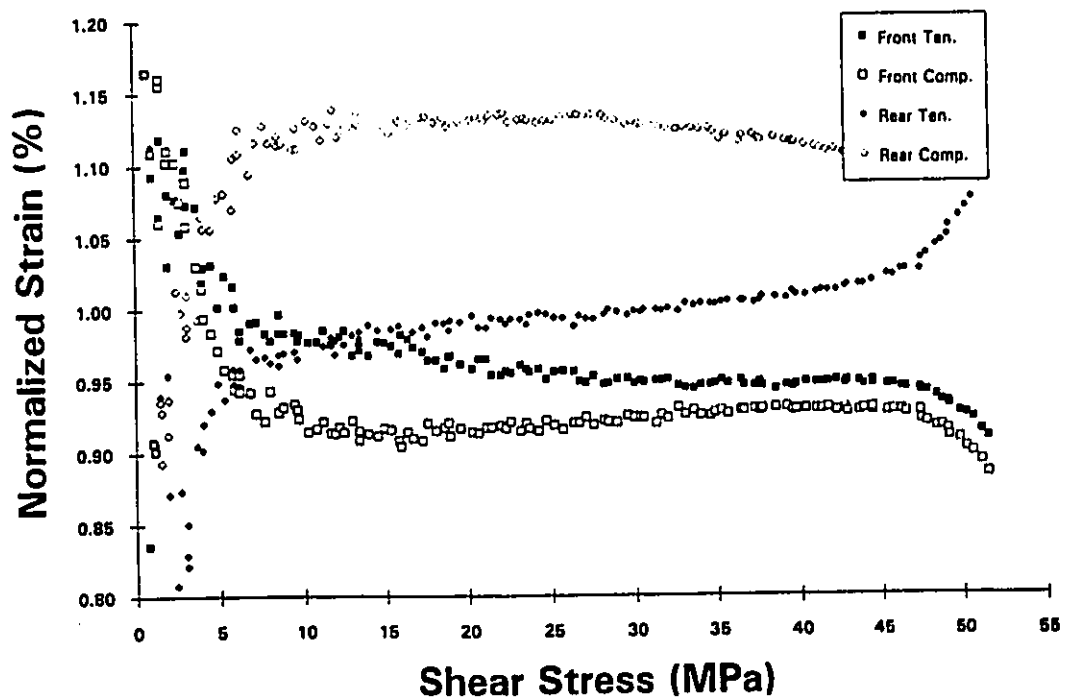
Normalized Strain Curve For Shear3 2



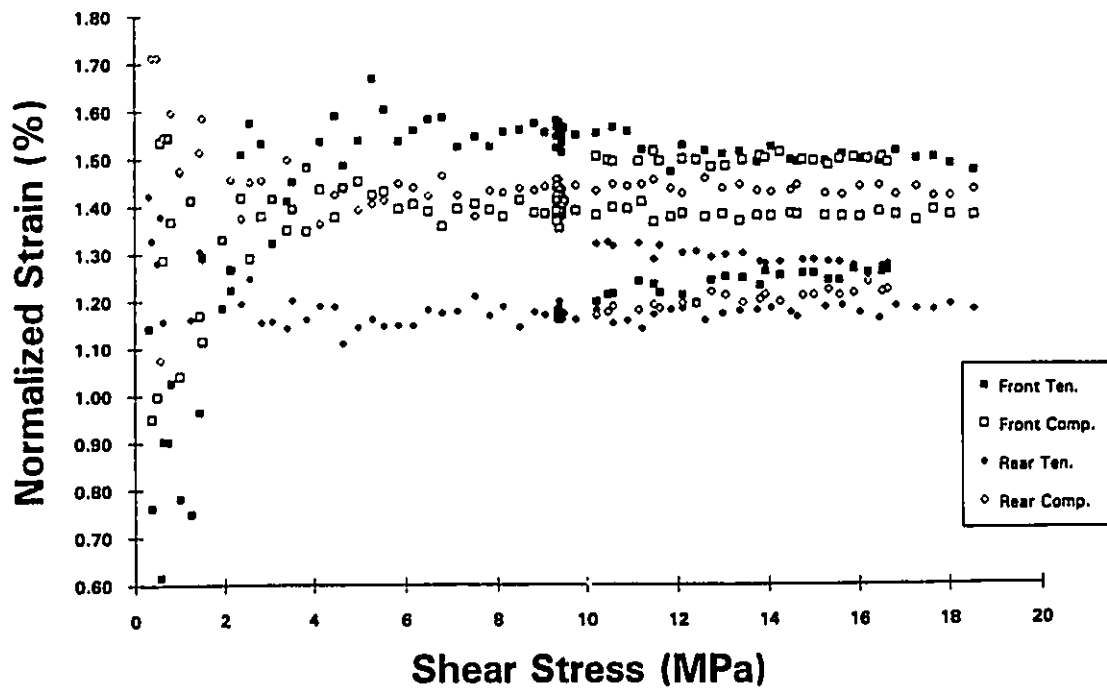
Normalized Strain Curves For Shear3 3



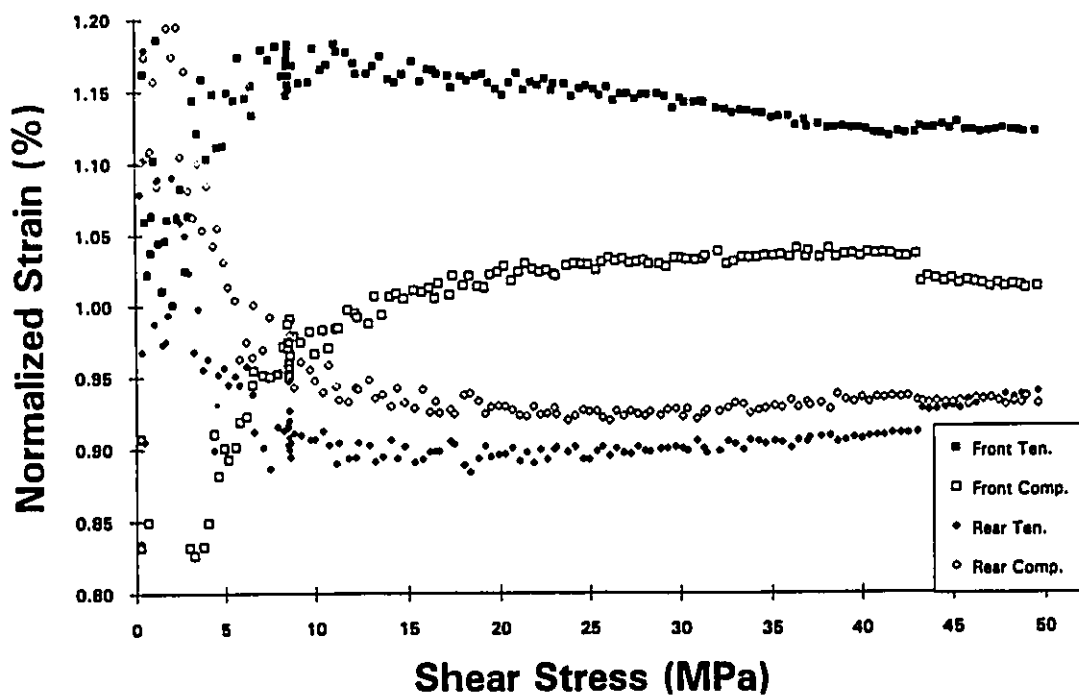
Normalized Strain Curve For Shear3_4



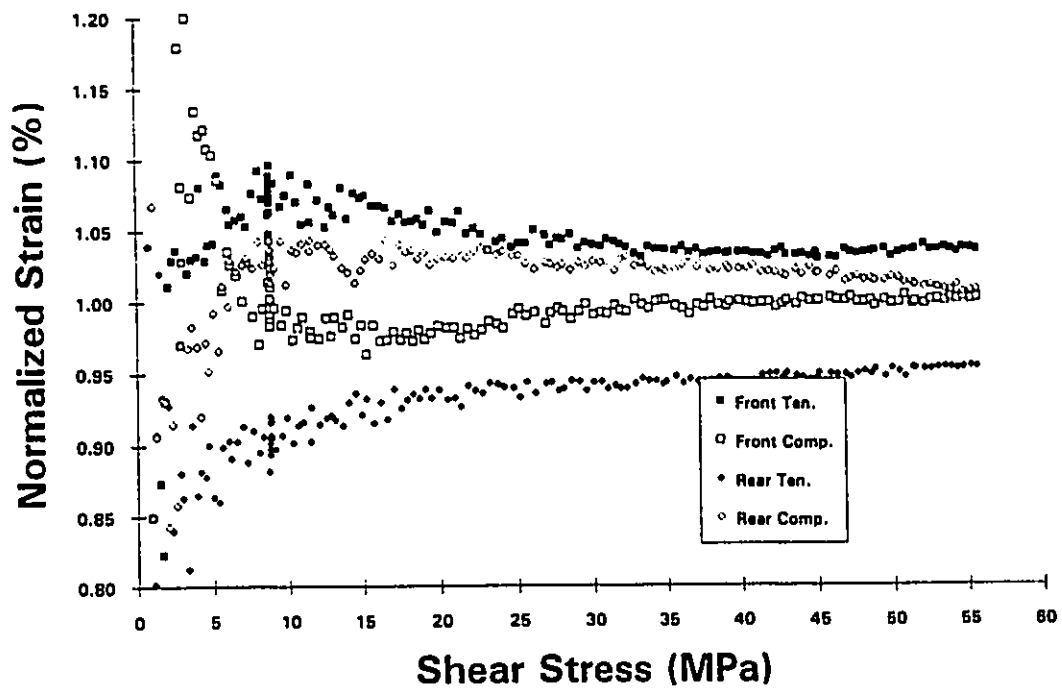
