Joining and Repair of Composite Structures

Keith T. Kedward and Hyonny Kim, Editors

ASTM Stock Number: STP1455

ASTM International
100 Barr Harbor Drive
PO Box C700
West Conshohocken, PA 19428-2959

Printed in the U.S.A.
Foreword

This publication, Joining and Repair of Composite Structures, contains selected papers presented at the symposium of the same name held in Kansas City, Missouri, on 17–18 March, 2003. The symposium was sponsored by Committee D-30 on Composite Materials. The symposium chairmen and co-editors were Keith T. Kedward and Hyonny Kim.
Contents

Overview vii

SECTION I. ADHESIVELY BONDED ATTACHMENTS

Application of a Sublamine Method to the Analysis of Bonded Joints—G. V. FLANAGAN AND S. CHATTERJEE 3

Adhesive Nonlinearity and the Prediction of Failure in Bonded Composite Lap Joints—H. KIM AND J. LEE 22

Box Beam Lap Shear Torsion Testing for Evaluating Structural Performance of Adhesive Bonded Joints—J. S. TOMBLIN, W. P. SENEVIRATNE, H. KIM, AND J. LEE 42

Performance of a Composite Double Strap Joint with Attachments—H. QIAN AND C. T. SUN 55


SECTION II. ADHESIVELY BONDED REPAIR

Static and Dynamic Strength of Scarf-Repaired Thick-Section Composite Plates—B. A. GAMA, S. MAHDI, C. CICHANOWSKI, S. YARLAGADDA, AND J. W. GILLESPIE, JR 95

Installation of Adhesively Bonded Composites to Repair Carbon Steel Structure—D. ROACH, K. RACKOW, AND D. DUNN 110

SECTION III. BOLTED ATTACHMENTS

Bolted Joint Analyses for Composite Structures—Current Empirical Methods and Future Scientific Prospects—L. J. HART-SMITH 127
IBOLT: A Composite Bolted Joint Static Strength Prediction Tool—
J. R. EISENMANN AND C. Q. ROUSSEAU 161

Damage and Failure Mechanisms in Composite Bolted Joints—H. BAU 182

Development of Compression Design Allowables for Composite Bolted Joints
Using ASTM Standard D 6742—A. J. SAWICKI 199
Overview

This book is a peer reviewed summary of the works of a majority of the authors who participated in the Symposium on Joining and Repair of Composite Structures, which took place on March 17 and 18, 2003, in Kansas City, Missouri under sponsorship of the ASTM Committee D30. This symposium addressed a critical and enabling component of composites technology, which was last featured by ASTM International as a Special Technical Publication in 1980 (STP 749). The use of composite structural assemblies in the aerospace, automotive, marine, and recreational industries has seen extensive growth in the intervening period. Inevitably, the joining, assembly, and repair of structures in all these industries continues to severely limit the expanded usage of composites. Certification and associated standards in testing are also key issues for industries that are continuously concerned with the joining, repair, and maintenance of composite structures.

The objective of the symposium was to provide a forum for interaction and synergy between the design, analysis, testing, and fabrication of structural joint and attachment configurations. The challenges faced in repair approaches that are needed to maintain composite and metallic structures add another dimension to the complexities of joining composites. The papers contained in this publication address this objective by covering a spectrum of topics relevant to the joining of composites. Papers focused on design, analysis, and testing are all represented. These are organized in this book by the general topic categories of adhesively bonded attachments, repair, and bolted attachments.

Adhesively Bonded Attachments

The papers in this section cover a wide range of topics encompassing the design, analysis, testing, and fabrication issues associated with adhesive bonding of composites. First, a general analysis of adhesive joints based on the sublamine analysis methodology (Flanagan and Chatterjee) was shown to be capable of predicting the peel and shear stress distributions in joints of arbitrary lap-like configuration and loading. In another work the nonlinear adhesive constitutive behavior was accounted for in a combined closed-form/numerical calculation of the joint shear stress for joints loaded under in-plane shear (Kim and Lee). Both of these analysis techniques are founded on closed-form model development, but take advantage of current computer technology to obtain solutions. Such analyses remain ultimately useful for the study of the effects of joint parameters on performance of the joint. There are three combined experimental and analytical papers contained in this section. They focus on the development of a test specimen configuration suitable for the strength measurement of lap joints loaded under in plane shear (Tomblin, Seneviratne, Kim, and Lee), and the investigation of a new double-strap joint design configuration (Qian and Sun) that makes use of extra attachments to improve significantly the joint strength. The fifth paper in this subgroup includes the correlation between analysis and testing of thick section thermoplastics composite-to-titanium for a marine application (Leon, Trezza, Hall, and Bittick). The final paper of the section addresses the often controversial issue of "bondable" peel ply application for bonding fiberglass skins to a polyamide honeycomb core (Kieronski, Knock,
Fallon, and Walker). This work indicated that the adhesion appears to be dominated by a mechanical interlocking mechanism in this particular assembly.

Adhesively Bonded Repair

Two papers in this book focus on the topic of repair. The repair of new armor concepts that are to be used on advanced composite military vehicles was investigated, with particular focus on characterizing the dynamic response of the adhesive joints formed in scarf repairs (Gama, Mahdi, Cichanowski, Yarlagadda, and Gillespie). A split Hopkinson pressure bar was used for these experiments. The repair of thick steel structures used in earth excavation equipment was reported by another group of authors (Roach, Rackow, and Dunn). Bonded composite patches were argued to be more capable than welded repairs for suppressing crack growth in these structures. A primary aspect driving the success of this use of bonded composite repair technology was in determining the best surface preparation technique specifically compatible with both the structure and the application environment.

Bolted Attachments

The four papers contained in this section are on the topic of mechanically-fastened joints. The first in this series gives an overview of the history of bolted and riveted composite joint analyses (Hart-Smith). While these analyses have largely been empirically based, the author projects into the future and describes a physically-based method for joint analysis employing the Strain Invariant Failure Theory (SIFT). Two other works in this section are focused on bolted joint failure prediction. In the first of these, the bolted joint analysis code IBOLT is described in detail (Eisenmann and Rousseau). This code is capable of analyzing multiaxially loaded composite joints with various bypass and bearing loading ratios. The second paper demonstrates the use of nonlinear finite element analyses for predicting failure in composite joints based on lamina-level failure criteria (Bau). These predictions were correlated with experimentally-measured ultimate strength databases. Finally, the last paper in this book focuses on the use of standardized ASTM test methods for obtaining filled hole and bolted attachment allowables (Sawicki). Fastener-hole clearance was identified as a key parameter governing composite filled hole strength.

Areas of Future Research

An open forum discussion among the attendees of this symposium was held to discuss the challenges that need to be addressed in the area of joining and repairing composites. The discussion was focused on adhesive joints, particularly on the topic of standardized methods for measuring properties, and for evaluating joints specifically having composite adherends; it was pointed out that most test methods are developed for metal adherends. Determining adhesive properties was of considerable concern among the industrial participants. Existing test methods, e.g., ASTM D 5656 thick adherend, have been cited as being difficult and sometimes nonrepeatable. Ultimately, empirically and theoretically based investigations are needed in order to establish relationships between bulk-measured properties and joint properties where the adhesive exists as a highly confined thin layer. Finally, the scarcity of
information on the dynamic properties of adhesives, as well as the creep behavior of joints were also cited as topics of needed activity.

Hyonny Kim  
Purdue University  

Keith T. Kedward  
University of California, Santa Barbara  

Symposium Co-Editors
SECTION I:
ADHESIVELY BONDED ATTACHMENTS
Application of a Sublaminate Method to the Analysis of Bonded Joints


ABSTRACT: The sublaminate method consists of using stacked and interconnected plates to evaluate interfacial tractions. A high-order plate theory that includes shear and through-thickness stretching is used for each layer. For composites, the stacking sequence information is included. Because the method is an accurate and convenient way to evaluate debond between layers, it is natural to apply the technique to bonded joints. Previous work had focused on exact solutions of these systems. To create a practical tool for bonded joints, nonlinear material properties had to be included. This was accomplished with an approximate method using the P-element technique. One unusual feature is that the material property distribution is approximated using the same functions. The paper outlines the method, and gives examples that highlight the capability of the code. In particular, the bending behavior of joggled joints can be evaluated. The code can also be used to determine strain-energy-release rate for an existing crack between layers.

KEYWORDS: bonded joint, laminate, sublaminate, fracture, adhesive

Introduction

An analysis code called SUBLAM has been under development at the Materials Sciences Corporation. The code uses a sublaminate approach that allows laminated plates to be stacked. The plate theory includes shear deformation and through-thickness stretching. Within a single plate (or sublaminate), laminate stacking information is used to develop stiffness matrices that depend on the distribution of material, similar to classical lamination theory. The plate theory has also been extended to the case of a cylindrically curved plate. The approach allows one to evaluate the tractions at the plate interfaces using the plate equilibrium equations. Using the equilibrium equations yields a better representation of the forces that tend to debond layers than conventional displacement based methods such as finite elements. An unusual feature of SUBLAM is that all of the coupled plate equations for a linear problem can be solved in closed-form. This means that there is no discretation error in the method. In addition, plates can be combined in a manner similar to the finite element method, and general boundary conditions can be applied at the edges of plates. These features allow one to model complex structural elements. Figure 1 shows some of the classes of problems that can be solved using the method. Reference [1] discusses the use of the method for problems involving crack propagation and the determination of strain-energy-release-rate.

This paper focuses on the use of the sublaminate method for bonded joints. In this regard, an adhesive layer is treated mathematically like an additional sublaminate layer.

1 Technical Director, Materials Sciences Corporation, Fort Washington, PA 19034.
2 Senior Scientist, Materials Sciences Corporation, Fort Washington, PA 19034.
Thus, all of the stiffness properties of the adhesive are taken into consideration, not just the shear stiffness. The method allows one to rapidly analyze complex joints with greater flexibility in applying boundary conditions than is possible with most existing bonded joint codes. One major advantage over some existing approaches is that bending behavior of the joint is included in the analysis. 

The exact, closed-form solution method used in SUBLAM is limited to linear material properties. For greater utility, the code had to be extended to handle nonlinear adhesives. Thus, an approximate solution was added. The approximate solution is based on the P-element approach in which the order of the interpolation functions can be increased until convergence is obtained. This approach allows for large elements, similar to the models employed with the exact solution. With the approximate solution, the equilibrium equations are still used to obtain the interfacial tractions. Thus, part of the accuracy advantage of the method is retained. Exact and approximate elements can be mixed in a single model.

![Diagram of capabilities of the SUBLAM code.](image)

**FIG. 1—Capabilities of the SUBLAM code.**

**Theoretical Approach**

A sublaminate analysis is defined by the use of a plate theory to describe a portion of the total thickness of a composite laminate. The complete laminate is represented by two or more stacked sublaminates. The interface tractions between the plates, as determined from the plate equilibrium equations, can be used to find the interlaminar stresses. Pagano [2] used a similar approach to determine the free-edge stress distribution in laminates. Whitney applied a high-order plate theory to analyze the double-cantilever-
beam (DCB) specimen [3], and the strain-energy-release-rate (SERR) for an edge delamination [4]. Armanios and Rehfield [5] used the sublaminate method, with a shear deformable plate theory, to determine the Mode I and II components of the total SERR for edge delaminations. Chatterjee [6] applied a similar plate theory to analyze Mode II fracture toughness specimens.

**Plate Theory**

The selected displacement field assumes a linear distribution of \( u \) and \( v \) displacements, and a quadratic distribution of \( w \) displacements. This gives a plate that is shear deformable, and that allows stretching through the thickness. Using the coordinate system shown in Fig. 2, the displacement field is

\[
\begin{align*}
  u(x, y, z) &= \frac{1}{2}[u_2(x, y) + u_1(x, y)] + \frac{z}{h}[u_2(x, y) - u_1(x, y)] \\
  v(x, y, z) &= \frac{1}{2}[v_2(x, y) + v_1(x, y)] + \frac{z}{h}[v_2(x, y) - v_1(x, y)] \\
  w(x, y, z) &= \frac{1}{2}[w_2(x, y) + w_1(x, y)] + \frac{z}{h}[w_2(x, y) - w_1(x, y)] + \\
  \Psi_w(x, y) &= \left(\frac{2z}{h}\right)^2 - 1
\end{align*}
\]

For convenience when stacking sublaminates, we have chosen to express the displacement field in terms of surface quantities, rather than the traditional midplane quantities. This represents a simple change of variables, and does not influence the mechanics of the plate problem. \( \Psi_w \) is a generalized displacement coefficient associated with a quadratic term in the \( w \) displacement, needed so that a linear distribution of \( \varepsilon_z \) strain can be represented.

![FIG. 2—Coordinate system for single sublaminate.](image)

A variational approach is taken to derive the equilibrium equations and natural boundary conditions. The strain-energy density per unit area is given by

\[
U = \frac{1}{2} \int_{h/2}^{h/2} \left( \bar{\varepsilon}^T C \bar{\varepsilon} - 2\Delta T \bar{\alpha}^T C \bar{\varepsilon} + \Delta T^2 \bar{\alpha}^T C \bar{\alpha} \right) dz
\]

where \( \Delta T \) is the change in temperature from a stress-free condition, and, in contracted notation
\[ \bar{\varepsilon} = \{ \varepsilon_1, \varepsilon_2, \varepsilon_3, \varepsilon_4, \varepsilon_5, \varepsilon_6 \} \]
\[ \bar{\alpha} = \{ \alpha_1, \alpha_2, \alpha_3, 0, 0, \alpha_6 \} \]  

Symbols shown in bold represent a matrix. Thirteen elastic constants, \( C_{ij} \), are needed to describe an orthotropic ply with an arbitrary orientation in the \( x-y \) plane (monoclinic material). The \( \alpha_i \) are the ply thermal expansion coefficients. The integration of Eq 2 through the thickness proceeds stepwise to account for the changing material properties with each ply. We define the following integrations
\[ \{ A_i, B_i, D_i \} = \int_{h/2}^{h/2} C_{ij} \{ 1, z, z^2 \} \, dz \quad (i, j = 1, 2, ..., 6) \]
\[ \{ N_i', M_i', R_i' \} = \Delta T \int_{h/2}^{h/2} C_{ij} \alpha_j \{ 1, z, z^2 \} \, dz \quad (i, j = 1, 2, 3, 6) \]

The \( A, B, \) and \( D \) matrices are similar to those defined in classical lamination theory, except that plane stress assumptions cannot be made. \( N_i', M_i', \) and \( R_i' \) are plate resultants of effective thermal loads. \( N_i', M_i' \) are the conventional thermal result load and moment, assuming a fully constrained laminate. \( R_i' \) is a higher-order moment of the thermal stress. It appears as a consequence of the assumed displacement field, but it does not correspond to a load with any conventional engineering meaning. In addition, we require the following higher order moments for the shear stiffness distribution
\[ (D_y^{(3)}, D_y^{(4)}) = \int_{h/2}^{h/2} C_{yj} \{ z^3, z^4 \} \, dz \quad (i, j = 4, 5) \]

The work due to external forces, per unit area, is given by
\[ V = u_1 s_1 - u_2 s_x + v_1 t_1 - v_2 t_2 + w_1 p_1 - w_2 p_2 \]
where \( s_i, t_i, \) and \( p_i \) are the tractions in the \( x, y \) and \( z \) directions respectively, for the \( i \)'th surface. The total potential per unit area is then
\[ \Pi = U + V \]
Using variational principles to assure that the first variation of the potential vanishes results in seven equilibrium equations in terms of the surface displacements and \( \Psi_{uv} \).

The natural boundary conditions for the faces of the plates are also determined from the variational principle. The natural boundary conditions on the \( x-y \) faces of the plates are in terms of six nodal forces, plus one generalized force in the \( z \) direction. A typical equation for the nodal forces is
\[ F_{x1} = \frac{\partial \Pi}{\partial u_{1,v}} \]
where \( F_{ij} \) is the force in the \( i \) direction \( (i=x,y,z) \), applied at the \( j \)'th surface of the plate \( (j=1,2) \). The nodal lines boundary conditions can be related to plate force resultants by
\[ F_{x1} = \frac{1}{2} N_6 - M_z/h \]
\[ F_{x2} = \frac{1}{2} N_6 + M_z/h \]
\[ F_{y1} = \frac{1}{2} N_2 - M_z/h \]
\[ F_{y2} = \frac{1}{2} N_2 + M_z/h \]
where $\Gamma_z$ is a generalized force, and
\begin{align*}
\{N_i, M_i\} &= \int_{h/2}^{h/2} \sigma_i \{1, z\} \, dz \\
\{V_4, R_4, S_4\} &= \int_{h/2}^{h/2} \sigma_4 \{1, z, z^2\} \, dz
\end{align*}

The higher moments of the vertical shear, $R_4$ and $S_4$, are not classical plate resultants, but are formally required based on the assumed displacement distribution.

A similar derivation is used for the case of a cylindrically curved plate. The curved plate is useful in modeling the details of typical composite cross sections. These sections often have small radius to thickness ratios, and therefore, thin-shell approximations cannot be made.

**Exact Solution**

If one assumes that all of the surface displacements are uniform in the $x$-direction, as in a generalized plane-strain case, then it is possible to solve the governing equations for the coupled plate problem in closed-form. Making the plane-strain assumption leads to a system of ordinary differential equations. These equations can be expressed in the following matrix form
\begin{equation}
H_0 u + H_1 u' + H_2 u'' + P = t
\end{equation}

where primes indicate differentiation with respect to $y$, and
\begin{align*}
u &= \{u_1, v_1, w_1, \Psi_w, u_2, v_2, w_2\} \\
t &= \{s_1, t_1, p_1, 0, s_2, t_2, p_2\}
\end{align*}

The vector $P$ contains functions of the applied axial strain and thermal loads. The vector $t$ contains the interface tractions, where $s, t, p$ are in the $x, y,$ and $z$ directions, respectively. This system of equations can be expanded to include multiple, stacked plates. To assemble the expanded system, we superimpose the surface tractions so that there are zero net tractions on the internal interfaces. The assembly process also accounts for the shared displacements at the interface. Using surface quantities in Eq 1 simplifies the assembly process (note that the quadratic term associated with $\Psi_w$ evaluates to zero at the interfaces).

The assembly procedure described above yields homogeneous system of equations, plus a nonhomogeneous part due to the thermal expansion terms and uniform axial strain. Assume that solutions to the homogeneous part of Eq 11 have the form
\begin{align*}
u(y) &= c e^{\beta y} \\
u'(y) &= \hat{c} e^{\beta y}
\end{align*}

The dummy variable $\hat{c}$ is introduced so that a system of first order equations can be obtained. Substituting Eq 12 into Eq 11, and assuming there are no surface tractions present, yields the following general eigensystem
8 JOINING AND REPAIR OF COMPOSITE STRUCTURES

\[
\begin{bmatrix}
    H_0 & 0 \\
    0 & I
\end{bmatrix}
\begin{bmatrix}
    \epsilon \\
    \dot{\epsilon}
\end{bmatrix} + \beta
\begin{bmatrix}
    H_1 & H_2 \\
    -I & 0
\end{bmatrix}
\begin{bmatrix}
    \epsilon \\
    \dot{\epsilon}
\end{bmatrix} = \{0\}
\]

(13)

where \( I \) is an identity matrix. The assembled submatrices \((H_j)\) will be of order \(8n+6\), where \( n \) is the number of sublaminates.

The eigensystem defined by Eq 13 will result in \(8n-2\) nonzero values of \( \beta \) for a flat plate, and \(8n+2\) nonzero values for a curved plate. Additional solutions can be found by assuming displacement functions with polynomial forms. The undetermined coefficients can be found by substituting the assumed polynomials into Eq 11, matching powers of \( y \), and solving for the resulting linear system. The system is singular, requiring the use of singular value decomposition. More details of this procedure are given in [2]. For cylindrical plates, the eigensystem given in Eq 13 results in repeated roots. Additional solutions can be found using standard differential equation procedures, recast for arbitrarily large systems of equations. The details are tedious, but are outlined in [1].

**Exact Finite Elements**

The natural boundary conditions at the edges of the plate can be expressed in matrix form as

\[
f = F\bar{\epsilon} + f_p
\]

(14)

where the vector \( \bar{\epsilon} \) can contain any of the undetermined function coefficients defined above (\( \epsilon, \ddot{\epsilon} \), and undetermined coefficient from the polynomial solutions). The vector \( f \) can be interpreted as forces on the nodal lines located at the corners of the plate on the \( y \) faces (see Fig. 2), plus the generalized forces. \( f_p \) is a vector of nodal forces that result from the applied axial strain and thermal loads.

Similarly, the displacements at the nodal lines can be expressed as

\[
d = D\bar{\epsilon} + u_p
\]

(15)

Eliminating the vector of coefficients between Eqs 14 and 15 yields a direct relation between nodal line forces and displacements

\[
f = FD^{-1}(d - u_p) + f_p
\]

or

\[
k d = f - \dot{f}_p
\]

(16)

where

\[
k = FD^{-1}
\]

\[
\dot{f}_p = f_p - k u_p
\]

The matrix \( k \) has all the properties of a stiffness matrix, and may be assembled in the manner of the finite element method. Boundary conditions and nodal forces may also be applied in the same manner as in the finite element method. Once the nodal displacements, \( d \), have been found from Eq 16, the function coefficients are computed using

\[
\bar{\epsilon} = D^{-1}(d - u_p)
\]

(17)

Once the function coefficients have been obtained, the interfacial tractions can be computed by substituting the results back into the equilibrium equations (Eq 11) The
boundary condition equations (Eqs 8 and 9) can be used to compute the plate force resultants for any value of \( y \) within the plate.

**Approximate Solution Approach**

Exact solutions cannot be found if the material properties are nonlinear. The approximate solution uses a polynomial series to describe the displacement distribution. The approximate solution uses a Legendre polynomial series. The number of terms in the series can be selected to meet some convergence criterion. Thus, the method is very similar to the P-method approach used in certain finite element codes. In the P-method approach, convergence is achieved by increasing the order of the approximating function. This contrasts to the H-method in which convergence is achieved by decreasing the element size. The P-method is well suited to SUBLAM because it allows one to model with a small number of large elements, similar to the approach taken when the exact solution is employed. The P-method was also well suited to this problem because it allows for continuity between sublaminates along the entire interface, while keeping the end boundary conditions referenced to nodal lines. The solution uses a strain-energy minimization method, similar to the finite element method.

The approximate elements integrate seamlessly within the SUBLAM system. The input required is nearly identical to the exact elements. Exact and approximate elements can be mixed within a single model. The approximate solutions use most of the same subroutines for evaluating interface tractions, strain-energy-release-rate, and other output quantities.

A conventional P-element interpolation function is employed. The function is defined over the region \( \xi = -1 \) to 1, as

\[
\Psi(0, \xi) = \frac{(1 - \xi)}{2} \\
\Psi(1, \xi) = \frac{(1 + \xi)}{2} \\
\psi(m, \xi) = \frac{(P_m(\xi) - P_{m-2}(\xi))}{\sqrt{2(2m-1)}} m > 1
\]

where \( P_m \) is a Legendre polynomial of order \( m \). The first two terms are linear, nodal interpolation functions. The remaining terms are zero at the endpoints (Fig. 3). The total displacement as a function of \( \xi \) is then given by

\[
u(\xi) = u_0 \Psi(0, \xi) + u_1 \Psi(1, \xi) + \sum_{m=2}^{\text{term}} \psi(m, \xi)
\]

where \( u_0 \) and \( u_1 \) are nodal values, and the \( u_m \), for \( m > 1 \), are generalized displacements.
For numerical reasons and internal self-consistency, SUBLAM uses the same Legendre polynomial series to describe the material property variation as is used to represent the displacements. In addition, the same order polynomials are applied. When the material properties are updated, the property as a function of local strain is computed for a series of points. The Legendre polynomial is then computed using a least-squares method.

Because the polynomial order is variable, the energy integration scheme must also be flexible. SUBLAM uses a Gaussian integration method with a variable number of integration points. A heuristic relation between the number of Gaussian points and the polynomial order has been established to assure the energy integrals are accurate. Exact integration is theoretically possible, but the Gaussian method was found to be faster and equally precise.

The method for handling material nonlinearity is not theoretically restricted to the transverse shear modulus, and could be applied to all the properties of an orthotropic material. Recall that an adhesive layer in SUBLAM is not any different than any other sublaminate layer. However, the current implementation of method only updates the transverse shear modulus based on the effective layer shear strain

$$\gamma_{eff} = \sqrt{\gamma_{xx}^2 + \gamma_{zz}^2}$$  \hspace{1cm} (20)

The solution is not restricted in the form of the material model. The nonlinear version of SUBLAM has the capability of accepting the definition of a material shear stress-strain curve in terms of either a Ramberg-Osgood fit, an elastic-perfectly-plastic model, or direct table lookup.

**Iteration**

The solution also requires an iteration scheme to achieve equilibrium. A secant modulus approach was chosen because it was relatively easy to implement, and it converges reliably even for highly nonlinear materials. Figure 4 illustrates the scheme. The disadvantage of this approach is that a large number of iterations may be required to achieve a specified degree of convergence.
Fracture Calculation

For stress analysis of laminated composites, laminated plate theories of various orders and sublamine assemblage models are often employed. Obviously, the delaminations considered are between two adjacent laminae. It is known [7] that the displacements $u, v, w$ (in $x, y, z$ directions, $z$ being the thickness coordinate) have to be continuous across the delamination front (or periphery) but their spatial derivatives with respect to $n$, $n$ being measured in the direction normal to the front (in $x$-$y$ plane) are usually discontinuous. As a result, the gradients of the displacement discontinuities across the delamination surfaces with respect to $n$ do not have the inverse square root singularity at the delamination front (as in the case of elasticity solution for homogeneous materials), but they have finite values. It will be illustrated later with a simple example that the discontinuities in the gradients of the displacements at the delamination front yield singularities in stress fields in the form of interactive concentrated line forces at the front (between each of the sublaminates used for stress analysis). This is illustrated in Figs. 5-7 with the opening or peel mode displacements and associated tractions near a delamination tip in a 2-D problem. In this case, the direction $n$ (normal to delamination front) coincides with $y$-axis.

FIG. 5—Opening mode deformation due to delamination between sublaminates 1 and 2 on the right hand-side of front.
In using Irwin's virtual crack closure technique \[8\] one needs to consider a virtual self similar extension of the delamination by an amount \( \delta a \) as shown in Fig. 7. In this case, it is in the negative \( y \) direction. Since we are considering infinitesimal extension, the calculated displacements (shown in Fig. 6) for the extended delamination are the same as those for the original delamination. Only the origin or the tip is shifted to the left by an infinitesimal amount \( \delta a \) and the displacements and tractions for the extended delamination should now be considered as functions of \( y_1 \) (instead of \( y \)) measured from the shifted tip. The energy release rate is computed as the work required to close the extended delamination to the original configuration (work done by the pressure which is equal and opposite the interactive traction \( t_3(y) \) between the sublaminates 1 and 2 for the original configuration on the negative of the opening displacement for the extended case), divide it by the area of extension \( (b \ \delta a, \ b \) being the width or dimension in \( x \) direction, which will be taken as equal to unity) and take the limit \( \delta a \to 0 \). For elastic case

\[
G_i = \text{Limit}_{\delta a \to 0} \left[ \frac{1}{2\delta a} \int_0^{\delta a} t_3(y) w^*(y_1) dy_1 \right]
\]  

(21)
It may be noted from Fig. 6 that the traction \( t_3(y) \) consists of the distributed traction \( p(y_1) \), which does not contribute to \( G_I \) (because of the limit \( \delta a \to 0 \)) and the concentrated line force \( F_3 \) acting at \( y_1 = \delta a^{-} \) or \( y = 0^{-} \). Since \( w* = 0 \) at \( y_1 \leq 0 \), one can write a Taylor series expansion for \( w*(y_1) \) for positive values of \( y_1 \), i.e.,

\[
  w*(y_1) = w*' (O^+) + w** (O^+) y_1^2 / 2 + ...
\]

where ' denotes the derivative of \( w* \) with respect to \( y_1 \). Since the concentrated line force can be considered as a delta-dirac function, it follows that

\[
  G_I = F_3 w*' (0^+) / 2
\]

where \( F_3 \), the concentrated nodal force and \( w** (0^+) \), the gradient of the opening displacement are known for the original configuration. Similar calculations are required considering concentrated line forces \( F_2 \) and \( F_1 \) in directions normal (\( n \) or \( y \)) and tangential (\( t \) or \( x \)) to the delamination front and the corresponding displacement (sliding and tearing) discontinuity gradients \( v*(0^+) \) and \( u*(0^+) \) at the front to compute \( G_{II} \) and \( G_{III} \).

\[
  G_{II} = F_2 v*(0^+) / 2
\]

\[
  G_{III} = F_1 u*(0^+) / 2
\]

Examples

Double Lap Joint

A simple double-lap joint is a convenient problem for exploring some of the features of this analysis approach. The model being considered is shown in Fig. 8. The adherend layers are aluminum. The adhesive is \( 0.16h \) (0.20 mm) thick. For the linear examples, the adhesive is treated as an isotropic material with a shear modulus of 4.1 GPa.

One strength of the sublaminar method is the ability to accurately represent the interfacial tractions between sublaminates. The shearing interfacial traction is a measure that could be used to predict adhesive failure. However, some care must be used in interpreting these tractions. The first problem is that the adhesive layer has two interfaces, and the tractions may not be identical. Figure 9 shows the interfacial tractions in the boundary region at the left side of the joint. Also shown on the graph is the average adhesive layer shear. This is obtained by evaluating the layer natural boundary conditions to determine a net shear force. Dividing by layer thickness gives an average shear stress. As expected, the average shear stress falls between the bounds of the interface values. The average value approaches zero as required by the stress boundary conditions. The average does not exactly equal zero because of the shared boundary conditions with the adjacent adherend layers. The nodal degrees-of-freedom do not represent sufficient boundary conditions to independently satisfy the free-edge conditions for each layer.
JOINING AND REPAIR OF COMPOSITE STRUCTURES

FIG. 8—Double-lap-joint model.

FIG. 9—Comparison of adhesive layer tractions and average layer stress, local detail.

Figure 9 has been truncated to show only positive shears. The interface values overshoot, and have large negative magnitudes at \( y=0 \). The sudden traction reversal near the transition can be viewed as the plate theory response to the singularities that would be present in an elasticity solution. In other problems, such as the free-edge stresses in a laminated composite, these extremely large gradients correlate well with elasticity results. In the case of a bonded joint, values that change rapidly in a dimension comparable to the bondline thickness are probably best treated as artifacts of exact solution method. For the purposes of joint evaluation, the peak stress shown in Fig. 9 is more meaningful. This value can be shown to be stable with respect to modeling details, and additional through-thickness discretization. The peak stress exactly at the free-edge is sensitive to modeling details. If the interface is to be evaluated, then fracture methods are recommended.

The tractions corresponding to peel stress are shown in Fig. 10. Again, the plot has been truncated because the edge values are very large. The near-vertical solid line at \( y=1 \) is the sudden reversal in value for one of the interfaces, going from a large negative value, to a larger positive value in a distance equal to a fraction of the bondline thickness. This shows the power of the exact solution method to extract rapidly changing stress. For the purpose of failure prediction, fracture mechanics is probably more meaningful.
For linear material properties and uniform thickness elements, the exact method is available. The P-method can be applied to the same joint for the purpose of determining approximate solution accuracy. Although the P-elements may be large, experience shows that it is impractical to model an entire joint with a single element. The approximation functions will attempt to follow the large edge tractions discussed above and therefore generate large errors elsewhere. A better modeling approach is to provide small “sacrificial” elements at the joint ends, as shown in Fig. 11. The joint is also subdivided in the center to further increase accuracy. The SUBLAM code allows one to mix exact and approximate elements. In this model, the extensions to the adherends beyond the joint region are exact elements.

The number of terms in the Legendre polynomial approximation function, referred to as the P-Order determines the convergence of the method. Figure 12 compares the results near the left edge of the joint for a low-order solution (P=4), and a higher order solution (P=10). One distinguishing difference is the presence of a discontinuity at the boundary between two elements (y/b=0.016). The element boundary conditions do not enforce continuity of the interface tractions. Figure 13 shows how the magnitude of the discontinuity decreases with the P order. For this problem, P=10 gives an accurate result. Figure 14 shows the shear traction near the left edge of the joint for an exact solution and an approximate solution with P=10. The plot also includes a distribution for a classical Volkerson-type solution which assumes that all axial load is carried by the adherends, and all shear deformation is in the adhesive. Peel stress is given in Fig. 15.
JOINING AND REPAIR OF COMPOSITE STRUCTURES

FIG. 12—Comparison of low and high p-order solutions for adhesive layer shear traction, local detail.

FIG. 13—Maximum discontinuity in tractions (relative to average shear) at element boundary versus p-order.

FIG. 14—Comparison of exact and approximate solutions for average shear in adhesive layer, local detail.
The same joint configuration was used in conjunction with an elastic-perfectly-plastic adhesive model. The plastic stress was set to 21 MPa. Figure 16 shows the average shear stress for three values of applied load, where $\tau_{avg} = N_y/b$. The elastic-plastic material model is challenging for the method because the sudden change in slope cannot be represented exactly with the continuous approximation functions. Getting acceptable results required further subdividing the model into a total of 6 elements, counting the 2 small sacrificial elements at the ends. Figure 16 shows that the solution tends to overshoot and oscillate at the points where the stress reaches the plastic value. The figure shows the ability of the method to follow the transition between elastic and plastic regions, independent of the location of the element boundaries.

FIG. 15—Comparison of exact and approximate solutions for normal traction (peel) for adhesive layer. Local detail.