Normalized Strain Curve For Shear3_6



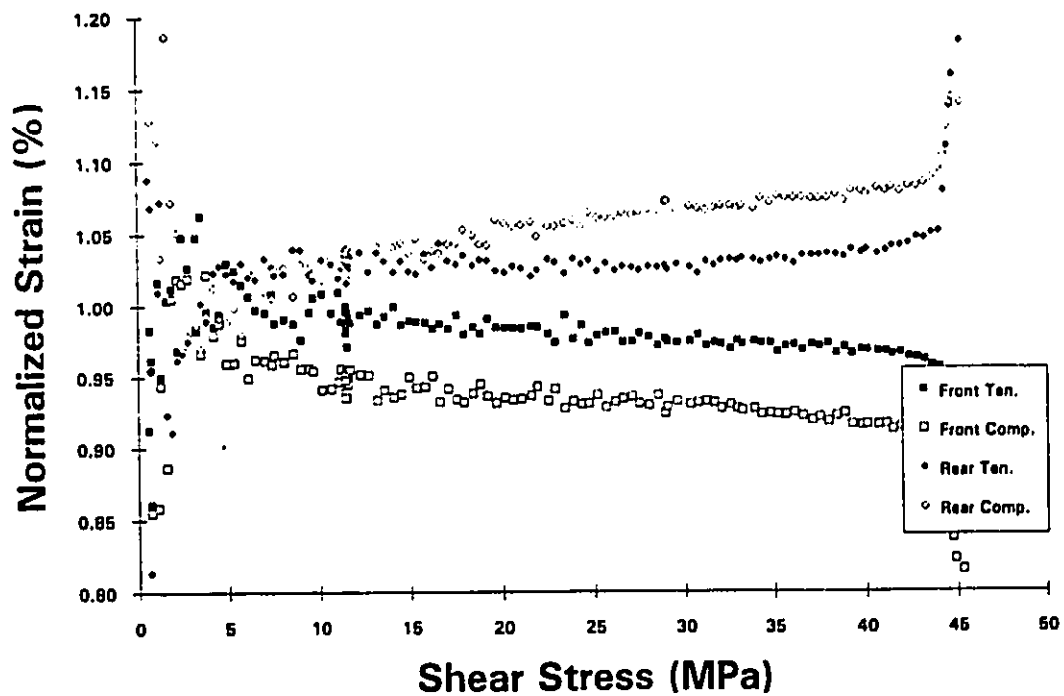
Normalized Strain Curves For Shear3_7



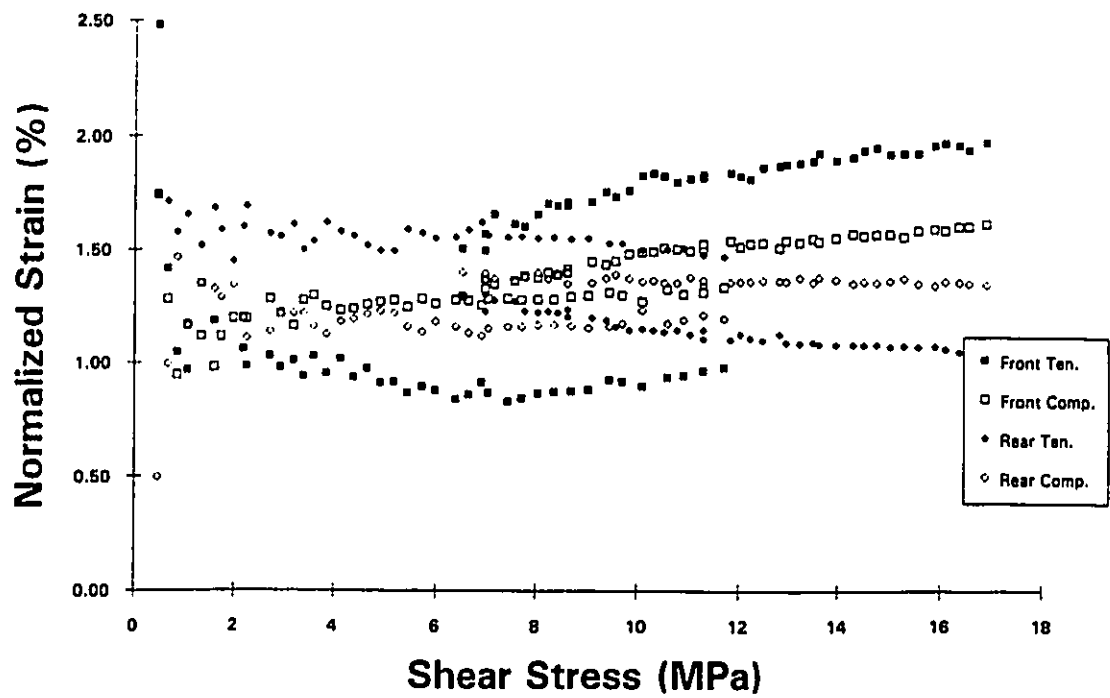
Normalized Shear Curves For Shear3_8



Normalized Strain Curve For Shear3_9

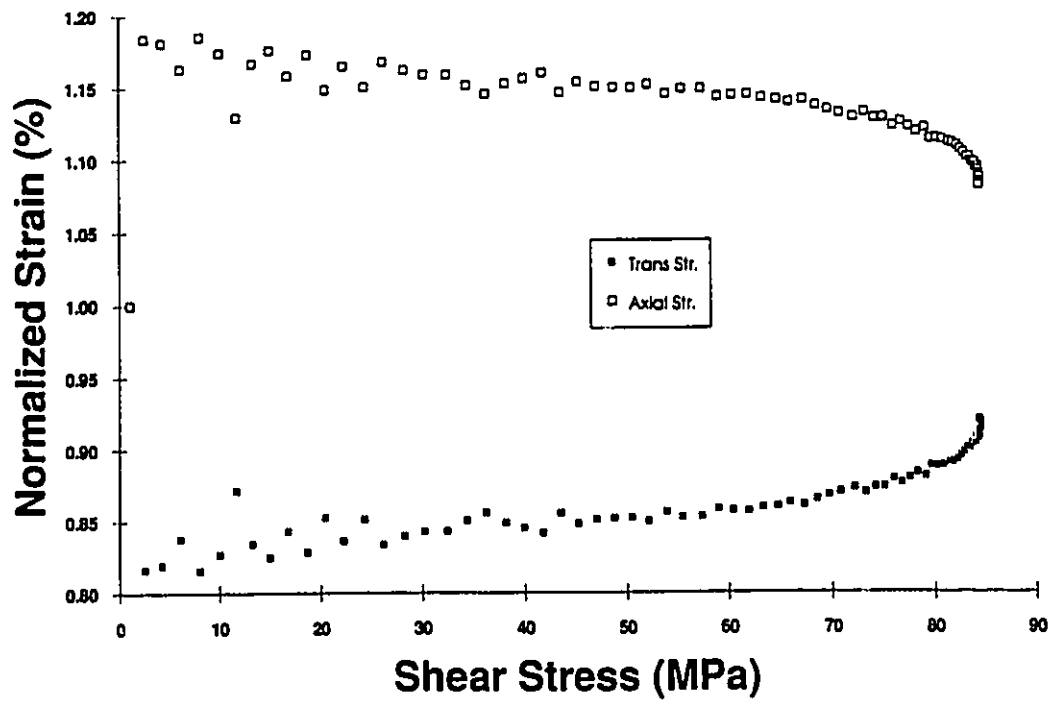


Normalized Strain Curve For Shear3_10

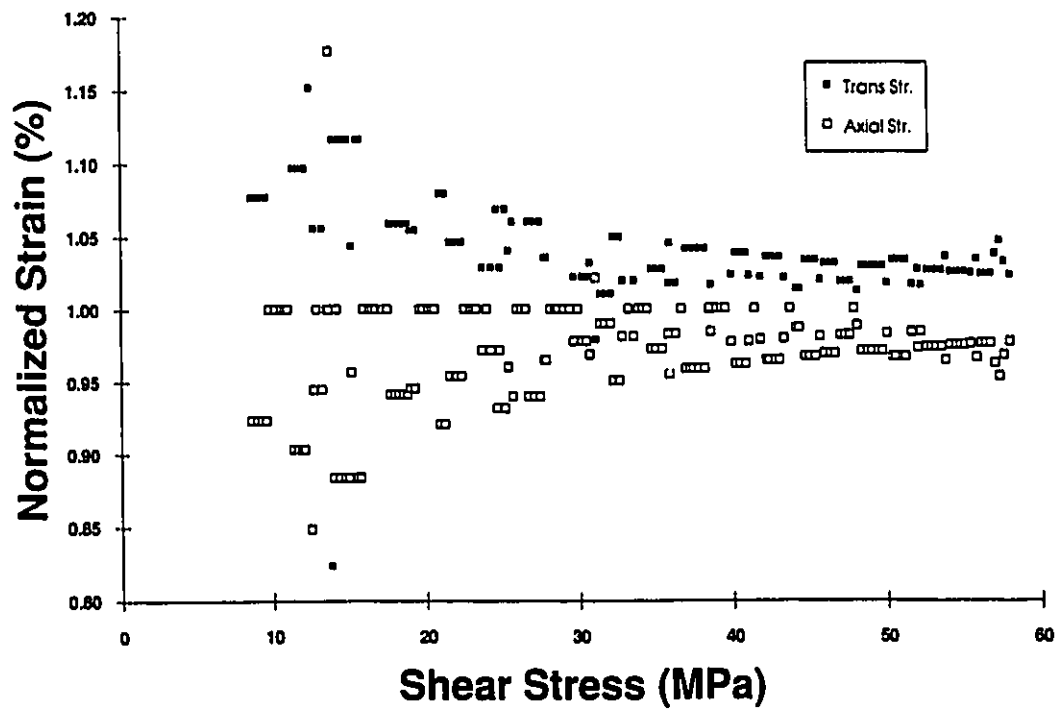


Normalized Strain Curves For Shear3_11

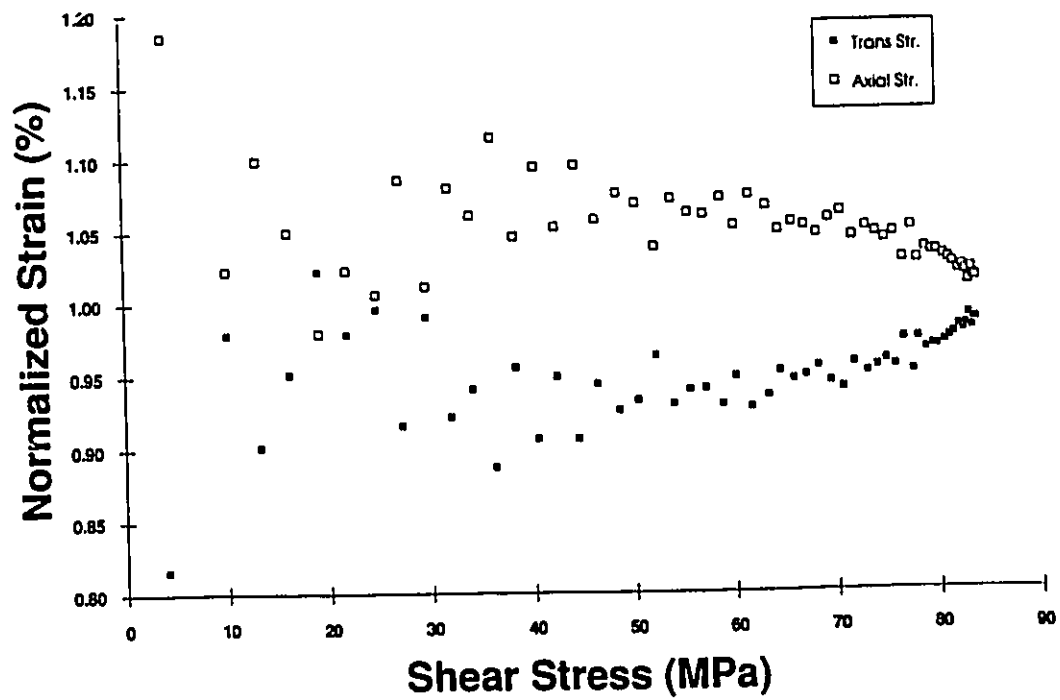
**Normalized Strain Curves for
 $\pm 45^\circ$ Tensile Shear Test**