FIG. 16—Shear stress distributions for elastic-plastic material model.
Joggled Joints

To further demonstrate the utility of the methodology, we next consider three forms of an overlap joint, as shown in Fig. 17. The first is a straight overlap. The second uses a joggle at each end of the joint so that the load line runs along the center of the adhesive layer. The third joint applies a single joggle such that the two adherends are parallel. An exact, linear solution will be used. When dealing with eccentric joints, it must be emphasized that the existing code does not include large-deflection, geometric nonlinearity. Figure 18 shows the deformed shape for a finite element model of the symmetric overlap joint. The finite element solution uses a commercial P-element approach. The contours show variations in axial stress.

The boundary conditions and overall dimensions for one of the joints is shown in Fig. 19. Each of the models was constructed to have the same overall dimensions. The loading introduction is by a uniform displacement at the ends, with the magnitude of the displacement selected such that the integrated load is unity. The adherends are E-Glass vinylester (\(E_x=E_y=23 \text{ GPa}, G_{xy}=G_{yz}=G_{zx}=1.1 \text{ GPa}, v_{xy}=0.17\)), with a [(45/0)\(2/0\)]_s layup. The adhesive layer is 0.15h thick, with a shear modulus of 1.4 GPa.

![FIG. 17—Deformed plots for (a) straight overlap (b) symmetric joggle and (c) single joggle joints. Deformations are to the same magnification scale.](image1)

![FIG. 18—Deformed plot for P-based finite element solution of symmetric joggle](image2)

![FIG. 19—Boundary conditions and dimensions for joggled joint.](image3)
Figure 20 shows the average adhesive shear stress for each of the joints. The two symmetric joints have a symmetric stress distribution about \( y=0.5 \), while the single joggle has a higher stress at \( y=0 \). A surprising result is that the peak stresses are actually lower for the straight overlap than for either of the joggled configurations. There is a constant shear load in the straight overlap so that the shear stress distribution never goes to zero at the center of the joint. This constant shear comes from the particular choice of end boundary conditions, in particular vertical restraints at each end, and serves to transfer a portion of the load. Also shown on the plot is the shear stress distribution from the finite element (FE) model shown in Figure 18 for the symmetric joggle. The FE results track reasonably well, but there is an inherent difference in the solutions. This is believed to be related to the difference in bending stiffness between the models. The SUBLAM result uses the bending stiffness terms from lamination theory, whereas the FE model treats the adherend as a smeared orthotropic solid.

The peel stress distribution (Fig. 21) is more difficult to interpret because of the extreme stress gradients at the ends. However, the average peel stress (integrated over an arbitrary distance of two adhesive thicknesses) for the straight joint is less than either of the joggled joints by a substantial factor. The single joggle joint has an average peel stress at the edge that is twice the value for the straight joint. The FE result is also included for comparison.

FIG. 20—Comparison of average adhesive shear stress for three joint types.
A more meaningful measure of peel is to look at the strain-energy-release-rates ($G$) for joints with pre-existing cracks. Each of the models was modified to include a $2h$ debond between the adhesive layer and the adjacent adherent. Figure 22 shows a detail of one model with the crack under load, using the same boundary conditions as before. The resulting values of $G$ are shown in Table 1. Again, one obtains the counter-intuitive result that the straight joint is predicted to have better performance than either of the joggled configuration, for both Mode I (peel), and Mode II (shearing) modes of crack growth. There are significant differences in the three cases. The difference in $G$ between cases is much larger than the difference in overall joint compliance. Therefore, one must look at changes in local bending and transverse shear energy with crack extension to account for the energy release.

These results are highly dependent on the choice of boundary conditions. The eccentric joints have a different response if one of the vertical restraints is removed, or if uniform load is applied instead of uniform displacement. The example highlights the importance of an analytical tool that can handle general boundary conditions, and has the flexibility to model realistic situations.

FIG. 21—Comparison of interface peel stress for three joint types. detail near left transition.

FIG. 22—Local detail of joggled joint with bondline crack.
TABLE 1—Strain-energy-release-rates for a crack length of 2h.

<table>
<thead>
<tr>
<th>Joint</th>
<th>$G_{ij} / (N_i^2 / E_x h) \times 10^6$</th>
<th>$G_{ij} / (N_i^2 / E_x h) \times 10^6$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Straight Overlap</td>
<td>0.85</td>
<td>0.34</td>
</tr>
<tr>
<td>Symmetric Joggle</td>
<td>1.27</td>
<td>0.60</td>
</tr>
<tr>
<td>Single Joggle</td>
<td>3.61</td>
<td>1.30</td>
</tr>
</tbody>
</table>

Summary

The examples demonstrate that the sublaminate approach is an effective and efficient method for analyzing bonded joints. The addition of approximate elements makes it possible to use the methodology for nonlinear adhesive properties. Verification examples show that the approximate technique matches well with the exact solutions. The ability to analyze joints with realistic boundary conditions is an important capability.

The authors wish to acknowledge the support of the FAA Volpe Center, and the assistance of Dr. Peter Shyprykevich.

References


Adhesive Nonlinearity and the Prediction of Failure in Bonded Composite Lap Joints


ABSTRACT: A theoretical model is derived that predicts failure in adhesively bonded lap joints loaded by in-plane shear. This model is based on shear lag assumptions and accounts for a nonlinear adhesive shear stress-strain relationship so that the development of plastic strain is tracked. Failure of the joint is determined when the plastic strain reaches its ultimate value. By accounting for adhesive plasticity, this model permits the design of joints with higher load-carrying efficiency than designs based on simple elastic-to-failure adhesive constitutive behavior. Example calculations presented in this paper show that the theoretical model predicts failure load accurately for a carbon/epoxy adherend joint (within 6%), in comparison with finite element analysis (FEA) predictions, for joints with adhesive bondlines of 0.33 mm or less. For the thicker bondlines studied, a severe strain localization effect was observed in the FEA models to occur at the joint interface corners, and therefore the theoretical model over-predicted failure load by up to 22% for 2.08 mm bondline carbon/epoxy adherends.

KEYWORDS: adhesively bonded joint, plasticity, failure prediction, in-plane shear

Introduction

An adhesively bonded lap joint loaded by in-plane shear is a generic structural configuration found in bonded composite assemblies. Some examples are the fuselage splice joint and the bonded wing leading edge of modern small aircraft. These structures carry significant torque loads in the form of in-plane shear flow that must be transferred across the joints. In general, bonded composite structures are designed to be loaded only up to their elastic limit. However, when designing for ultimate load, if the structure can operate to the joint's failure limit, the structure can be used more efficiently.

The analytical treatment of a bonded lap joint where the adherends are loaded in tension has been considered extensively by many authors. Hart-Smith [1, 2] extended the shear-lag model that was presented by Volkersen [3] to include elastic-to-perfectly plastic adhesive behavior. Goland and Reissner [4] and Oplinger [5] accounted for adherend bending deflections to predict the peel stress in the adhesive. Tsai, Oplinger, and Morton [6] provided a correction for adherend shear deformation, resulting in a
simple modification of the Volkersen's theory based equations. Nguyen and Kedward [7] introduced a nonlinear adhesive constitutive model composed of three fitting parameters and used it to predict the adhesive shear strain distribution of a tubular adhesive scarf joint loaded to failure in tension.

Adhesively bonded lap geometries loaded by in-plane shear have been discussed by Hart-Smith [1], van Rijn [8], and the Engineering Sciences Data Unit [9]. The authors of these works indicate that shear loading can be analytically accounted for by simply replacing the adherend Young's moduli in the tensile loaded lap joint solution with the respective adherend shear moduli. This assumption is valid only for simple cases with one dimensional loading, whereas in-plane shear loaded joints are generally two or three dimensional.

Although finite element analysis (FEA) can be applied to predict failure limit accurately, FEA is a time consuming process and may not easily be performed for all joint configurations. Due to the inherent three-dimensional nature of the joint geometry and shear loading conditions, three-dimensional elements need to be used in FEA modeling of shear flow transfer across a lap joint. Creating a mesh having enough element refinement to capture the high stress gradients in the thin adhesive layer can easily result in a FEA model of unsolvable size. Failure limit load predictions by simple theoretical methods are therefore quite useful if they can provide accurate predictions for much less effort than FEA.

The objective of the research presented in this paper is to establish a theoretical model which can estimate the failure limit of in-plane shear loaded adhesively bonded joints. In order to accurately predict the failure limit, the nonlinear adhesive behavior must be accounted for. Therefore, a two parameter version of the Nguyen and Kedward [7] adhesive material model is used in the derivation of a second order differential equation that governs the adhesive shear strain. The numerical solution to this governing equation permits the calculation of the joint failure load. (i.e., the load at which the failure strain in the adhesive is reached). FEA incorporating nonlinear adhesive behavior is conducted for comparison with the theoretical predictions presented in this paper.

Theory and Solution

Adhesive Constitutive Behavior

The shear stress-strain curve (τ_a vs. γ_a) for a ductile epoxy adhesive can be modeled by a two parameter exponential fitting curve [7], such as Eq. 1. In this equation k and B_1 are fitting parameters chosen in order to match the fitting curve to the experimental stress-strain data, and G_a is the elastic shear modulus.

\[ \tau_a = (G_a - kB_1)\gamma_a + B_1(1 - e^{-k\gamma_a}) \]  

(1)

In Figure 1, the experimentally measured constitutive behavior (by ASTM D5656) of a two part paste epoxy adhesive, PTM&W ES6292 [10], is plotted. Note that in general the failure strain and the ultimate strength of adhesives have been measured to decrease as the adhesive thickness is increased (see in Figure 1). Also it is possible for the final stress at the failure strain to be less than the ultimate strength such that the constitutive curve ends with a negative slope. Fitting curves to the data should be
JOINING AND REPAIR OF COMPOSITE STRUCTURES

capable of reflecting all of these attributes. Note that when fitting to the adhesive shear stress versus strain data, the shear modulus $G_a$ should be chosen so as to fully reflect the slope over the entire elastic range, and not only the slope of the adhesive in the small starting range of the experimental data. For the data plotted in Figure 1, the elastic range was defined using a 0.2% offset rule. A best linear fit was made to the data for each curve. Since the variation in $G_a$ between each curve was no greater than 7%, the average value was used in all subsequent calculations.

Determining Fitting Parameters

The parameters $k$ and $B_1$ are chosen based on the following conditions: (i) the stress at ultimate strain $\gamma_a^{\text{ult}}$ should equal the average between the ultimate and final stress ($\tau_{\text{ult}}$ and $\tau_{\text{final}}$), and (ii) the area of the fitting curve should match the area of the experimental data.

Condition (i) can be expressed using Eq. 1 as

$$ (G_a - kB_1)\gamma_a^{\text{ult}} + B_1(1 - e^{-k\gamma_a^{\text{ult}}}) = \frac{1}{2}(\tau_{\text{ult}} + \tau_{\text{final}}) $$

(2)

The manipulation of Eq. 2 yields an expression relating $B_1$ and $k$.

$$ B_1 = \frac{0.5(\tau_{\text{ult}} + \tau_{\text{final}}) - G_a\gamma_a^{\text{ult}}}{1 - k\gamma_a^{\text{ult}} - e^{-k\gamma_a^{\text{ult}}}} $$

(3)

In order to satisfy condition (ii), the integration of Eq. 1 with respect to $\gamma_a$ between the limits 0 to $\gamma_a^{\text{ult}}$ should be same as the area under the experimentally measured stress-strain curve,

$$ \frac{1}{2}(G_a - kB_1)(\gamma_a^{\text{ult}})^2 + B_1[\gamma_a^{\text{ult}} + \frac{1}{k}(e^{-k\gamma_a^{\text{ult}}} - 1)] = W_{\text{ROT}} $$

(4)

where $W_{\text{ROT}}$ is the total work density of the adhesive and is equivalent to the total area under the experimental data curve.

Finally, $B_1$ from Eq. 3 can be inserted into Eq. 4 resulting in a transcendental equation for $k$ which can be solved numerically, e.g., using standard bisection or Newton methods. For the adhesive data shown in Figure 1, the parameters $k$ and $B_1$ were determined. The results for fits to the data are shown in Figures 2 to 4. Table 1 summarizes the parameters needed by Eq. 1 for each bondline thickness, $t_a$. 

www.polycomposite.ir
Figure 1 - Shear stress-strain data for PTM&W ES6292.

Figure 2 - Fit to data for PTM&W ES6292, $t_a = 0.33$ mm.
Figure 3 - *Fit to data for PTM&W ES6292, $t_s = 1.07$ mm.*

Figure 4 - *Fit to data for PTM&W ES6292, $t_s = 2.08$ mm.*
Table 1 - Adhesive constitutive model fitting parameters \( k \) and \( B_1 \).

<table>
<thead>
<tr>
<th>Adhesive</th>
<th>( t_a ) (mm)</th>
<th>( k )</th>
<th>( B_1 ) (MPa)</th>
<th>( \gamma^{\text{ult}} )</th>
</tr>
</thead>
<tbody>
<tr>
<td>PTM&amp;W ES6292</td>
<td>0.33</td>
<td>36.4</td>
<td>25.4</td>
<td>0.350</td>
</tr>
<tr>
<td>( G_o = 0.923 ) GPa</td>
<td>1.07</td>
<td>35.6</td>
<td>26.2</td>
<td>0.241</td>
</tr>
<tr>
<td>Z08</td>
<td>2.08</td>
<td>36.9</td>
<td>25.8</td>
<td>0.130</td>
</tr>
</tbody>
</table>

**Governing Equation**

The single lap joint shown in Figure 5 is loaded by in-plane shear stress. The differential element in this figure shows the in-plane shear stress acting on the inner and outer adherends, \( r_x^i \) and \( r_x^o \), as well as two components of the adhesive shear stress \( r_{xy}^a \) and \( r_{yz}^a \). The following conditions have been assumed:

- constant bond and adherend thickness
- uniform shear strain through the adhesive thickness
- adherends carry only in-plane stresses
- adhesive carries only out-of-plane shear stresses

In Figure 5, the applied shear stress resultant \( N_{xy} \) is continuous through the overlap region and at any point must equal the sum of the product of each adherend shear stress with its respective thickness \( t_i \) and \( t_o \):

\[
N_{xy} = r_x^i t_i + r_x^o t_o
\]  

(5)

The adhesive shear strains are written based on the assumption of uniform shear strain through the thickness of the adhesive,

\[
\gamma_{xy}^a = \frac{1}{t_a} (u_o - u_i) \quad \text{and} \quad \gamma_{yz}^a = \frac{1}{t_a} (v_o - v_i)
\]  

(6) and (7)

where \( t_a \) is the thickness of the adhesive and \( u \) and \( v \) are the in-plane deformations in each adherend. Differentiating Eq. 7 with respect to \( x \), Eq. 6 with respect to \( y \), and adding the two resulting equations,

\[
\frac{\partial \gamma_{xy}^a}{\partial y} + \frac{\partial \gamma_{yz}^a}{\partial x} = \frac{1}{t_a} (\gamma_{xy}^o - \gamma_{xy}^i) = \frac{1}{t_a} \left( \frac{t_o}{G_o} - \frac{t_i}{G_i} \right)
\]  

(8)

From Eq. 5, the shear stress in the inner adherend can be written as,

\[
r_{xy}^i = \frac{N_{xy} - r_{xy}^o t_o}{t_i}
\]  

(9)

Substituting Eq. 9 into Eq. 8 yields,
JOINING AND REPAIR OF COMPOSITE STRUCTURES

\[
\frac{\partial \gamma_{xy}^a}{\partial y} + \frac{\partial \gamma_{yz}^a}{\partial x} = t_o \left( \frac{\tau_{xy}^o}{t_o} + \frac{\tau_{yz}^o}{t_i} \right) - \frac{N_{xy}}{t_o G_t t_i} \tag{10}
\]

Force equilibrium performed on a differential element of the outer adherend, shown in Figure 6, results in relationships between the adhesive stress components and the outer adherend shear stress.

\[
\tau_{xy}^o = t_o \frac{\partial \tau_{xy}^o}{\partial y} \quad \text{and} \quad \tau_{yz}^o = t_o \frac{\partial \tau_{yz}^o}{\partial x} \tag{11} \text{ and } (12)
\]

![Figure 5 - Lap joint transferring shear stress resultant \(N_{xy}\) and differential element showing adherend and adhesive stresses.](image)

![Figure 6 - Adhesive and adherend stresses acting on element of outer adherend.](image)

Summing the derivative of Eq. 10 with respect to \(x\) with the derivative of Eq. 10 with respect to \(y\), and simplifying using Eqs. 11 and 12 results in,

\[
\frac{\partial^2 \gamma_{xy}^o}{\partial y^2} + \frac{\partial^2 \gamma_{xy}^o}{\partial x \partial y} + \frac{\partial^2 \gamma_{yz}^o}{\partial x^2} + \frac{\partial^2 \gamma_{yz}^o}{\partial x \partial y} = \frac{1}{t_o} \left( \frac{1}{G_t} + \frac{1}{t_i} \right) (\tau_{xy}^o + \tau_{yz}^o) \tag{13}
\]
For the one dimensional joint shown in Figure 7, all partial derivatives with respect to \( x \) would be zero. By incorporating the adhesive constitutive behavior from Eq. 1, the governing equation for this problem is derived:

\[
\frac{d^2 y_{xy}^a}{dy^2} = \lambda^2 [(1 - \frac{kR}{G_o})y_{xy}^a + \frac{R}{G_u} (1 - e^{-\lambda y_{xy}^a})]
\] (14)

where

\[
\lambda^2 = \frac{1}{t_a G_{o f_o}} + \frac{1}{G_t t_t}
\] (15)

Since this governing equation cannot be solved directly, the numerical Runge-Kutta fourth order with shooting method \([12]\) is applied to obtain a solution. The boundary conditions for this problem are defined as:

at \( y = -c \), \( \tau_{xy}^o = 0 \) and at \( y = c \), \( \tau_{xy} = \frac{N_{xy}}{t_o} \) (16) and (17)

These boundary conditions are transformed into conditions applicable to solving Eq. 14 using the one-dimensional form of Eq. 10.

\[
\frac{d y_{xy}^a}{dy} \bigg|_{y=-c} = \frac{N_{xy}}{t_o G_{o f_o}} \quad \text{and} \quad \frac{d y_{xy}^a}{dy} \bigg|_{y=c} = \frac{N_{xy}}{t_o G_{o f_o}}
\] (18) and (19)

In using the Runge-Kutta method, two initial conditions are required rather than two boundary conditions. Thus, to predict failure, the strain at either end of the joint is set as the failure strain. For a given load \( N_{xy} \), the slope of the strain calculated by Eq. 18 or 19 is used as the second initial condition, thus permitting the strain distribution within the adhesive to be numerically determined. At the other end of the joint, opposite to the side where the initial conditions were applied, the calculated slope of the strain is compared with the boundary condition (Eq. 18 or 19). If these values are not matched, the load \( N_{xy} \) is changed (this affects the slope boundary conditions) and the strain distribution is re-calculated. This process is repeated iteratively until both boundary conditions are satisfied. Figure 8 describes this iterative process. For a balanced joint (i.e., \( G_o t_o = G_t t_t \)), the strain of the adhesive would be the same at both ends of the joint.