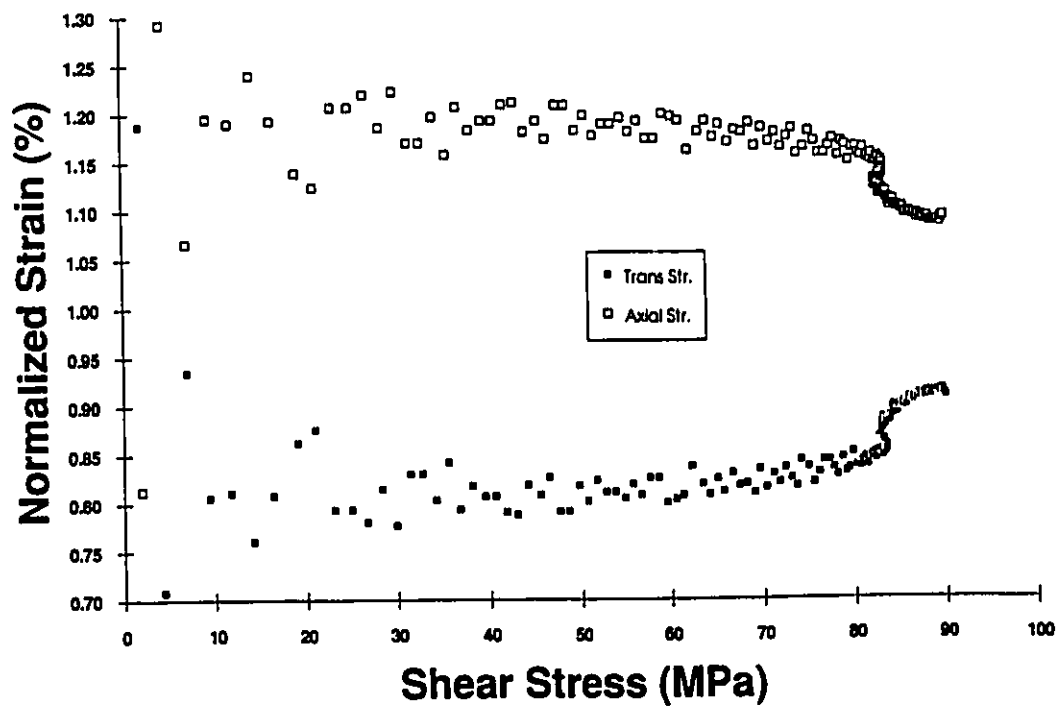
Normalized Strain Curves For Shear1_1



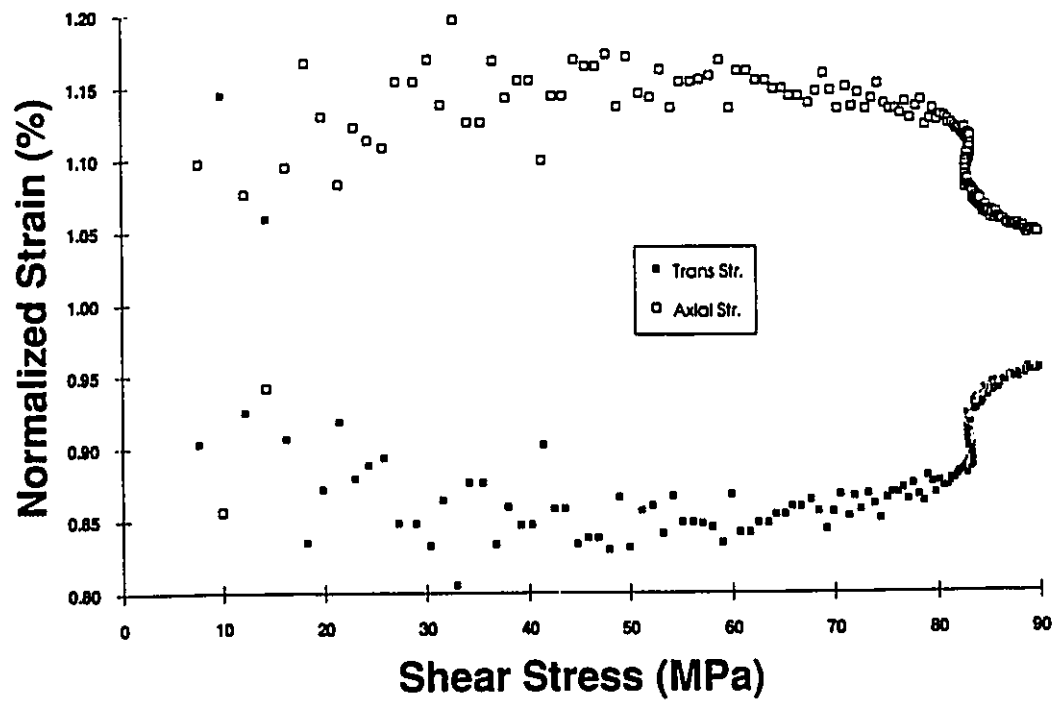
Normalized Strain Curves For Shear1_2



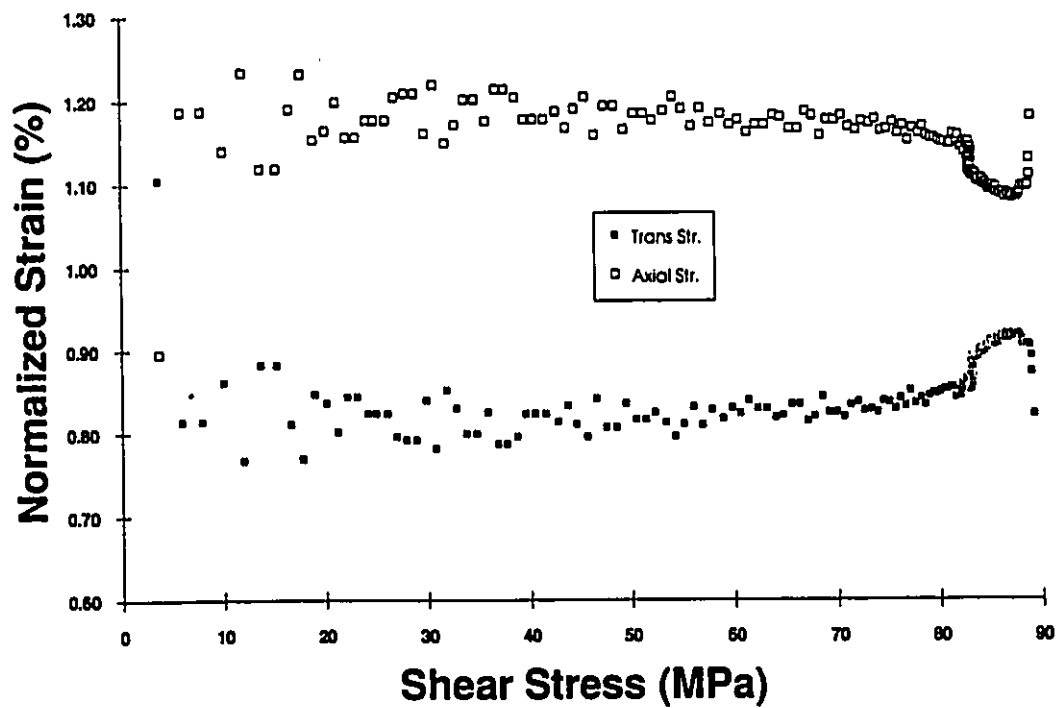
Normalized Strain Curves For Shear1_4



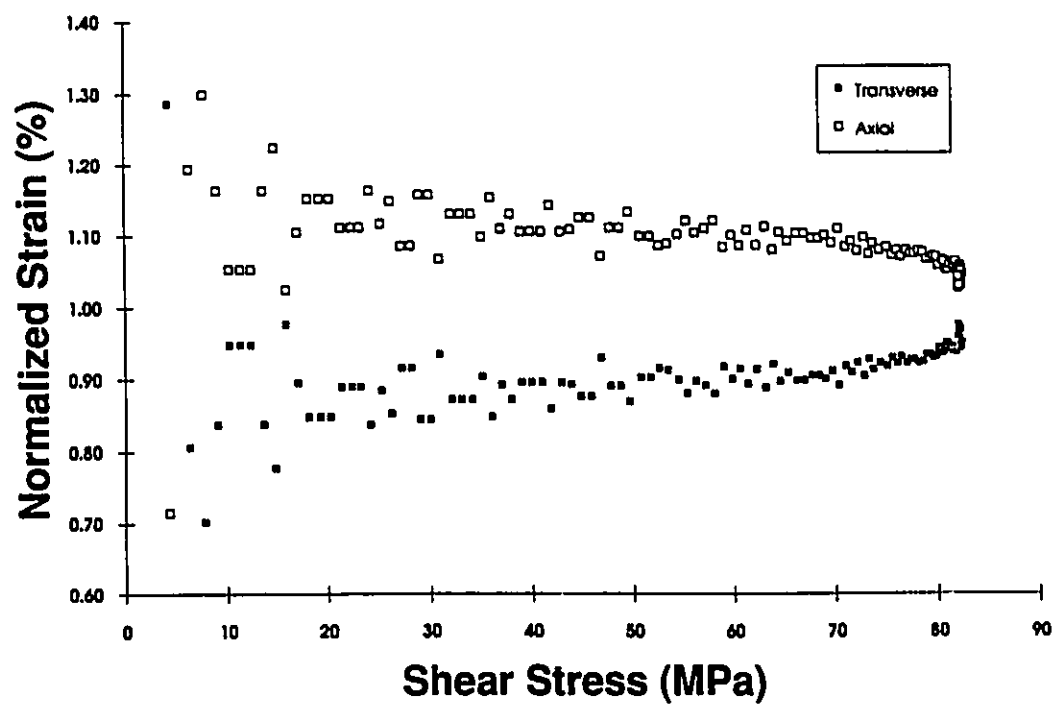