When using the Runge-Kutta numerical integration method for solving the governing equation, one must consider the following:
The boundary condition should be a mixed type: Dirichlet boundary condition for the strain and Neuman boundary condition for the gradient of the strain.

In order to find the final failure load $N_{xy}^f$ in a numerically stable manner, the failure load must be approached from below, with guesses for $N_{xy}^f$ not exceeding the final value. This latter condition results in an unstable prediction of the strain profile, and is an indication that lower values of $N_{xy}^f$ must be chosen.

Assume Applied Load $N_{xy}^f$
- Compute $\frac{d\gamma_x^a}{dy}|_{y=-c} = \frac{N_{xy}}{t_s G t_i}$

Assume Strain Initial Condition $\gamma_x^a|_{y=-c}$

Compute Strain Profile $\gamma_x^a$ for $-c \leq y \leq c$

$\frac{d\gamma_x^a}{dy}|_{y=-c} = \frac{N_{xy}}{t_s G t_i}$

Yes

$\gamma_x^a|_{y=-c} = \gamma_x^{alt}$ and $\gamma_x^a|_{y=c} \leq \gamma_x^{alt}$

or

$\gamma_x^a|_{y=-c} \leq \gamma_x^{alt}$ and $\gamma_x^a|_{y=c} = \gamma_x^{alt}$

Yes

$N_{xy} = N_{xy}^f$

End

Figure 8 – The iterative algorithm.
Example Calculation

Failure prediction is demonstrated for a joint with carbon/epoxy cloth adherends of layup [0/45/90/-45]_12s, overlap length \(2c = 25.4\) mm and bonded by PTM&W ES6292 adhesive. Generic properties have been assumed for carbon/epoxy cloth, and values are listed in Table 2. The elastic limit load \(N_{xy}^e\) can be calculated based on the assumption of elastic-to-failure adhesive stress-strain behavior [11].

\[
N_{xy}^e = \frac{2c \tau_{ult} \tanh \frac{2c}{\lambda c}}{\lambda c}
\]  \hspace{1cm} (20)

Eq. 20 can be considered as a conservative prediction of joint failure since it does not account for any adhesive plasticity. The strain and the stress profile corresponding to the elastic limit load are shown in Figure 9. \(\tau_{ult}\) can be selected to be either the yield stress or the ultimate stress listed in Table 2.

When conducting the nonlinear failure prediction (using Eq. 14), the applied load is increased in the iterative manner previously described until the ultimate failure strain \(\gamma_{ult}\) in the adhesive is reached, thereby revealing the failure load, \(N_{xy}^f\). This condition is shown by plots of adhesive strain and stress in Figure 9. At failure load, the adhesive shear stress profile shows significant plasticity development throughout the joint. Note that these profiles based on the failure load are much higher than those corresponding to an elastic limit calculation, as shown in Figure 9. For this joint, the elastic (Eq. 20) and the failure (Eq. 14) limit loads are predicted to be \(N_{xy}^e = 150.8\) N/mm and \(N_{xy}^f = 645.3\) N/mm, respectively. Comparing these loads shows that the elastic limit is conservative by a factor of four times for this example case.

Table 2 - Example calculation joint parameters for carbon/epoxy adherends.

<table>
<thead>
<tr>
<th>Joint Parameters</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>(t_o, t_i) (mm)</td>
<td>2.03</td>
</tr>
<tr>
<td>(t_o) (mm)</td>
<td>0.33</td>
</tr>
<tr>
<td>(G_o, G_i) (GPa)</td>
<td>22.03</td>
</tr>
<tr>
<td>(G_s) (GPa)</td>
<td>0.927</td>
</tr>
<tr>
<td>(\tau_{ult}) (MPa)</td>
<td>26.3</td>
</tr>
<tr>
<td>(\gamma_{ult})</td>
<td>0.35</td>
</tr>
</tbody>
</table>
Finite Element Analysis Comparison

Finite Element Analysis (FEA) was used to evaluate the accuracy of the theoretical predictions of the shear stress distribution in a single lap joint. In order to model a state of pure in-plane applied shear using two dimensional FEA, axisymmetric elements were used to model two thin-walled cylinders with large radius ($r = 2.03$ m) bonded to each other. The cross section of this joined cylinder represents the single lap joint described in Figure 7. A rotation about the symmetry axis was applied at one boundary of the cylinder, and the other end was fixed against rotation, thereby producing a state of in-plane shear. This modeling approach allowed a two-dimensional axisymmetric model (with non-axisymmetric loading) to be used instead of a fully three-dimensional model. Two-dimensional axisymmetric quadratic 8-node elements CGAX8R in ABAQUS [13] were used in this analysis, which incorporated the nonlinear adhesive behavior shown in Figure 2.

It should be noted that for FEA software stability reasons, slight modifications to the constitutive curves shown in Figures 2 to 4 were necessary. Since a negative stress versus strain slope causes instability, values of constant stress were inputted for strain levels beyond that corresponding to the maximum stress (see Figures 2 to 4). In all FEA models, the fitting curve values were used for FEA input.

The mesh for the joint described by Table 2 is shown in Figure 10. The FEA results for this model are compared with those plotted in Figure 9 to assess the theoretical model's accuracy. Failure of the joint occurs when the strain anywhere in the adhesive reaches its ultimate value, $\gamma_{ult}$. As can be seen in Figure 10, the peak predicted plastic strain localizes at the interface corner between the adhesive and adherend, at the end of the overlap along path 3, and similarly at the opposite end, along path 1. Paths 1 to 3 are paths which adhesive stress and strain results are taken,
with paths 1 and 3 passing through the integration points in the adhesive elements located closest to the interface (0.018 mm). If the plastic strain at the integration point nearest to this interface corner reaches the failure strain of the adhesive, the analysis is terminated, and the corresponding load is interpreted as the failure load.

The strain predicted by the FEA model for a joint with $t_a = 0.33$ mm, carbon/epoxy cloth adherends of layup $[0/45/90/-45]_{2S}$ and overlap length $2c = 25.4$ mm is plotted in Figure 11 along the three paths indicated in Figure 10. Note that path 2 is through the adhesive centerline. The divergence of results plotted along these paths indicates that there exists a gradient in strain through the adhesive thickness near the ends of the overlap. The theoretical model prediction, which assumes uniform strain through the adhesive thickness, is also shown in Figure 11. For this case study, the applied load associated with failure is predicted to be 645.3 N/mm for the theoretical model and 609.4 N/mm for the FEA model.
Case Studies

Two additional bondline thickness cases of 25.4 mm lap length joints with carbon/epoxy adherends were studied and compared with FEA. The adherend properties used in the FEA are listed in Table 2, and the nonlinear adhesive properties were inputted into the FEA software as tabular data so as to represent the curves shown in Figures 2 to 4. In Figure 12, the total strain estimated by the FEA and theoretical models is plotted for a joint with $t_a = 1.07$ mm. As can be seen in the figure, the localized effect is more obvious than the $t_a = 0.33$ mm case. Since the FEA-predicted failure load is based on the maximum strain occurring in this localized zone, and the theoretical model predicts only a uniform strain profile through the bond thickness, the FEA model generally predicts a failure load that is lower than the theoretical model.

In Figure 13, the total strain for a joint with $t_a = 2.08$ mm is plotted. The localized plastic zone effect is even more dominant for this adhesive thickness than for the $t_a = 1.07$ mm adhesive thickness case and therefore the theoretical model grossly overpredicts the failure load. Figures 11 through 13 show that the intensity of the strain localization increases with $t_a$. For these cases studied, the theoretical prediction of failure load was found to be accurate within 13% for the considerably thick adhesive bondlines, up to 1.07 mm. For more conventional thickness joints, less than 0.33 mm, the theoretical prediction is within 6% accuracy. The results are summarized in Table 3. The theoretical estimations for a joint with $2c = 50.8$ mm bonded lap length are also reported in Table 3. Generally a long bonded lap joint can
carry more load than a short one. Therefore the theoretically estimated final failure load increases.

To investigate the effect of adherend stiffness, glass/epoxy cloth adherends having layup \([0/45/-45/0]\) and thickness 2.49 mm were analyzed. The shear modulus for both adherends was 10.07 GPa, and the adhesive properties for each thickness were the same as those used in the carbon/epoxy cases. The predicted strain and stress for these cases are plotted in Figures 14 to 16 for joints with 25.4 mm overlap. Since the adherends are more flexible, the plastic strain localization is more dominant than the carbon/epoxy adherends cases. Consequently the differences between the theoretical failure load estimations and the FEA results are greater than the carbon/epoxy cases for all adhesive thickness. Table 4 summarizes the glass/epoxy joint predictions for bond length of 25.4 mm and 50.8 mm.

---

Figure 12 - Shear strain and shear stress at failure load for carbon/epoxy joint with \(t_e = 1.07\) mm.
Figure 13 - Shear strain and shear stress at failure load for carbon/epoxy joint with $t_a = 2.08$ mm.

Figure 14 - Shear strain and shear stress at failure load for glass/epoxy joint with $t_a = 0.33$ mm.
Figure 15 - Shear strain and shear stress at failure load for glass/epoxy joint with $t_a = 1.07$ mm.

Figure 16 - Shear strain and shear stress at failure load for glass/epoxy joint with $t_a = 2.08$ mm.
JOINING AND REPAIR OF COMPOSITE STRUCTURES

Table 3 - Theoretical model and FEA comparison of carbon/epoxy joints.

<table>
<thead>
<tr>
<th>t_a (mm)</th>
<th>Failure Limit, N'_sy (N/mm)</th>
<th>l = 25.4 mm</th>
<th>l = 50.8 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>FEA</td>
<td>Theoretical (% Difference)</td>
<td>Theoretical</td>
</tr>
<tr>
<td>0.33</td>
<td>609.4</td>
<td>645.3 (5.9)</td>
<td>702.3</td>
</tr>
<tr>
<td>1.07</td>
<td>560.4</td>
<td>630.5 (12.5)</td>
<td>1005</td>
</tr>
<tr>
<td>2.08</td>
<td>458.8</td>
<td>556.9 (21.4)</td>
<td>928.1</td>
</tr>
</tbody>
</table>

Table 4 - Theoretical model and FEA comparison of glass/epoxy joints.

<table>
<thead>
<tr>
<th>t_a (mm)</th>
<th>Failure Limit, N'_sy (N/mm)</th>
<th>l = 25.4 mm</th>
<th>l = 50.8 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>FEA</td>
<td>Theoretical (% Difference)</td>
<td>Theoretical</td>
</tr>
<tr>
<td>0.33</td>
<td>429.1</td>
<td>537.9 (20.2)</td>
<td>540.4</td>
</tr>
<tr>
<td>1.07</td>
<td>453.2</td>
<td>631.3 (39.3)</td>
<td>778.6</td>
</tr>
<tr>
<td>2.08</td>
<td>377.9</td>
<td>560.4 (48.3)</td>
<td>730.1</td>
</tr>
</tbody>
</table>

Discussion

Predictions of joint failure have been found to be non-monotonically dependent on bondline thickness. Specifically, the intermediate thickness, t_a = 1.07 mm, was found in most cases to be the strongest, except for the short overlap (2c = 25.4 mm) carbon/epoxy joints (see Tables 3 and 4). The reason for this non-monotonic thickness-dependent trend can be explained by two competing effects: increasing the bondline reduces the adhesive strain resulting in potentially higher strength, while the ultimate failure strain decreases for increasing bondline thickness (see Figure 1). To isolate these effects, Figures 17 and 18 plot the failure load of carbon/epoxy and glass/epoxy joints as a function of bondline thickness, with the same constitutive curve (corresponding to t_a = 0.33 mm) used in each analysis. This removes any material strength effects from the calculation. In both these figures, this “ideal” case shows the strength to always increase with increasing bondline thickness. The increase is more obvious for joints having longer overlap 2c, and more compliant adherends. Also plotted in Figures 17 and 18 are the “real” results of Tables 3 and 4. These results include both aforementioned competing effects, and show that after t_a = 1.07 mm, the thickness-dependent failure strain effect dominates.

When conducting the FEA, material constitutive behavior was inputted such that no negative stress versus strain slope exists. Doing so prevents numerical instabilities from occurring during the analyses. However, such slight modification to the constitutive behavior is an inconsistency between the theoretical models and the FEA. To investigate the effects this inconsistency has on the predictions, the analytical model was re-run for the cases listed in Tables 3 and 4, using the same “leveled” constitutive curve applied in the FEA models. These leveled constitutive curve results are referred to as “Model 2” whereas the previous determined results, “Model 1,” are those values listed in Tables 3 and 4. Models 1 and 2 are compared in Tables 5 and 6. The greatest discrepancy between their predicted failure loads is 3.2%. Thus, for constitutive curves having little negative slope behavior, such as those analyzed in...
this paper, whether the negative slope in included or not does not significantly affect the results.

Figure 17 - The failure load prediction of carbon/epoxy joints using idealized \((t_a = 0.33 \text{ mm})\) and real constitutive behavior.

Figure 18 - The failure load prediction of glass/epoxy joints using idealized \((t_a = 0.33 \text{ mm})\) and real constitutive behavior.
Table 5 – Effect of constitutive curve leveling for carbon/epoxy.

<table>
<thead>
<tr>
<th>$t_a$ (mm)</th>
<th>Failure Limit, $N_{y}^f$ (N/mm)</th>
<th>$l = 25.4$ mm</th>
<th>$l = 50.8$ mm</th>
<th>Model 1</th>
<th>Model 2 (% Difference)</th>
<th>Model 1</th>
<th>Model 2 (% Difference)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.33</td>
<td>645.3</td>
<td>655.5 (1.6)</td>
<td>702.3</td>
<td>725.1 (3.2)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.07</td>
<td>630.5</td>
<td>630.5 (0)</td>
<td>1005</td>
<td>1005 (0)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2.08</td>
<td>556.9</td>
<td>565.4 (1.5)</td>
<td>928.1</td>
<td>936.9 (1.0)</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Table 6 – Effect of constitutive curve leveling for glass/epoxy.

<table>
<thead>
<tr>
<th>$t_a$ (mm)</th>
<th>Failure Limit, $N_{y}^f$ (N/mm)</th>
<th>$l = 25.4$ mm</th>
<th>$l = 50.8$ mm</th>
<th>Model 1</th>
<th>Model 2 (% Difference)</th>
<th>Model 1</th>
<th>Model 2 (% Difference)</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.33</td>
<td>537.9</td>
<td>541.1 (0.6)</td>
<td>540.4</td>
<td>543.9 (0.6)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1.07</td>
<td>631.3</td>
<td>631.3 (0)</td>
<td>778.6</td>
<td>778.6 (0)</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>2.08</td>
<td>560.4</td>
<td>577.9 (3.1)</td>
<td>730.1</td>
<td>737.3 (1.0)</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Conclusions

A theoretical model predicting failure of in-plane shear loaded bonded joints has been derived based on tracking the development of adhesive shear strain. FEA was used to evaluate the accuracy of the theoretical predictions of the shear strain and stress distribution in the adhesive. The following conclusions can be made based on the analysis results.

1. Failure load based on ultimate strain is significantly higher than elastic-to-failure based predictions. Higher load carrying structures can be designed when analysis tools such as the one presented herein can account for adhesive plasticity. While the nonlinear analysis presented herein is nonconservative, when used together with an elastic-to-failure based prediction, the upper and lower bounds of joint failure strength can be well defined.

2. The theoretical model predictions of failure load are more accurate for thinner bondlines, and for stiffer adherends. Cases analyzed show accuracy within 13% can be attained for bondlines less than 1 mm, and when adherends have shear modulus greater than 20 GPa. Limitations of this model are towards the opposite spectrum: thick bondline and compliant adherends. Specifically, the model should not be used for bondlines greater than 1 mm.

3. Although the shear stress-strain data for thinner adhesive layers generally exhibit significantly higher failure strain, joints with thinner bondlines are not in general the strongest. For example, the glass/epoxy joints with 1.07 mm bondline thickness were predicted to be stronger than the 0.33 and 2.08 mm thickness joints. This is due to competing geometric versus material influences on joint strength.

4. Plastic strain was predicted by the FEA to localize at the interface corners of the overlap ends. These locations are where failure load in joints have been experimentally observed to initiate. This localization effect is more severe for
joints having thicker bondlines. Due to the assumption of uniform strain through the adhesive thickness, the theoretical model did not capture these strain localization phenomena, and therefore the model's predictions were not conservative.

The authors acknowledge Dr. Larry Ilcewicz and Mr. Peter Shyprykevich of the Federal Aviation Administration, and the Purdue Research Foundation for the support of this research. Additional acknowledgement is due to Dr. John Tomblin and Mr. Waruna Seneviratne at Wichita State University for providing the adhesive constitutive data.

References

Box Beam Lap Shear Torsion Testing for Evaluating Structural Performance of Adhesive Bonded Joints


ABSTRACT: In this study, failure strengths of in-plane shear loaded bonded joints were compared with analytical predictions of Shear Loaded Bonded Joint (SLBJ) theory. The investigation was carried out in two phases. Phase I was conducted with a particular focus placed on the effect of bondline thickness on joint strength. These specimens were fabricated using E-glass/epoxy cloth and PTM&W ES6292 two-component paste adhesive. A box beam torsion test fixture was used to apply a shear loading. Phase II was carried out to investigate adhesive and/or adherend variability in SLBJ predictions. These specimens were fabricated using aluminum and carbon adherends with Loctite and Hysol EA9360 paste adhesives. Several joggle (production-style) joints were tested to investigate the effects of joggle adherend on strength of the adhesive joint. Furthermore, a failure analysis was conducted to study the failure mechanism of these joints. Experimental data and SLBJ predictions indicated a decrease in strength as the bondline thickness was increased. SLBJ predictions for thin bondlines were comparable with experimental data, but for thick bondlines SLBJ predictions were lower than the experimental data. Substrate failure of EA9360 specimens resulted in significantly lower experimental data than SLBJ predictions. An accumulation of large plastic strains in thin bondlines resulted in high adherend interlaminar strains and caused substrate failure. The unstable damage development of thick bondlines resulted in adhesive cracking in multiple locations with a cohesive type failure and lower failure strengths than that of the thin bondlines.

KEYWORDS: adhesive characterization, stress analysis, box beam test, lap shear

Introduction

Growing applications of adhesive bonded structures require validation of joint strength both analytically and experimentally. Stress analysis of these joints requires an experimentally validated analytical model that can predict the elastic limit and ultimate joint strength. Various certification-related issues arise in the application of adhesive joining. For small manufacturers, there is a trend toward the use of unusually large bond-layer thicknesses beyond the range for which structural performance data are readily available. There is a general lack of agreement on stress analysis methods and failure criteria for the design of adhesive joints. Limited structural data has been released which allows for some validation of the modeling effort to be substantiated, although a thorough
experimental validation has not been initiated. The box beam torsion lap shear test program was designed to support the modeling efforts of adhesive joints and to characterize adhesive joint strength in a sub-component level.

For SLBJ prediction [1], the experimentally measured constitutive behavior of adhesive by ASTM Test Method for Thick Adherend Metal Lap-Shear Joints for Determination of the Stress-Strain Behavior of Adhesives in Shear by Tension (D5656), as shown in Figure 1 for ES6292 adhesive, was modeled by a two-parameter exponential fitting curve, and the adherends were assumed to be linear elastic. Two to five specimens per bondline thickness were tested according to D5656 procedure and one of the stress-strain curves were selected to represent the corresponding bondline thickness. SLBJ predicted failure loads for joints having similar bondline thicknesses as tested D5656 coupons. Therefore, these predictions were linearly fitted and recalculated for box beam lap shear test specimens.

Box beam torsion testing was conducted in two phases. Experimental results in coupon-level tests using different ASTM standard test methods have indicated a significant decrease in apparent shear strength with increasing bondline thickness regardless of the test method [2]. Therefore, Phase I focused on bondline thickness effects on bonded joints and compared the results with SLBJ predictions. Bondlines were selected to represent General Aviation (GA) composite joint thicknesses ranging from 1.27 mm to 5.08 mm. In Phase II, different adhesive-adherend combinations were tested and compared with SLBJ predictions. Joggle joints are commonly exploited in production of airframes with adhesive joints. Therefore, several joggle joints were tested and compared with flat joints to investigate the effects of joggle adherend on the strength of the adhesive joint. Coupon level testing has demonstrated significant changes in failure modes for different bondline thicknesses and adhesive-adherend combinations [2]. Failure analysis is a crucial part in joint characterization. Failure modes such as interlaminar failure (composite) and shear buckling (aluminum) of adherends do not represent the adhesive failure, but rather joint failure.

![Figure 1 - ES6292 Adhesive constitutive behavior for different bondline thicknesses.](image-url)
Analytical Predictions

The failure of in-plane shear loaded bonded joints is predicted with a shear lag-based theoretical model. This model accounts for the development of large plastic strains in the adhesive prior to failure, by modeling experimentally obtained shear stress-strain curves using two-parameter exponential fitting curves. Failure of the adhesive is predicted by solving the governing differential equations using the Runge-Kutta method and the failure strain as measured by ASTM D5656 and simulated by the curve fit as the initial conditions. Therefore, for this failure prediction analysis to be successful, a series of ASTM D5656 tests were performed for bondline thicknesses of interest. A procedure for describing adhesive plasticity in the form of a nonlinear constitutive relationship and the calculation of joint failure is detailed in [1]. This analysis modeled the nonlinear shear stress-strain behavior for a ductile adhesive by a two-parameter exponential fitting curve [3]:

$$\tau_a = (G_a - kB_\text{I})\gamma_a + B_1(1 - e^{-k\gamma_a})$$  \hspace{1cm} (1)

In this equation, $k$ and $B_1$ are fitting parameters chosen in order to match the fitting curve to experimentally measured adhesive shear stress-strain data, and $G_a$ is the initial elastic shear modulus. The constitutive behavior of the adhesives under investigation was experimentally measured in accordance with ASTM D5656 using a modified KGR-type extensometer [4]. The parameters $k$ and $B_1$ were determined for the three adhesive systems, and are summarized in Table 1 for each bondline thickness. The ultimate shear strain, $\gamma_a^{\text{ult}}$, for ES6292 bondline thicknesses 1.829 and 4.267-mm were lower than the expected values, because these specimens were fabricated using a different adhesive batch. This reflected on the SLBJ predictions as well.

<table>
<thead>
<tr>
<th>Adhesive to</th>
<th>$t_a$ (mm)</th>
<th>$k$</th>
<th>$B_1$ (MPa)</th>
<th>$\gamma_a^{\text{ult}}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>PTM&amp;W ES6292</td>
<td>0.330</td>
<td>36.4</td>
<td>25.367</td>
<td>0.350</td>
</tr>
<tr>
<td></td>
<td>1.067</td>
<td>35.6</td>
<td>26.222</td>
<td>0.241</td>
</tr>
<tr>
<td>$G_a = 0.924$ GPa</td>
<td>1.829</td>
<td>33.4</td>
<td>29.655</td>
<td>0.073</td>
</tr>
<tr>
<td></td>
<td>2.083</td>
<td>40.0</td>
<td>25.829</td>
<td>0.130</td>
</tr>
<tr>
<td></td>
<td>3.048</td>
<td>36.8</td>
<td>28.028</td>
<td>0.070</td>
</tr>
<tr>
<td></td>
<td>4.267</td>
<td>60.6</td>
<td>14.438</td>
<td>0.031</td>
</tr>
<tr>
<td>Hysol EA9360</td>
<td>0.991</td>
<td>26.2</td>
<td>32.689</td>
<td>0.425</td>
</tr>
<tr>
<td>$G_a = 0.855$ GPa</td>
<td>2.489</td>
<td>23.5</td>
<td>39.708</td>
<td>0.155</td>
</tr>
<tr>
<td>Loctite</td>
<td>0.838</td>
<td>27.3</td>
<td>17.541</td>
<td>0.334</td>
</tr>
<tr>
<td>$G_a = 0.483$ GPa</td>
<td>1.651</td>
<td>21.5</td>
<td>22.671</td>
<td>0.305</td>
</tr>
</tbody>
</table>

The failure load, $N_{sy}$, was predicted using SLBJ for the three different adhesives for in-plane shear-loaded joints having the adherends specified in Table 2. In this model, the
adherends were assumed to be linear elastic. All joints had an overlap length of 12.7 mm and were configured as balanced joints, which means that the product of thickness and effective shear modulus were the same for both the outer and inner adherends. In Table 3, predicted failure loads $N_{sv}^f$ are listed for adhesive/adherend configurations and bondline thickness similar to those experimentally tested according to ASTM D5656. Except for the Loctite adhesive, the predicted failure load $N_{sv}^f$ was found to decrease for greater bondline thickness. The Loctite joints show a reverse trend due to the adhesive showing higher ultimate strength for a thicker bondline.

Table 2 – Adherend properties.

<table>
<thead>
<tr>
<th>Material</th>
<th>Thickness (mm)</th>
<th>Effective Shear Modulus (GPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Glass/Epoxy</td>
<td>2.489</td>
<td>5.812</td>
</tr>
<tr>
<td>Carbon/Epoxy</td>
<td>2.286</td>
<td>9.467</td>
</tr>
<tr>
<td>Aluminum (2024-T3)</td>
<td>1.270</td>
<td>27.994</td>
</tr>
</tbody>
</table>

Table 3 – SLBJ failure load prediction.

<table>
<thead>
<tr>
<th>Adhesive</th>
<th>$t_a$ (mm)</th>
<th>Adherend</th>
<th>Predicted $N_{sv}^f$ (kN/m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>PTM&amp;W ES6292 (F)</td>
<td>0.330</td>
<td>Glass/Epoxy</td>
<td>327.487</td>
</tr>
<tr>
<td></td>
<td>1.067</td>
<td>(F)</td>
<td>315.228</td>
</tr>
<tr>
<td></td>
<td>1.829</td>
<td>(F)</td>
<td>264.441</td>
</tr>
<tr>
<td></td>
<td>2.083</td>
<td>(F)</td>
<td>282.830</td>
</tr>
<tr>
<td></td>
<td>3.048</td>
<td>(F)</td>
<td>236.421</td>
</tr>
<tr>
<td></td>
<td>4.267</td>
<td>(F)</td>
<td>162.868</td>
</tr>
<tr>
<td>Hysol EA9360 (E)</td>
<td>0.991</td>
<td>Carbon/Epoxy</td>
<td>416.802</td>
</tr>
<tr>
<td></td>
<td>2.489</td>
<td>(C)</td>
<td>352.005</td>
</tr>
<tr>
<td>Loctite (L)</td>
<td>0.838</td>
<td>Aluminum</td>
<td>241.675</td>
</tr>
<tr>
<td></td>
<td>1.651</td>
<td>(A)</td>
<td>272.322</td>
</tr>
</tbody>
</table>

Experimental Method

Test Specimens

Two different types of specimens were tested in a torsion-only loading configuration; flat and joggle (production style). Specimens with composite adherends consisted of two flat laminates with a 10-ply layup [04/45/-45/04]. Loctite specimens in Phase II were fabricated using 1.270 mm thick 2024-T3 aluminum adherends. In order to obtain failure of the specimen within the capacity of the fixture, a 12.7-mm overlap length was used. Coupon level testing conducted to characterize adhesives according to ASTM D5656
used an overlap length of 9.525 mm and demonstrated satisfactory test results. Thus, a 12.7-mm overlap was considered to be sufficient for this investigation. In addition, the overall length of the specimen was 43.815 cm (Figure 2), in order for the middle region of the specimen to have a uniform shear flow, which can statically be determined.

![Figure 2 - Test specimen with two flat adherends.](image)

The second series of specimens in Phase II had joggle joints with a cross-section as shown in Figure 3. These specimens are well representative of actual production style joints. Two aluminum molds were machined to fabricate joggle joints with 1.270-mm and 2.540-mm bondline thicknesses for 10-ply E-glass adherends. The curvature of the joggle section was designed to minimize resin-rich areas due to bridging of the composite plies.

![Figure 3 - Test specimen with joggle joint.](image)

Please refer to the following nomenclature for specimen identification. The first letter indicates the loading configuration. All testing in this investigation was conducted in torsion-only (T) configuration. The second letter indicates the type of joint, i.e., flat
(F) or joggle (J). Third and fourth letters indicates the adhesive and adherend material, respectively. Please refer to the letters indicated in parenthesis next to adhesive and adherend listed in first and third columns of Table 3, respectively. The next three numbers represent the approximate bondline thickness in thousandth of an inch. The final number represents the replica number with the same parameters given by previous letters and numbers.

**Test Fixture and Instrumentation**

A torsion test fixture (Figure 4) was designed with a maximum capacity of 7 kN.m (60 in-kip). It consists of two major sections: fixed-end and pivot-end or loading-end. This unique design facilitates torsion-only loading by allowing axial float of the loading-end. A 6.35-cm needle bearing mounted in the loading-end post only allows rotation and translation in the axial direction. A twin-plate moment arm connected by a 6.35-cm shaft through the needle bearing assures that the loading plate does not swivel during the load application. The moment arm has flexibility to change to 15.24, 22.86, or 30.48 cm. All testing in this investigation was conducted with a fixed moment arm of 30.48 cm. In addition, the swivel end of the actuator has complete rotational degrees of freedom, which prevents any side loads. In order to ensure that the measured load was orthogonal to the loading plate, the load cell was mounted between the loading plate and the swivel joint of the actuator.

![Figure 4 - Box Beam Torsion Test Fixture](image-url)
The distance between the loading-end side plate (inboard) and the fixed-end block was approximately 73.66 cm for this particular test setup. However, this distance can be increased in 60.96-cm increments without additional fixturing. Slack in the bolt holes allows the parts to move in a horizontal direction. Since loading-end and fixed-end bases were separate units, aligning the fixture was a crucial part of testing. A 2.54-cm hole was drilled through each end-plug assembly and through the fixed-end block so that a 1-inch steel rod could be inserted through these holes to align the fixture. Vertical alignment was achieved using brass shims. A torque wrench was used to bolt specimens to the test fixture to minimize stress concentrations around bolt holes and to apply even pressure.

Data Reduction

The Bredt-Batho Theory [5], a theory of torsion of closed thin-walled beams, was utilized to calculate shear flow through box beam walls assuming that: (a) stresses do not vary through the thickness, and (b) direction in which the stress acts is tangent to the median line drawn through the wall thickness. It is imperative that the load case is pure torsion for application of these formulas. Shear flow throughout the cross section of the box beam, $q_{xs}$, and the average shear stress acting over the thickness of each wall, $\tau_{xs}$, are given in Eqs. 2 (Bredt-Batho Formula) and 3 (for thin-walled beams). The mean area enclosed within the boundary of the centerline of box beam wall thickness is denoted as $\bar{A}$.

$$ q_{xs} = \frac{T}{2 \cdot \bar{A}} \quad (2) $$

$$ \tau_{xs} = \frac{q_{xs}}{t} \quad (3) $$

Substituting Eq. 3 into Hooke's Law and rearranging terms, shear flow in the linear elastic range is expressed in Eq. 4.

$$ q_{xs} = G_{xs} \cdot \gamma_{xs} \cdot t \quad (4) $$

Results

Calibration of the test fixture with aluminum plates on both sides indicated constant shear flow through all four box beam walls. Shear flow on each wall was calculated using Eq. 4. In addition, these values were comparable with the experimental shear flow obtained using the Bredt-Batho Formula. Axial strain data indicated insignificant values, confirming negligible axial forces. Table 4 compares the maximum shear flow obtained for specimens in Phase I from Bredt-Batho Formula (Experimental), and SLBJ theory. SLBJ shear flow predictions were based on the thicknesses available from ASTM D5656 characteristic shear responses (Table 3). Therefore, the SLBJ predictions were linearly curve-fitted, and a representative equation was derived to obtain the analytical approximations for the corresponding bondline thicknesses presented in Table 4. ASTM
D5656 test data were not available for 5.080-mm bondlines. Therefore, no comparisons were performed beyond 4.267-mm bondlines. The load carrying capability of lap joints decreased for thicker bondlines as predicted by SLBJ theory. This observation was noted in coupon level testing as observed in [2] and [6]. Figure 5 graphically compares maximum shear flow data obtained from the above-mentioned two methods with respect to bondline thickness. The rate of joint strength drop for increasing bondline thickness for SLBJ predictions was higher than that of experimental data. This resulted in higher experimental failure strengths than SLBJ predictions for thick bondlines. Linear regression presented in Figure 5 for experimental data and SLBJ predictions indicates that the SLBJ predictions were 4.2%, 9.1%, 15.1%, and 22.4% lower than experimental data for bondline thickness of 1, 2, 3, and 4 millimeters, respectively.