Normalized Strain Curves For Shear1_5



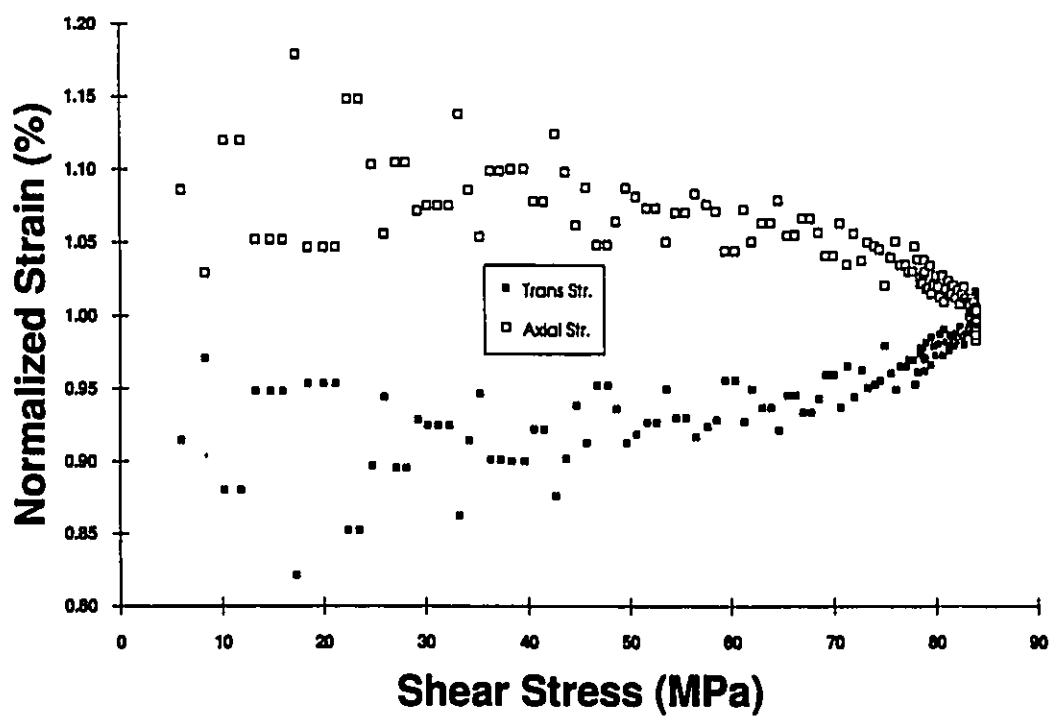
Normalized Strain Curves For Strain1_6



Normalized Strain Curves For Shear1_7

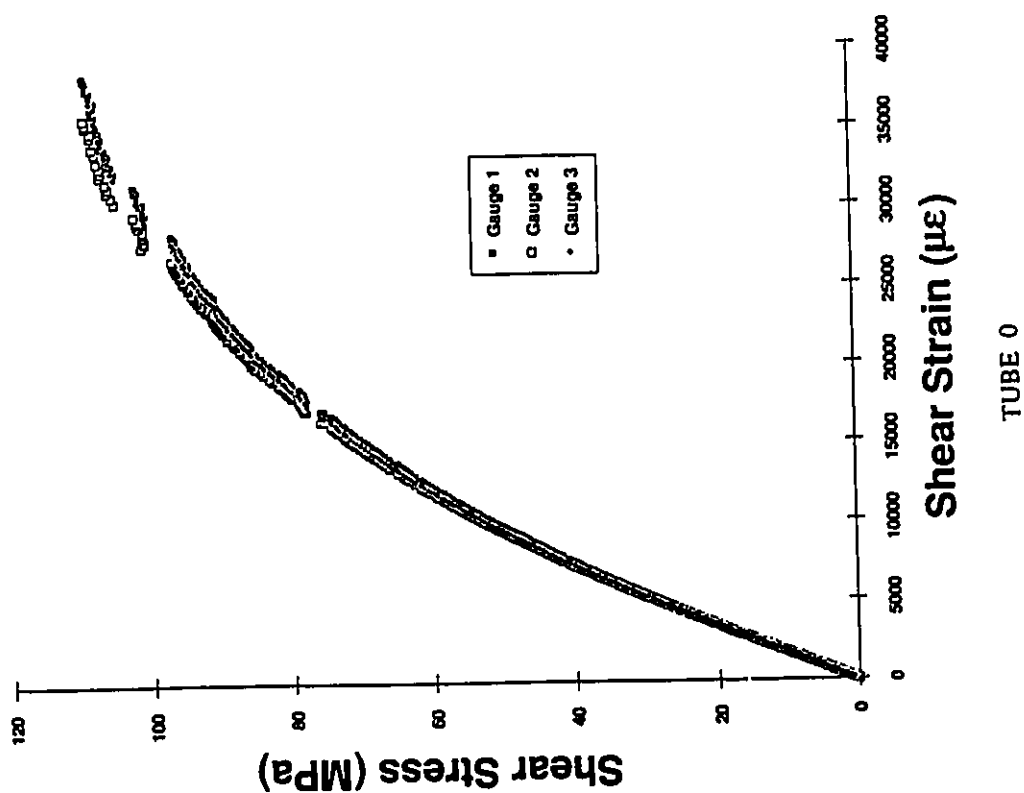
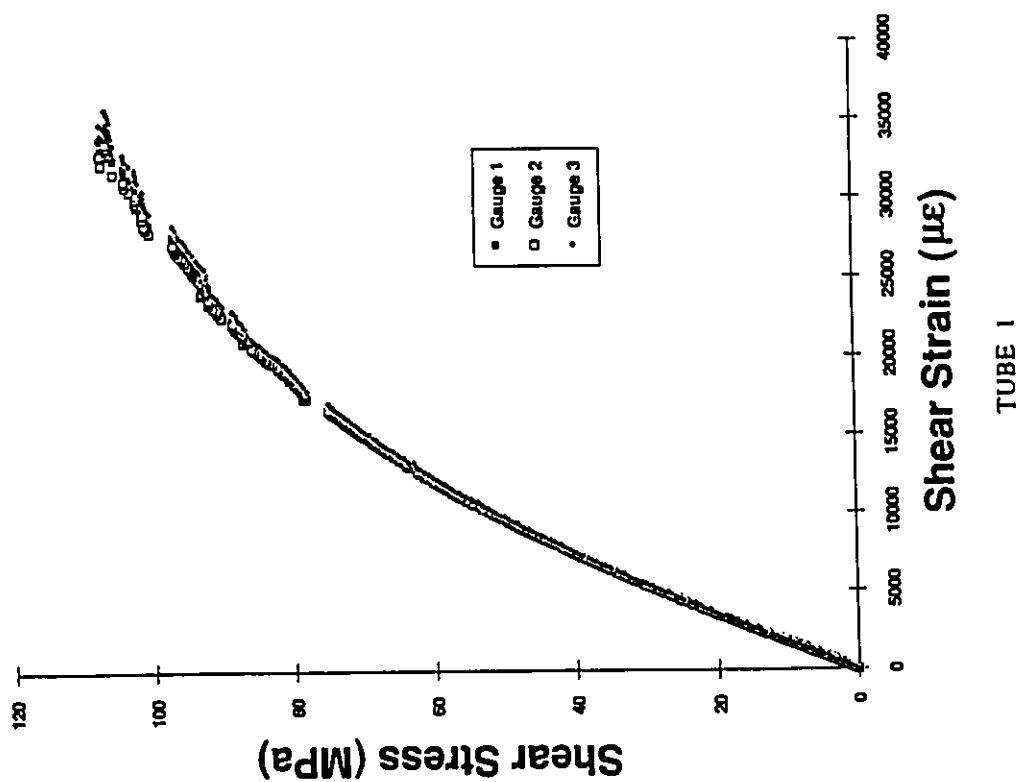