Unlike on both steel channels and aluminum side plates, the in-plane shear strains recorded on the overlap region, especially for thin bondlines, indicated a significant non-linearity. Average failure strains of the outer adherend calculated using Eq. 4 for each bondline thickness were superimposed on an in-plane shear stress-strain curve obtained from a test conducted according to ASTM Test Method for Shear Properties of Composites Materials by the V-Notched Beam Test (D5379) for a 20-ply ([0\45/-45/0\45]s) laminate in Figure 6. The SLBJ predictions assumed linear-elastic behavior of the adherend. However, failure strains indicated that the adherend had exceeded the linear elastic limit of the laminate, which may have caused the non-linearity in the lap joint strain data. For a 1.270-mm bondline, the failure strains were substantial and the specimens resulted in adherend failure. Therefore, the comparison of the SLBJ predictions with experimental data for these specimens might be misleading.

<table>
<thead>
<tr>
<th>Specimen Name</th>
<th>Bondline Thickness (mm)</th>
<th>Maximum Shear Flow (kN/m)</th>
<th>Experimental</th>
<th>SLBJ</th>
</tr>
</thead>
<tbody>
<tr>
<td>TF-PF-050-1</td>
<td>1.283</td>
<td>353.632</td>
<td>299.103</td>
<td></td>
</tr>
<tr>
<td>TF-PF-050-2</td>
<td>1.252</td>
<td>300.780</td>
<td>300.327</td>
<td></td>
</tr>
<tr>
<td>TF-PF-050-3</td>
<td>1.318</td>
<td>286.704</td>
<td>297.623</td>
<td></td>
</tr>
<tr>
<td>TF-PF-050-4</td>
<td>1.179</td>
<td>305.230</td>
<td>303.407</td>
<td></td>
</tr>
<tr>
<td>TF-PF-100-1</td>
<td>2.596</td>
<td>283.693</td>
<td>244.438</td>
<td></td>
</tr>
<tr>
<td>TF-PF-100-2</td>
<td>2.352</td>
<td>303.064</td>
<td>254.570</td>
<td></td>
</tr>
<tr>
<td>TF-PF-100-3</td>
<td>2.685</td>
<td>315.114</td>
<td>240.682</td>
<td></td>
</tr>
<tr>
<td>TF-PF-100-4</td>
<td>2.680</td>
<td>285.607</td>
<td>240.894</td>
<td></td>
</tr>
<tr>
<td>TF-PF-160-1</td>
<td>4.079</td>
<td>215.839</td>
<td>182.612</td>
<td></td>
</tr>
<tr>
<td>TF-PF-160-2</td>
<td>3.955</td>
<td>228.397</td>
<td>187.797</td>
<td></td>
</tr>
<tr>
<td>TF-PF-160-3</td>
<td>4.031</td>
<td>222.371</td>
<td>184.636</td>
<td></td>
</tr>
<tr>
<td>TF-PF-160-4</td>
<td>3.947</td>
<td>216.571</td>
<td>188.074</td>
<td></td>
</tr>
<tr>
<td>TF-PF-200-1</td>
<td>5.141</td>
<td>205.028</td>
<td>188.074</td>
<td></td>
</tr>
<tr>
<td>TF-PF-200-2</td>
<td>5.108</td>
<td>248.218</td>
<td>188.074</td>
<td></td>
</tr>
<tr>
<td>TF-PF-200-3</td>
<td>5.083</td>
<td>204.576</td>
<td>188.074</td>
<td></td>
</tr>
<tr>
<td>TF-PF-200-4</td>
<td>5.062</td>
<td>206.830</td>
<td>188.074</td>
<td></td>
</tr>
</tbody>
</table>
Shear flow results for both flat and joggle joints in Phase II are depicted in Table 5. The experimental shear flow of EA9360 specimens was significantly lower than SLBJ predictions. A failure mode analysis of these joints revealed the cause for this observation. Failure modes for EA9360 specimens were adherend failure. At least one ply was attached to the adhesive layer, and there was no indication of adhesive fracture. Therefore, the failure load indicated for these joints may not reflect actual adhesive failure, but failure of the adherend, which was below the failure load of the adhesive. In standard joint design practices, whenever possible, the joint is designed to ensure that the adherends fail before the adhesive [7]. This is because the failure in the adherends is fiber controlled, while the failure in the adhesive is resin dominated and thus subject to effects of voids and other defects, thickness variations, environmental effects, processing variations, deficiencies in surface preparation, and other factors that are not always adequately controlled.

Loctite joint results indicate that the SLBJ predictions were within 8% of the experimental shear flow. However, shear flow obtained from strain gage data was significantly higher than SLBJ predictions. Shear buckling of the adherend caused strain gage data to increase significantly, in turn causing the maximum shear flow obtained from strain gage data using Eq. 4 to be higher than SLBJ predictions. These joints failed in a combination of adherend shear buckling and adhesive peel. Therefore, the failure strength comparison can be misleading, because the SLBJ theory predicted the failure of the adhesive by assuming linear elastic behavior in adherend. Aluminum adherends used in these specimens were relatively thin, and analyses show that for such cases, stresses in the adhesive will be small enough to guarantee that the adherends will reach their load capacity before failure can occur in the adhesive [7].
Figure 6 — Average failure strains of outer adherend superimposed on an ASTM D5379 in-plane shear stress-strain curve of a 20-ply laminate.

Table 5 — Shear flow comparison of specimens in Phase II.

<table>
<thead>
<tr>
<th>Specimen Name</th>
<th>Bondline Thickness (mm)</th>
<th>Maximum Shear Flow (kN/m)</th>
<th>Experimental</th>
<th>SLBJ</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Flat Joints</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>TF-LA-050-01</td>
<td>1.671</td>
<td>265.312</td>
<td>273.047</td>
<td></td>
</tr>
<tr>
<td>TF-LA-050-02</td>
<td>1.681</td>
<td>253.394</td>
<td>273.431</td>
<td></td>
</tr>
<tr>
<td>TF-EC-100-01</td>
<td>2.553</td>
<td>244.808</td>
<td>349.276</td>
<td></td>
</tr>
<tr>
<td>TF-EC-100-02</td>
<td>2.570</td>
<td>263.559</td>
<td>348.453</td>
<td></td>
</tr>
<tr>
<td>TF-EC-100-03</td>
<td>2.537</td>
<td>231.857</td>
<td>349.963</td>
<td></td>
</tr>
<tr>
<td><strong>Joggle Joints</strong></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>TJ-EC-090-01</td>
<td>2.362</td>
<td>260.125</td>
<td>357.472</td>
<td></td>
</tr>
<tr>
<td>TJ-EC-090-02</td>
<td>2.383</td>
<td>285.690</td>
<td>356.677</td>
<td></td>
</tr>
<tr>
<td>TJ-EC-130-01</td>
<td>3.452</td>
<td>249.482</td>
<td>310.398</td>
<td></td>
</tr>
<tr>
<td>TJ-EC-130-02</td>
<td>3.368</td>
<td>250.890</td>
<td>314.037</td>
<td></td>
</tr>
</tbody>
</table>
Joggle joints tested in Phase I failed in bearing at opposite-side corner bolt holes. Those results were excluded in the comparison. The results for EA9360-carbon joggle joints indicated no significant changes compared to flat joints (Table 5). Failure modes indicate adherend failure as observed on flat joints. Joggle joints indicated somewhat higher strength compared to flat joints. This observation can be explained by the extra adhesive left between the joggle and flat adherend during panel fabrication, as shown in Figure 3, which increased the bonded surface area. As observed for flat EA9360 specimens, SLBJ failure predictions for EA9360 joggle joints were substantially higher than experimental data. Adherend failure in joggle specimens may have caused premature failure of these joints.

![Diagram of failure modes and bondline thickness](image.png)

**Figure 7** - *Maximum shear flow and failure mode comparison of all adhesives.*

The primary failure mode indicated significant changes as bondline thickness increased (Figure 7). Use of Photogrammetry and Electronic Speckle Pattern Interferometry (ESPI) were explored to determine the failure initiation. The ESPI technique was found to be excessively sensitive to severe rigid body motion in torsion testing and resulted in confusion in the data. The sensitivity of the ARAMIS system was incapable of capturing any anomalies as the in-plane strain gradient increased towards the failure of the joint. Failure analysis of thin joints indicated possible failure initiation at the loading-end (substrate failure), while for thick bondlines, it was a combination of cohesive and adhesive failures in the mid section of the joint. Each specimen was carefully examined to determine primary failure modes. Based on the failure analysis, Figure 7 shows the primary failure mode of ES6292 specimens. Thin adhesive bondlines indicated higher failure loads that may have been in close proximity to the ultimate loads of the adherend, i.e., interlaminar shear strength. Substrate failure observed at the
loading-end of each 1.27-mm specimen indicated that this might be the primary mode of failure. Extensive coupon level testing conducted in reference [6] also revealed first-ply failure of the substrate due to interlaminar failure for thin bondlines. ASTM D5656 data shows that apparent shear strain at failure is significantly higher for thin bondlines than that of thick bondlines, indicating an accumulation of substantial plastic strain before failure of thin bondlines [8]. For box beam torsion testing, the high angle of twist data gathered for thin bondlines also concurred with this assessment. As a result, the adherends experienced substantial strains, which consequently resulted in interlaminar failure. As the bondline increased, the adhesive yielding occurred at low stress levels compared to thin bondlines and there was virtually no plastic strain accumulation before failure, resulting in an unstable damage development process. Therefore, thick bondlines resulted in adhesive cracking in multiple locations with a cohesive type failure and lower failure strengths than that of the thin bondlines.

Both flat and joggle EA9360 joints indicated first-ply failure and no damages to the adhesive layer. However, these thick bondlines resulted in adherend failure, contradicting the conclusions based on observations in the failure analysis of Phase I results. Though the stiffness of EA9360 is comparable with ES6292, the plastic strain accumulation of EA9360 thick bondlines was more stable than that of ES6292. In addition, the failure strain of EA9360 was higher than that of ES6292. Further, the interlaminar shear strength of carbon cloth was lower than that of 7781 E-glass. Combination of these adhesive-adherend material properties caused the adherend failure of EA9360 joints. This mode is considered a joint failure rather than an adhesive failure. Loctite specimens indicated significant shear buckling that resulted in adherend yielding and adhesive peeling at specimen edges. Typically, this leads to a catastrophic failure of the joint, as seen in both ES6292 and EA9360 adhesives. However, the ductility and large plastic strain accumulations at lower stress levels of the Loctite adhesive tolerated large deformation due to shear buckling of the adherend.

Conclusion

Shear loaded lap joints decreased in strength as bondline thickness increased. This observation is analogous to results obtained for specimen level testing conducted on adhesive single lap shear testing. Analytical predictions using Shear Loaded Bonded Joint (SLBJ) theory also represented this behavior, because this theory utilized ASTM D5656 single lap coupon test data to model the constitutive behavior of adhesive by means of a two-parameter exponential curve. Phase I test results also indicated changes in failure modes as bondline thickness increased; thinner bondlines indicated substrate failure, and thicker bondlines indicated primarily cohesive failure. An accumulation of large plastic strains in thin bondlines resulted in high adherend interlaminar strains and caused substrate failure. The unstable damage development of thick bondlines resulted in adhesive cracking in multiple locations with a cohesive type failure and lower failure strengths than that of the thin bondlines. SLBJ predictions for thin bondlines were comparable with experimental data. However, for thick bondlines, they were lower than the experimental data.

Phase II testing indicated substrate failure of EA9360 specimens regardless of the thickness or joint configuration, i.e., flat or joggle. Low interlaminar shear strength of
the adherend and plastic strain characteristics of the adhesive, compared to material used in Phase I, caused adherend failure of EA9360 joints rather than adhesive failure. These results, however, indicated a decrease in strength as bondline thickness increased, mainly because of the low yield strength of thick adhesives. In addition, joggle joint strength was somewhat higher than that of flat joints, due to the additional surface area for adhesive bonding. Loctite specimens indicated significant shear buckling and peel failure. The ductility and high plastic strain accumulation at lower stress levels of the Loctite adhesive permitted peel failure but continued to carry more load. However, both ES6292 and EA9360 specimens failed catastrophically with a loud noise. SLBJ predictions for EA9360 joints were significantly higher than the experimental data due to the interlaminar failure, which is lower than the failure strength of the adhesive.

Acknowledgments

This research was conducted as a part of a Federal Aviation Administration funded project. The authors would like to acknowledge the guidance and support of Mr. Peter Shyprykevich and Dr. Larry Ilcewicz of the Federal Aviation Administration, and Dr. Keith Kedward at the University of California, Santa Barbara. The authors thank Cessna Aircraft Company of Wichita, Kansas, and Cirrus Design Corporation of Duluth, Minnesota, for supplying adhesive.

References

Performance of a Composite Double Strap Joint with Attachments


ABSTRACT: Thin composite laminate attachments were designed and added onto the conventional double-strap joints so that additional load transfer paths were created and localized interfacial stress concentrations near the joint edge were reduced. Experiments were performed and the result revealed that the new double-strap joint with attachments had significantly greater strengths than the conventional double-strap joint. Joints with attachments having different angular configurations and lay-ups were tested and different failure modes were observed. Interfacial stresses along the bondlines were analyzed in order to understand the increase in joint strength and the change of failure modes in joints with attachments.

KEYWORDS: bonded joint, interfacial stress, attachment, composite laminate

Introduction

Adhesive bonding is an attractive jointing method for composite structures because of the absence of holes as required in the mechanical joints [1,2]. In bonded joints, the adhesive bondline is usually the weakest link. Moreover, the load transfer from one adherend to the other often takes place in a localized region. Thus, the load transfer efficiency is relatively low. Further, in the case of lap joints, highly concentrated interfacial stresses are present leading to premature failures and low joint strengths. There have been a number of traditional methods proposed to minimize the stress concentration of the interfacial stresses with certain degree of success. A common method is to taper the adherend near the joint end in order to reduce the high peel stress. Changing the shape of the adherend and attachments or adding fillets at the ends will also help reduce the stress concentration but the effect is somewhat modest [4-7]. Some researchers proposed to use varying toughness of adhesive along the bondline to increase the joint strength [8].

1 Graduate Student, School of Aeronautics and Astronautics, Purdue University, West Lafayette, IN 47906.
2 Neil Armstrong Distinguished Professor of Aeronautical and Astronautical Engineering, School of Aeronautics and Astronautics, Purdue University, West Lafayette, IN 47906.
Recently, Zeng and Sun [3] designed a wavy single lap joint which exhibits a drastically different interfacial stress distribution: the interfacial normal stress becomes compressive and the shear stress distribution becomes more uniform. They found that the wavy joint could substantially increase the joint strength over the conventional straight lap joint design. However, it is not easy to manufacture such wavy joints. Moreover, the wavy geometry of the adherend may potentially weaken its strength.

The purpose of the present study is to explore a new approach in improving adhesively bonded lap joints. Instead of altering the geometry of the adherend, additional load paths at the joint using a step attachment are created with the result of reducing the interfacial stresses at the joint. A double-strap joint, which is relatively simple to manufacture, is used to demonstrate the concept. Experiments on the new joint made of a fiber composite were conducted to validate the concept.

**Conventional and New Double-strap Joint Designs**

FIG. 1 shows the sketches of a conventional double-strap joint and the new design. The conventional double strap joint consists of a pair of straight attachments (straps) for load transfer. It is easy to see that the new design is obtained by adding four step attachments on top of the original joint. The two step attachments on each side of the joint may be made as a continuous piece. In this study, they were made as two separate pieces. The angle \( \alpha \) of the step attachment plays a crucial role in determining the effectiveness of the step attachment. Two different angles are studied in this paper to investigate its effect on the joint strength. The inner and outer radii of curvature of the bend in the step attachment are denoted by \( R_1 \) and \( R_2 \), respectively.

The geometrical parameters of the two joints are given in TABLE 1. The width of the two double-strap specimen is \( d = 25.4 \) mm. The material of the adherend and the attachment is S2/8552 glass/epoxy composite of which the material constants are listed in
TABLE 2. The lay-up of the adherend is [0/90/0/90]_{4s}, the straight attachment is [0/90/0/90]_{2s}. Two different lay-up of the step attachments ([0/90/0]_{4} and [0/90/0]_{2}) are involved. The 0-degree direction here is parallel to the longitudinal direction, which is also the loading direction of the joint.

<table>
<thead>
<tr>
<th>TABLE 1—Joint geometrical parameters.</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Length of adherend (L)</strong>: 139.70 mm (5.5 in)</td>
</tr>
<tr>
<td><strong>Thickness of adherend (t_{i})</strong>: 2.67 mm (0.105 in)</td>
</tr>
<tr>
<td><strong>Length of straight attachment (l)</strong>: 50.80 mm (2.0 in)</td>
</tr>
<tr>
<td><strong>Thickness of straight attachment (t_{s})</strong>: 1.34 mm (0.053 in)</td>
</tr>
<tr>
<td><strong>Length of the straight part of the step attachment (l)</strong>: 25.4 mm (1.0 in)</td>
</tr>
<tr>
<td><strong>Thickness of step attachment t_{1} ([0/90/0]_{4})</strong>: 1.016 mm (0.04 in)</td>
</tr>
<tr>
<td><strong>Thickness of step attachment t_{2} ([0/90/0]_{2})</strong>: 0.508 mm (0.02 in)</td>
</tr>
<tr>
<td><strong>Bend radius (R_{1})</strong>: 1.34 mm (0.053 in)</td>
</tr>
<tr>
<td><strong>Bend radius (R_{2})</strong>: 2.356 mm (0.093 in)</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>TABLE 2—Elastic moduli of S2/8552.</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>E_{1} (GPa (msi))</strong>: 47.7 (6.92)</td>
</tr>
<tr>
<td><strong>E_{2} (GPa (msi))</strong>: 12.3 (1.78)</td>
</tr>
<tr>
<td><strong>E_{3} (GPa (msi))</strong>: 12.3 (1.78)</td>
</tr>
<tr>
<td><strong>G_{12} (GPa (msi))</strong>: 4.8 (0.70)</td>
</tr>
<tr>
<td><strong>G_{13} (GPa (msi))</strong>: 4.8 (0.70)</td>
</tr>
<tr>
<td><strong>G_{23} (GPa (msi))</strong>: 4.5 (0.70)</td>
</tr>
<tr>
<td><strong>\nu_{12}</strong>: 0.28</td>
</tr>
<tr>
<td><strong>\nu_{13}</strong>: 0.28</td>
</tr>
<tr>
<td><strong>\nu_{23}</strong>: 0.4</td>
</tr>
</tbody>
</table>

Specimen Fabrication

For the conventional double-strap joint, 30 cm x 30 cm flat composite panels were fabricated according to the selected lay-up sequences and cured following the standard curing cycle in an autoclave. After curing, the panels for adherends and for straight attachments were cut using a water jet into the desired dimensions. Film adhesive FM73M was used for bonding. The bonding areas were cleaned repeatedly by non-linting paper wiper dampened with acetone. After cleaning, the adherends and attachments were dried in an oven for 30 minutes. Subsequently, one layer of FM73M film adhesive was applied between the bonding surfaces of the adherend and attachment. The assembled
specimen was then cured in an autoclave. Finally, the cured part was again cut using a water jet into the desired size for test.

For the new double-strap joint with attachments, the step attachments were fabricated separately. The composite prepreg tape was laid on an aluminum mold with the desired shape and cured with the usual curing procedure. The step attachments were bonded to the conventional double strap joint using FM73M film adhesive at the temperature of 250°F. The top and side views of the final specimens are shown in FIG. 2 and FIG. 3, respectively.

![FIG. 2—Top views of the conventional (top) and the new (bottom) double strap joints.](image)

![FIG. 3—Side views of the conventional (top) and the new (bottom) double strap joints.](image)

**Experimental Results**

All specimens were tested on a 98 KN servo-hydraulic testing machine (22 Kips MTS testing machine) at room temperature. The crosshead displacement rate was 0.5 mm/min. For each joint design, five specimens were tested. The results for joint strength are listed in TABLE 3. The average peak load of the conventional joint is 37.42 KN, while the peak loads for joints with $\alpha = 30^\circ$ is 45.23 KN. For $\alpha = 15^\circ$ attachments, the averaged peak loads are 43.89 KN and 51.60 KN for [0/90/0]$_4$ and [0/90/0]$_2$ laminate attachments, respectively. There are some differences in strength among the three joints with step attachments, while they are all stronger than the conventional double-strap joint.
TABLE 3—*Test results of joint strength*

<table>
<thead>
<tr>
<th></th>
<th>Conventional Joints, KN</th>
<th>With 30° attachments, KN</th>
<th>With 15° attachments ([0/90/0]_4), KN</th>
<th>With 15° attachments ([0/90/0]_2), KN</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>37.75</td>
<td>46.01</td>
<td>42.17</td>
<td>49.21</td>
</tr>
<tr>
<td>2</td>
<td>37.41</td>
<td>47.79</td>
<td>43.94</td>
<td>53.47</td>
</tr>
<tr>
<td>3</td>
<td>36.87</td>
<td>39.53</td>
<td>43.26</td>
<td>52.69</td>
</tr>
<tr>
<td>4</td>
<td>37.17</td>
<td>46.75</td>
<td>42.17</td>
<td>51.56</td>
</tr>
<tr>
<td>5</td>
<td>37.91</td>
<td>46.06</td>
<td>NA</td>
<td>48.95</td>
</tr>
<tr>
<td>average</td>
<td>37.42</td>
<td>45.23</td>
<td>42.89</td>
<td>51.20</td>
</tr>
<tr>
<td>Increment</td>
<td>20.87%</td>
<td>14.62%</td>
<td>36.8%</td>
<td></td>
</tr>
</tbody>
</table>

**FIG. 4**—*Typical load displacement curves of joints.*

FIG. 4 presents typical load-displacement curves for the conventional and the new double-strap joints. It is seen that the addition of step attachments enhances both stiffness and strength of the new joint design.

Test results revealed that the failure modes for these four types of joints were quite different. The failure of the conventional joint took the form of adhesive failure, i.e., failure occurred in the adhesive along the interface between the adherend and the straight attachment. For the joint with 30° angle step attachments, failure occurred in the form of delamination between the 0° and 90° layers in one of the four step attachments as shown in FIG. 5. The delamination took place near the bend. As the load increased, more delamination would spread to other part close to the bend leading to the failure of the whole joint. The delamination caused a load drop in the load-displacement curve as can be seen from FIG. 4.

For the joint with 15° angle step attachments, the failure modes were different for different lay-ups. For the joint with [0/90/0]_4 attachments, the failure mode resembles that of the conventional joint, i.e., failure initiates as cohesive failure between the step attachment and the adherend near the leading edge of the attachment. The specimen failed suddenly like the conventional double-strap joint did. FIG. 6 shows the fracture
surfaces of the specimen. For $[0/90/0]_2$ attachments, the failure mode appeared to be the same as the joint with $30^\circ$ attachments. The failure occurred in the attachment near the bend.

FIG. 5—Typical failure mode in the joint with $30^\circ$ angle attachments (delamination in the step attachment).

FIG. 6—Top view of the typical failure mode in the joint with $15^\circ$ angle attachments (adhesive failure).

Numerical Stress Analysis

Two-dimensional plane strain finite elements analyses were performed using commercial code ABAQUS 6.3. In view of symmetry, only a quadrant of the joint was employed in the analysis. Thus, the $x$-symmetric boundary conditions were applied along the center line of the adherends, while the $y$-symmetric boundary conditions were applied along the vertical line through the mid span of the straight attachments. Since qualitative interfacial stress distributions were of interest, the adhesive along the bondline was not modeled. A uniform stress of 1 KPa was applied at the sliding end of the joint. The composite laminates of the adherends and attachments were modeled as homogeneous solids with effective elastic calculated based on the laminate lay-ups. The formulas for 3D effective moduli for composite laminates were given by Sun and Li [9]. We obtained
the effective elastic moduli for \([0/90/0/90]_{ns}\) and \([0/90/0]_n\) with the results listed in TABLE 4 and TABLE 5, respectively.

<table>
<thead>
<tr>
<th>TABLE 4—Effective moduli for ([0/90/0/90]_{ns}) laminate.</th>
</tr>
</thead>
<tbody>
<tr>
<td>(E_1) GPa (msi) (E_2) GPa (msi) (E_3) GPa (msi)</td>
</tr>
<tr>
<td>30.2 (4.38) (13.32) (1.93) (13.32) (1.93)</td>
</tr>
<tr>
<td>(G_{12}) GPa (msi) (G_{13}) GPa (msi) (G_{23}) GPa (msi)</td>
</tr>
<tr>
<td>4.830 (0.70) (4.648) (0.67) (4.648) (0.67)</td>
</tr>
<tr>
<td>(\nu_{12}) (\nu_{13}) (\nu_{23})</td>
</tr>
<tr>
<td>0.4830 0.4830 0.4648</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>TABLE 5—Effective moduli for ([0/90/0]_n) laminate.</th>
</tr>
</thead>
<tbody>
<tr>
<td>(E_1) GPa (msi) (E_2) GPa (msi) (E_3) GPa (msi)</td>
</tr>
<tr>
<td>36.13 (5.24) (24.24) (3.52) (13.23) (1.93)</td>
</tr>
<tr>
<td>(G_{12}) GPa (msi) (G_{13}) GPa (msi) (G_{23}) GPa (msi)</td>
</tr>
<tr>
<td>4.830 (0.70) (4.707) (0.68) (4.591) (0.67)</td>
</tr>
<tr>
<td>(\nu_{12}) (\nu_{13}) (\nu_{23})</td>
</tr>
<tr>
<td>0.1426 0.3496 0.3814</td>
</tr>
</tbody>
</table>

FIG. 7—Stress distributions of the conventional double-strap joint along the interface \(B\) at a tensile stress of 1 KPa.

Plane strain 8-noded elements were used for both adherends and attachments. Finer meshes were used near the interface especially near the joint ends where stress singularities are present. In order to compare numerically the magnitudes of stress
distributions near the singularity sites, identical element sizes were used in the analyses of all joints.

FIG. 7 shows the interfacial stresses in the conventional double-strap joint along interface B (see FIG. 1). It is clear that, at the left and right ends of the interface, the normal (peel) and shear stresses rise to very high values and are highly localized.

In FIG. 8, stress distributions in the joint with 30° attachments along interfaces A and B (see FIG. 1) are shown. Note that the interfacial distance was normalized with the length of the respective interface. Stress concentrations are still noted. To compare the peeling actions resulting from the peel stresses, the peeling forces (the area under the tensile portion of the peel stress) were calculated from these curves. The results are presented in TABLE 6. Because of the use of normalized distance, the "peeling force" has the dimension of stress (N/m²). The actual peeling force can be obtained from the respective numbers by multiplying the length of the interface. These peeling forces are used to determine the failure location at the interface.

![Stress distributions along interfaces for joint with 30° attachments subjected to a uniform tensile stress of 1 KPa, (a) Interface A, (b) Interface B.](image)

**TABLE 6—Comparison of interfacial peeling forces (N/m²).**

<table>
<thead>
<tr>
<th></th>
<th>Interface A</th>
<th>Interface B</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Left end</td>
<td>Right end</td>
</tr>
<tr>
<td>Conventional joint</td>
<td>NA</td>
<td>NA</td>
</tr>
<tr>
<td>New joint (30°, [0/90/0]₄)</td>
<td>2.22</td>
<td>3.36</td>
</tr>
</tbody>
</table>

From the result in TABLE 6, it is seen that, for the conventional joint, the maximum peeling force is at the left (leading) end of interface B; for the joint with 30° attachments, the maximum peeling force occurs at right end of interface A (i.e., near the bend). These are exactly the failure initiation locations observed in the experiment.

Stress Distributions for joints with 15° attachments along interface A subjected to a tensile load of 1 KPa are shown in FIG. 9. The corresponding peeling forces at the end zones of the interfaces are listed in TABLE 7. Several conclusions can be readily
obtained from the result in TABLE 7. For the same bend angle, the higher stiffness of the attachment leads to a higher peeling force at the leading edge (left end) of the attachment interface. On the other hand, if two attachments have the same in-plane stiffness, then a smaller bend angle would yield a smaller peeling force at the right end of the interface near the bend, while the peeling force at the leading edge remains unchanged.

Experimental observations confirmed the predicted failure locations except for the joint with 15° [0/90/0]₄ attachments for which failure initiated from the leading edge (left end) rather than the right end as predicted based on the peeling force. This discrepancy could be due to possible inaccurate descriptions of the geometry of the attachment in the bend region. A deviation from the actual geometry could affect the calculated interfacial stress distribution and, thus, the peeling force.

FIG. 9—Stress distributions along interface A in joints with 15° attachments subjected to a uniform tensile stress of 1 KPa. (a) [0/90/0]₄, (b) [0/90/0]₂.

TABLE 7—Comparison Peeling forces at the interfaces of joints with different attachments.

<table>
<thead>
<tr>
<th></th>
<th>30°, [0/90/0]₄</th>
<th>15°, [0/90/0]₄</th>
<th>15°, [0/90/0]₂</th>
</tr>
</thead>
<tbody>
<tr>
<td>Left end of the</td>
<td>2.22</td>
<td>2.17</td>
<td>1.21</td>
</tr>
<tr>
<td>interface</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Right end of the</td>
<td>3.36</td>
<td>2.53</td>
<td>1.94</td>
</tr>
<tr>
<td>interface</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

It has been stated earlier that the addition of the step attachments is to create an additional load path for transferring the load from one adherend to the other. In so doing, the level of interfacial stresses is lowered as evidenced by the numerical stress analysis results. FIG. 10-13 show the percentiles of loads carried by the adherend and the attachments, respectively, at different locations in the four different joints. It is noted that the amount of load carried by each component (adherend or attachment) is proportional to its stiffness and that for joints with step attachments, the load in the step attachment is transferred back to the strap at location (d) because of the discontinuity between the two attachments.
FIG. 10—Load distribution in the conventional joint.

FIG. 11—Load distribution in the joint with $30^\circ [0/90/0]_4$ attachments.

FIG. 12—Load distribution in the joint with $15^\circ [0/90/0]_4$ attachments.
Summary

Adding thin step attachments to the conventional double-strap joints can substantially increase their strengths. It was found that the bend angle of the step attachment plays a very important role in enhancing the joint strength. In general, a larger bend angle would give rise to higher peel stresses near the bend of the attachment and, as a result, would lower the joint strength. The selection of different bend angles may also alter the location of failure initiation of the joint. It was also found that the interfacial stresses, and, thus, the joint strength, can be changed by changing the stiffness of the attachment. Optimization of the joint strength may be performed on these parameters.

Acknowledgement Support for this research provided by AFOSR Grant No. F49620-02-0018 is gratefully acknowledged.

Reference


Evaluation of a Carbon Thermoplastic to Titanium Bonded Joint


ABSTRACT: Carbon thermoplastic (AS4/APC-2) to Titanium (6Al-4V) bonded and bonded/bolted joint designs were developed for use in the marine, Composite Storage Module (CSM) for the Office of Naval Research. Finite element modeling of the design was done using PATRAN and subsequent analyses were performed using the ABAQUS finite element code. In order to validate the design and to provide a measure of confidence with the fabrication and assembly of the joint, three, full-scale test specimens were designed, fabricated and tested in a representative tensile load environment. The three joint specimens were fabricated by Production Products and tested at the Naval Surface Warfare Center, Carderock Division. The carbon specimens were 1.04 m (40.9 in.) in length by 0.254 m (10 in.) in width by 28.5 mm (1.12 in.) in thickness and captured in the 70 mm (2.75 in.) thick titanium. Two of the specimens were bonded joints and the third was a bonded/bolted joint. The specimens were subjected to three cycles of tensile loading on a 2447 kN (550 kip) MTS test machine with the last loading cycle to failure. One bonded joint specimen failed in the adhesive bondline at 378 kN (85 kips) and the second bonded joint failed at 600 kN (135 kips). The bonded/bolted joint specimen showed adhesive failure at 489 kN (110 kips) and complete failure in the bolts at 556 kN (125 kips). This paper discusses the theoretical predictions, experimental results and correlation.

KEYWORDS: bonded joint, bonded/bolted joint, carbon thermoplastic, finite element analysis

Introduction

The Composite Storage Module (CSM) [1] incorporates a carbon thermoplastic (AS4/APC-2) to titanium (6Al-4V) joint design for access (Figure 1). In order to validate the design and to provide a measure of confidence with the fabrication and assembly of the joint three full-scale joint test specimens were designed, fabricated and tested in a representative tensile load environment. Two of the specimens were bonded joints and the third was a bonded/bolted joint. Since testing of the actual 1.2 m (48 in.) diameter joint section was not practical, a full scale joint specimen was designed which incorporated the maximum stress and the equivalent strain distribution of the CSM joint. The carbon specimens were 1.04 m (40.9 in.) in length by 0.254 m (10 in.) in width by 29.7 mm (1.17 in.) in thickness and captured in the 70 mm (2.75 in) thick titanium. The three joint specimens were fabricated by Production Products. The specimens were subjected to three cycles of tensile loading with the last loading cycle to failure in the 2447 kN (550 kip) MTS test machine at Naval Surface Warfare Center.