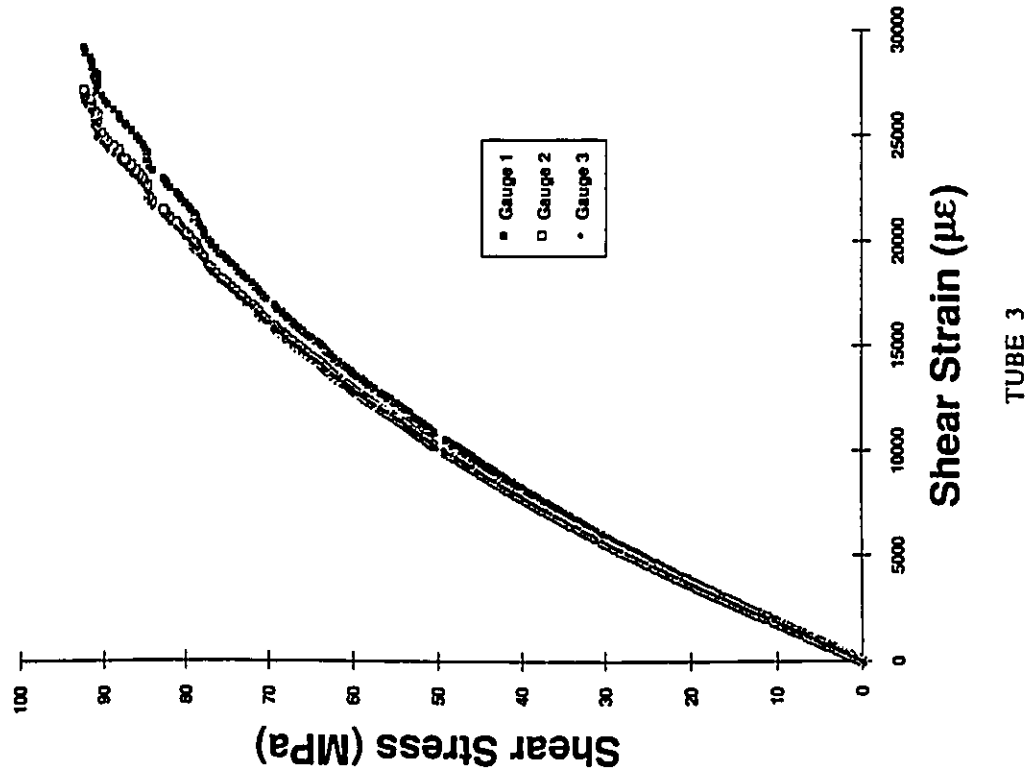
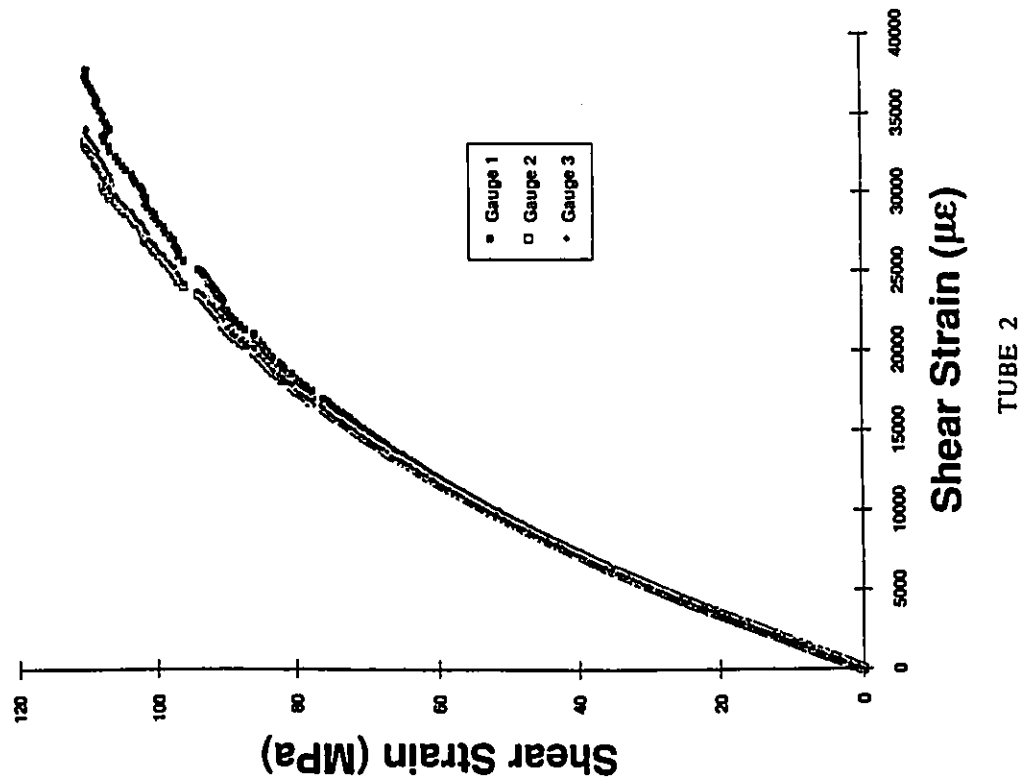
Normalized Strain Curve For Shear1_8

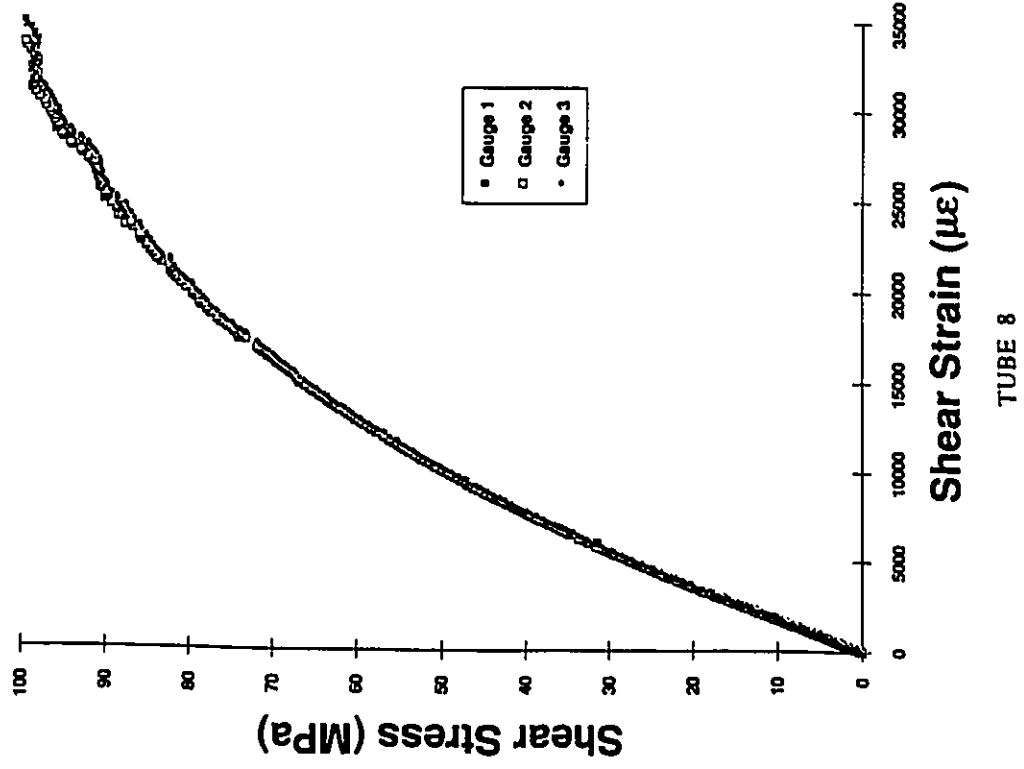
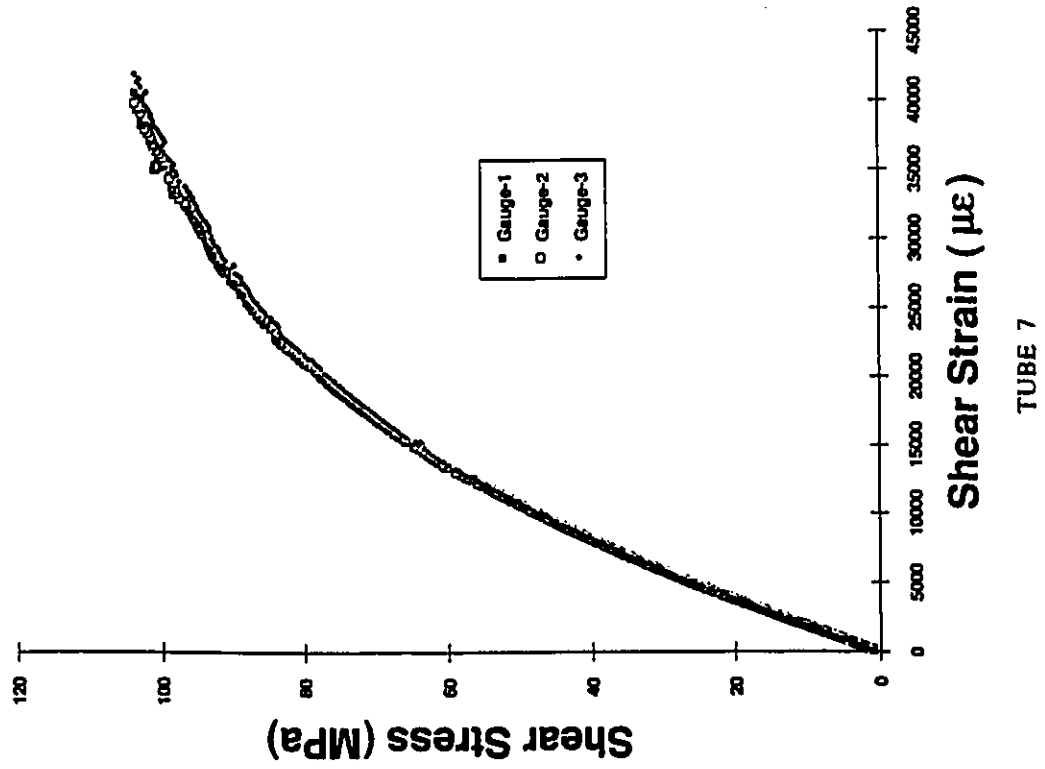


Normalized Strain Curves For Shear1_9

Stress-Strain Curves for Circumferential Gauges of Tubular In-plane Shear Specimens

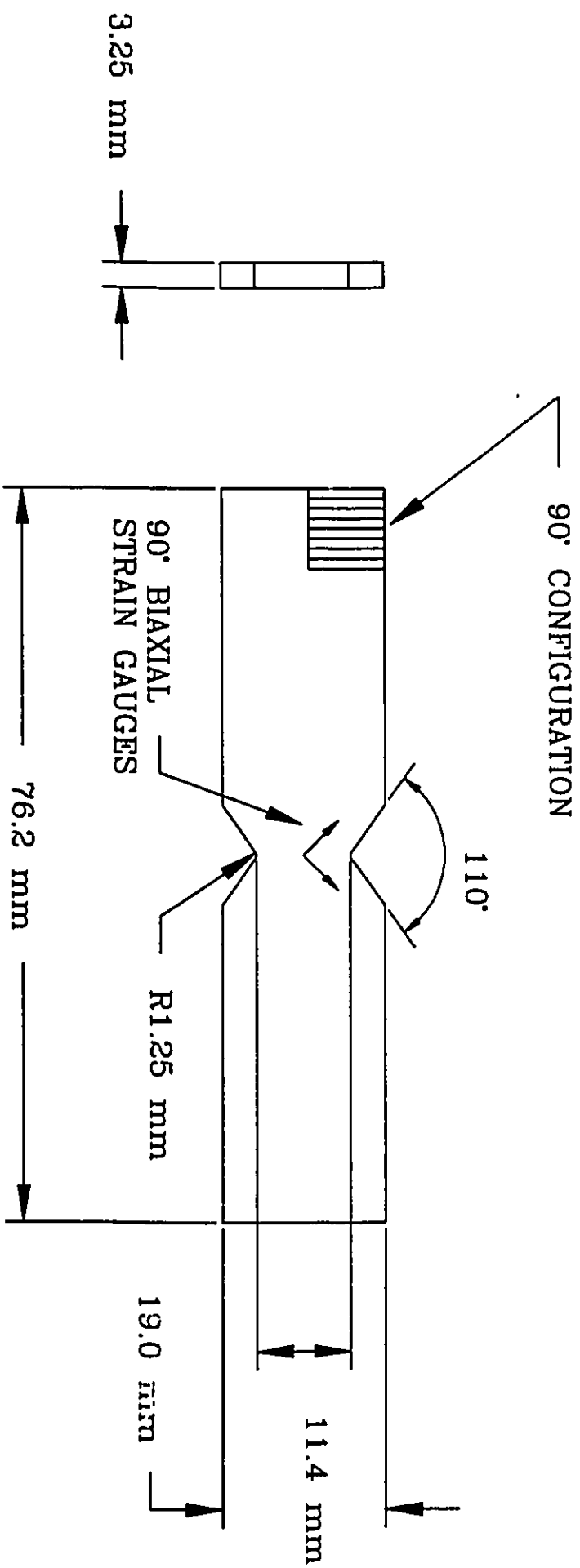




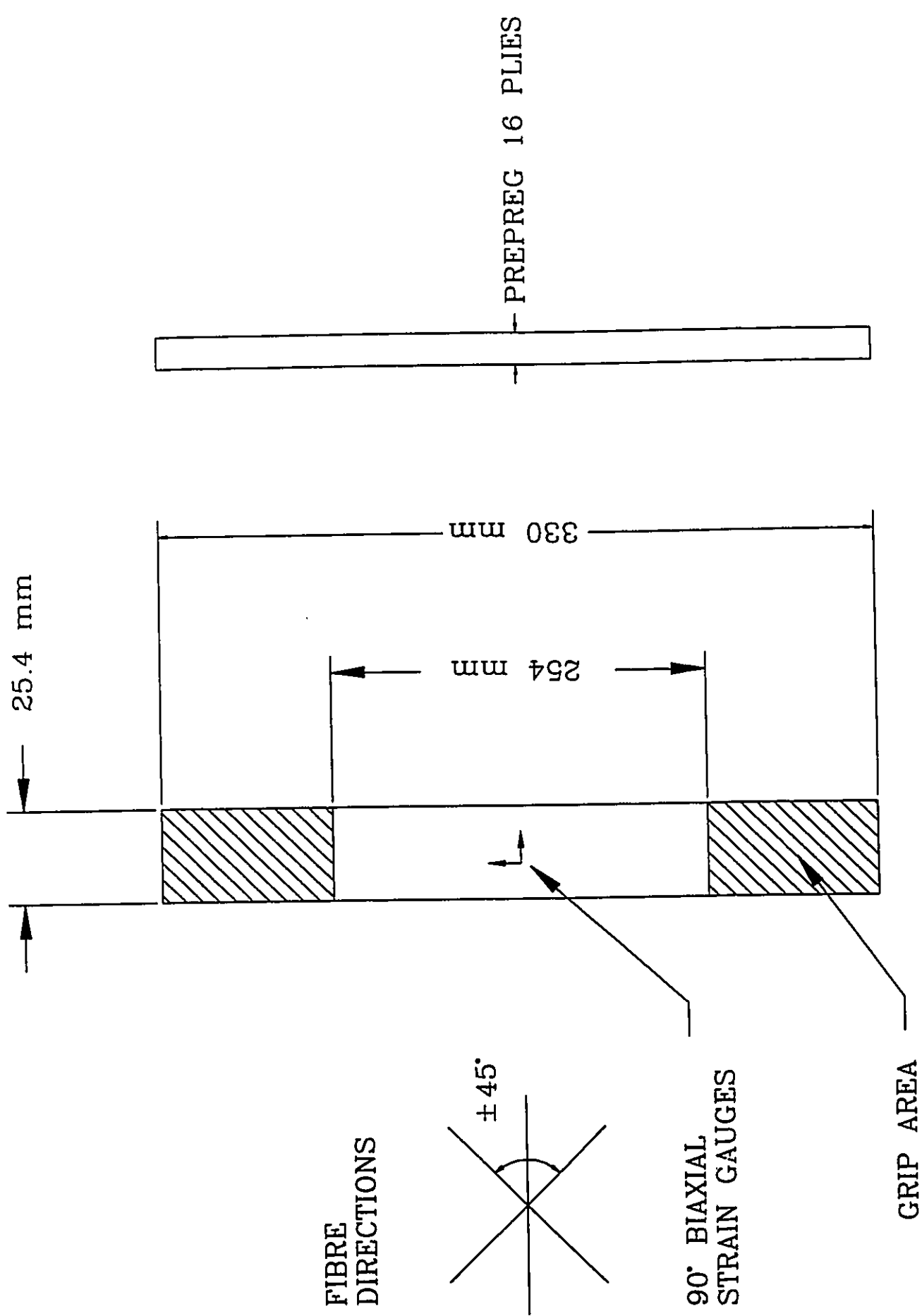


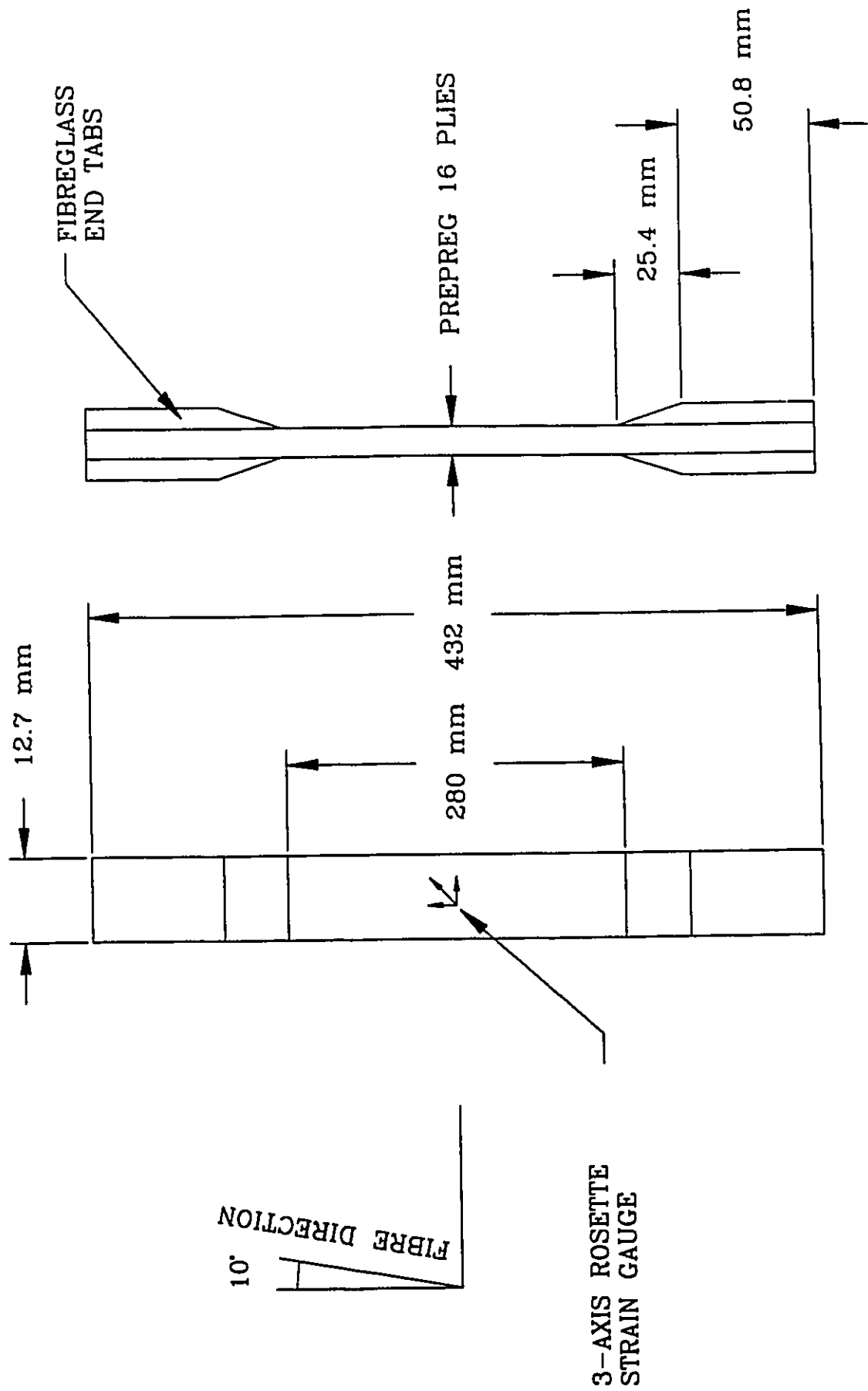
Appendix F

Dimensions of Coupon In-plane Shear Test Specimens

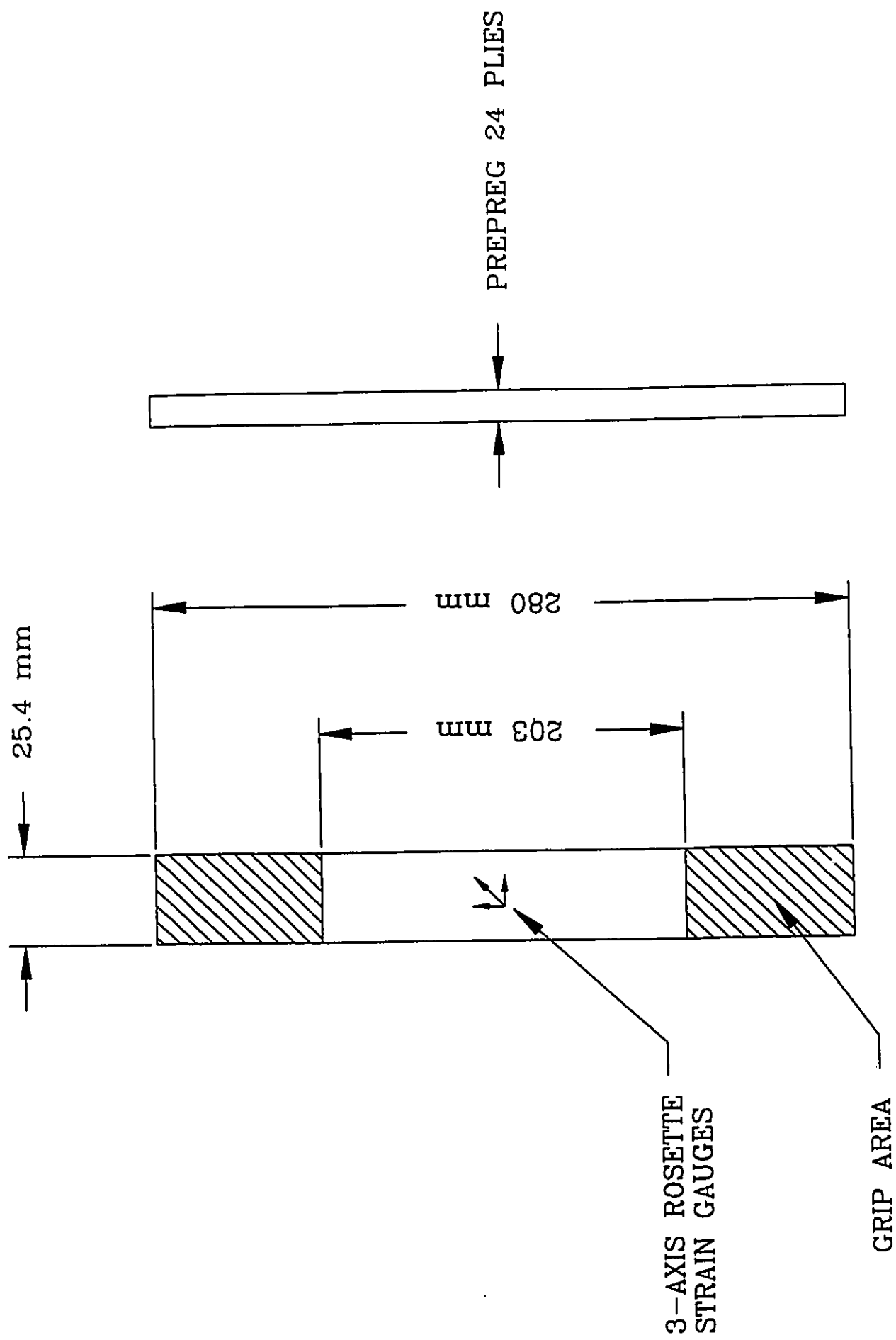


90° Isipescu Specimen





10° Off-axis Specimen



Laminate Specimen