1Program Manager, Senior Engineer, and Principal Engineer, General Dynamics Electric Boat Corporation, 75 Eastern Point Road, Groton, CT 06365.
2Program Manager, Production Products Manufacturing and Sales, 1285 Dunn Road, St. Louis, MO 63138.
Test Specimen Design and Analysis

The details of the actual CSM joint and finite element analysis are discussed in Reference 1. To confirm the design of the test specimen joint, a baseline comparison of overall joint behavior was done with the existing finite element model of the CSM joint [1] and the test specimen joint finite element model. After comparing the two baseline analyses, it was determined that the test specimen finite element model was acting similarly to the actual CSM joint finite element model under the same loading condition. It is therefore concluded that the test specimen finite element model is representative of the actual CSM joint. The test specimen finite element model was then analyzed for three cases as follows:

Case 1. Test specimen with bolt pre-load to 2/3 yield strength (689 MPa) and adhesive
Case 2. Test specimen with adhesive, no bolt
Case 3. Test specimen with a pin and adhesive (Not tested, only analyzed)

Figure 2 shows the finite element model of the test specimen. A tensile pressure loading was applied to the specimen, while constraining the model on the back edge of the titanium. Figure 3 shows the dimensions of the test specimen finite element model. The case 1 analysis was done in two steps; step 1 applied the pre-load to the bolt, and step 2 applied the tensile pressure loading. For cases 2 and 3, the analysis was done in only one step with the application of the pressure loading. No pre-loading exists for these cases. For cases 1 and 3, contact was modeled at the bolt/pin interfaces with the specimen (i.e. bolt contact with AS4/APC-2, pad-up, adhesives, and titanium). For cases 1 and 3, the same finite element model (Figure 1) was used to do the analyses. In case 3, the pre-load on the bolt is removed to represent a pin in the specimen. In case 2, the bolt elements were removed from the model in Figure 2 and the new model is shown in Figure 4.
All finite element modeling was done using PATRAN [2] and subsequent analyses were performed using the ABAQUS/Standard finite element code [3]. The materials used in the model are labeled in Figure 2 and Figure 4, with the details of the
material properties shown in Table 1. For the model with no bolt, all materials are the same as in the bolt model except for the elimination of the steel for the bolt. The carbon lay-up consisted of 22% 0° plies, 22% ± 45° plies, 56% 90° plies, where 0° is along the longitudinal axis (Figures 2-4). Both the FM73 Film Adhesive and EA9394 Paste adhesive were modeled as elastic/perfectly plastic. These analyses used 3D solid elements (ABAQUS C3D8 [3]) in the mathematical idealization.

Table 1 - Test specimen material properties.

<table>
<thead>
<tr>
<th>Composite</th>
<th>Material</th>
<th>$E_1$ (Gpa)</th>
<th>$E_2$ (Gpa)</th>
<th>$E_3$ (Gpa)</th>
<th>$v_{12}$</th>
<th>$v_{13}$</th>
<th>$v_{23}$</th>
<th>$G_{12}$ (Gpa)</th>
<th>$G_{13}$ (Gpa)</th>
<th>$G_{23}$ (Gpa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>AS4/APC-2</td>
<td>Fiberglass</td>
<td>10.82</td>
<td>19.64</td>
<td>19.64</td>
<td>0.17</td>
<td>0.17</td>
<td>0.28</td>
<td>3.31</td>
<td>3.31</td>
<td>7.72</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Isotropic (Elastic)</th>
<th>Material</th>
<th>$E$ (Gpa)</th>
<th>$v$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Titanium</td>
<td>113.69</td>
<td>0.33</td>
<td></td>
</tr>
<tr>
<td>Steel</td>
<td>206.70</td>
<td>0.30</td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Isotropic (Elastic)</th>
<th>Material</th>
<th>$E$ (Gpa)</th>
<th>$v$</th>
<th>$\sigma_y$ (Mpa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>FM73</td>
<td>0.99</td>
<td>0.36</td>
<td>53.74</td>
<td></td>
</tr>
<tr>
<td>EA9394</td>
<td>4.24</td>
<td>0.45</td>
<td>24.56</td>
<td></td>
</tr>
</tbody>
</table>

Table 2 compares the maximum and nominal axial strain for the three cases analyzed under the above tensile loading for information only. For the AS4/APC-2 the nominal strain is reported in the middle of the specimen along its length.

Table 2 - Maximum/nominal strain comparison.

<table>
<thead>
<tr>
<th>Material</th>
<th>Maximum Strain (mm/mm)</th>
<th>Nominal Strain (mm/mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Case 1</td>
<td>Case 2</td>
</tr>
<tr>
<td>AS4/APC-2</td>
<td>0.003267</td>
<td>0.002901</td>
</tr>
<tr>
<td>FM73</td>
<td>0.009391**</td>
<td>0.002332</td>
</tr>
<tr>
<td>EA9394</td>
<td>0.015055**</td>
<td>0.005110</td>
</tr>
<tr>
<td>Fiberglass</td>
<td>0.004724**</td>
<td>0.001155</td>
</tr>
</tbody>
</table>

* In bolt hole through thickness
** In bolt hole (Due to pre-load)

Another concern for the test specimen was whether the FM73 film adhesive used between the AS4/APC-2 and the titanium, and the EA9394 paste adhesive used between the fiberglass and titanium would yield under the tensile loading described above. Of the three cases subject to this tensile loading, case 2 (with no bolt) can be considered the worst on the adhesive. This is based on the fact that for case 1 (with bolt), and case 3 (with pin), the joint will get additional strength from the bolt/pin and local effects of the bolt/pin will not be considered in determining worst case for the adhesives. For case 2, the results of the finite element analysis shows that the adhesive materials do not yield. The maximum Von Mises stress in the FM73 film adhesive is 43.5 MPa (6310 psi) with a
yield stress of 53.7 MPa (7800 psi). The EA9394 paste adhesive maximum Von Mises stress is 23.2 MPa (3370 psi) with a yield stress of 24.5 MPa (3560 psi).

Fabrication and Assembly

Production Products, with support from The Boeing Company, manufactured the three thermoplastic joint specimens and bonded them to the titanium sections that were provided by Electric Boat. The lay-up for the panels was based on material thickness of 0.13 mm (0.0052 in.) for the AS4/APC-2 prepreg supplied by Hexcel Fiberite and required 225 plies for the 29.7 mm (1.17 in.) nominal thickness. The plies were collated and autoclaved consolidated. (The CSM cylinder will be fiber placed, in-situ consolidated [4]-[6]). After autoclave consolidation, the panels were visibly well consolidated. Visual inspection of the panels showed average thickness of 28.5 mm (1.12 in.) in lieu of the 29.7 mm (1.17 in.) required. Once the dimensions of the thermoplastic panels and titanium joint pieces were measured, the exact dimensions of the glass/epoxy pad-up was determined. Six layers of FM73 film adhesive were required to achieve the 0.76 mm (0.030 in.) bondline between the pad-up and the thermoplastic panel. The variation in the thermoplastic panel thickness was compensated for in the EA9394 paste adhesive bondline which was nominally 1.59 mm (0.0625 in.) thick. The glass/epoxy padup was machined from 17 plies of 24 ounce per square yard of woven roving and Shell Epon 862. The glass padup to AS4/PEEK was bonded with film adhesive following surface preparation and bagging. It was cured at 121°C (250°F) for one hour. Following surface preparation of specimen and titanium contact surfaces, the paste adhesive was applied and held in a fixture at 60°C (140°F) for 24 hours until cure. Concurrent AS4/PEEK panels were also fabricated to verify the mechanical properties used in the design [7].

Failure Predictions

For this analysis, a more detailed 2D solid model of the joint with no bolt, using the same materials as the 3D analyses, was created in lieu of the existing 3D solid models used in the sections above. The 3D solid models were not used for this failure analysis based on large model size and increased analysis time. The model was corrected for the as-built thickness variation. 2D solid models of the bolt/pin configurations were not analyzed based on overall joint failure was more likely in the adhesive only configuration. It was determined from the 3D analyses that the behavior of the specimen was basically symmetrical through the width and thickness of the model, and therefore a 2D solid model would adequately represent the specimen. The loading was applied to this model in 10 increments loading up to twice that of what was used in the above 3D solid models. With the load stepped into 10 increments, a load history could be created of the failure characteristics of the joint specimen. Failure of the specimen would be determined by assessing peak stresses in the EA9394 paste adhesive. This was due to the fact that the yield stress for this material is on the order of ½ that of the FM73 film adhesive. Figure 5 shows a load history of the averaged peak stress in the paste adhesive tracking where the adhesive begins to yield and become plastic. Peak stresses were
extracted from the analysis for each loading step at a node located in the beginning of the bondline. From the figure it can be seen that the paste adhesive has a peak stress at approximately 445 kN (100 kips) of loading.

![Figure 5 - EA9394 paste adhesive stress vs. load.](image)

**Test Results**

The specimens were tested in the 2446 kN (550 000 lb) MTS test machine at NSWC/CD as shown in Figure 6. Specimen #1 and #2 were bonded joints, while Specimen #3 was a bonded/bolted joint. Each specimen had 16 strain gages and one acoustic emission transducer attached as depicted in Figure 7. The design failure load of 366 kN (82 300 lb) was obtained from preliminary finite element analyses of the joint. Each specimen was subjected to three cycles of loading. The first cycle was from 0 to 146 kN (32900 lb) to 0, the second cycle was from 0 to 366 kN (82 300 lb) to 0, and the third cycle was to failure. Figure 6 also shows the loading spectrum for the test.

Strain gage predictions were developed for all 16 gages prior to test. Typical axial strain gage comparisons between predicted and tests are shown in Figure 8 for gages 1 and 10 on the AS4/APC-2 specimen away from the joint (membrane) and Figure 9 for strain gages 3, 4, 5, 12 in the fiberglass pad-up region. Some bending of the joint is observed in Figure 8 for Test 2 with no bolt. More significant bending of the specimen is observed in Figure 9 for the fiberglass pad-up. This is probably due to the asymmetric condition of the as-fabricated specimen dimensions and variations in the paste adhesive bondline and thermoplastic substrate. The figures show both loading and unloading for the test data.
Figure 6 - Bonded joint specimen in MTS test fixture and loading cycles.

<table>
<thead>
<tr>
<th>Loading (kN)</th>
<th>Cycle 1</th>
<th>Cycle 2</th>
<th>Cycle 3</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.00</td>
<td>0.00</td>
<td>0.00</td>
<td></td>
</tr>
<tr>
<td>44.50</td>
<td>44.50</td>
<td>44.50</td>
<td></td>
</tr>
<tr>
<td>89.00</td>
<td>89.00</td>
<td>89.00</td>
<td></td>
</tr>
<tr>
<td>133.50</td>
<td>133.50</td>
<td>133.50</td>
<td></td>
</tr>
<tr>
<td>146.41</td>
<td>146.41</td>
<td>178.00</td>
<td></td>
</tr>
<tr>
<td>0.00</td>
<td>178.00</td>
<td>267.00</td>
<td></td>
</tr>
<tr>
<td>222.50</td>
<td>366.24</td>
<td></td>
<td></td>
</tr>
<tr>
<td>267.00</td>
<td>400.50</td>
<td></td>
<td></td>
</tr>
<tr>
<td>311.50</td>
<td>445.00</td>
<td></td>
<td></td>
</tr>
<tr>
<td>366.24</td>
<td>489.50</td>
<td></td>
<td></td>
</tr>
<tr>
<td>267.00</td>
<td>534.00</td>
<td></td>
<td></td>
</tr>
<tr>
<td>133.50</td>
<td>578.50</td>
<td></td>
<td></td>
</tr>
<tr>
<td>0.00</td>
<td>623.00</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>667.50</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>712.00</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>755.50</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td>To Failure</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Figure 7 - Strain gage locations on test specimen.
Figure 8 - Load versus strain for strain gages #1 and #10.

Figure 9 - Load versus strain for strain gages #3, #4, #5, #12.
Specimen #1, a bonded joint, had a failure at approximately 378 kN (85 kips), whereas specimen #2, also a bonded joint, failed at approximately 600 kN (135 kips). Specimen #3, a bolted/bonded joint, had a partial paste adhesive failure occur at approximately 489 kN (110 kips) in the bond of the joint, and complete failure at 556 kN (125 kips) with bolt shear. Figure 10 shows the failed bonded specimen #2, which was used for comparison above. The failure occurred in the butt region of the EA9394 paste adhesive between the AS4/PEEK and titanium at post test observation.

Figure 10 - Bonded joint specimen #2 adhesive failure at 600 kN (135 kips).

Visual inspection of failed specimen #1 showed a large 97.6 mm (4 in.) diameter void in the butt region of the paste adhesive due to a trapped air pocket during the assembly of the joint. This defect probably contributed to the premature failure of the joint when compared to the results of specimen #2 and #3. Figure 11 shows the failure of bonded/bolted joint specimen #3. The EA9394 paste adhesive failed again on the butt region of the AS4/PEEK and then continued to fail along with bolt failure. The bolt was sheared at failure. A large gap was observed prior to failure between the AS4/PEEK and the titanium.
Summary

Validation of a carbon thermoplastic to titanium bonded joint design for the Composite Storage Module was performed through destructive testing and a comparison to theoretical finite element predictions. Three-dimensional and two-dimensional solid finite element analyses were developed to perform the large deflection analyses of the components of a carbon thermoplastic, AS4/APC-2, to 6Al-4V titanium bonded, and bonded/bolted joint using ABAQUS [3]. Three full scale specimens were fabricated by Productions Products. Analytical predictions were compared with tensile loading results obtained from tests conducted at Naval Surface Warfare Center. Preliminary failure load prediction of 445 kN (100 kips) for the paste adhesive was similar to the average adhesive failure observed of 489 kN (110 kips). Although the first adhesive joint failed at 378 kN due to a large void in the paste adhesive, failure was 1.03 above the design failure load. The insertion of a bolt in the joint provided for additional reserve strength (556 kN) after the adhesive failed at 489 kN (110kips). However, the addition of this “chicken bolt” adds significant cost and difficulty in assembly of the joint and is not required to satisfy the load requirements.

Acknowledgments

This research work was funded by the Office of Naval Research, under James J. Kelly/Ignacio Perez, Materials Program Manager.
References


Mechanism of Adhesive in Secondary Bonding of Fiberglass Composites with Peel Ply Surface Preparation


ABSTRACT: The surface composition and morphology of fiberglass/epoxy composites prepared by the use of "bondable" peel ply and sanding will be presented. "Bondable" peel ply is a material which provides a surface ready for secondary bonding following peel ply removal without requiring further treatment. The available results indicate mechanisms which may degrade bonds using "bondable" peel ply. Those same results also give insight into how "bondable" peel ply achieves a bondable surface when conditions are optimal. Optimal conditions can produce a surface whose 90° peel strength is 12-15 lbs/in-width versus sub-optimal conditions which produce strengths of 1-4 lbs/in-width. The case studied in this effort is 350°F curing fiberglass/epoxy bonded secondarily with 250°F curing toughened epoxy adhesive.

KEYWORDS: peel ply, composite bonding, surface preparation, adhesion.

Introduction

Materials known as peel ply and their influence on adhesive bonding between fiberglass skins and honeycomb in composite structures is the primary topic of this paper. It is known in industry that peel ply applied to the surface of a composite can increase the bondability of that surface, but the mechanism that promotes this behavior is largely unknown. Depending on the composition of the peel ply, this increase can eliminate the need for further surface preparation. This paper attempts to explain how non-release treated peel ply promotes bonding. Using film adhesive, fiberglass composite is bonded to polyamide honeycomb core and tested under peel loading to observe the peel strength at failure and the mode of failure. Nylon peel ply and two types of polyester peel plies (Type 1 considered "optimal" and Type 2 considered "sub-optimal") are examined to determine the similarities and differences among them. The surface morphologies of the fiberglass after removal of the peel ply, the adhesive surface after bonding and peeling, and the peel ply surface are also examined. The composite "skin" used in this experiment is composed of several layers of precured, unidirectional, preimpregnated fiberglass cured in contact with peel ply. Removal of the peel ply produces the surface to be subsequently bonded to the
honeycomb core. The core is a lightweight polyamide core. The adhesive is a 250°F-curing toughened epoxy film. An example of this structure is shown in FIG. 1.

![Composite Bonded Structure](attachment:composite_structure.png)

**a. Composite Bonded Structure**

![Peel Test](attachment:peel_test.png)

**b. Composite Bonded Structure Showing Peel Test**

**FIG. 1. Basic Composite Bonded Structure and Actual Test Article Showing Peel Test**

In this structure, there are several loci or modes of potential failure:

1. Cohesive within the honeycomb.
2. Cohesive within the fiberglass composite skin. This type of failure is usually associated with overly aggressive surface preparation of the precured skins for bonding.
3. Cohesive within the adhesive. This is usually a desirable locus of failure in an acceptable peel test but can result from weakening of the adhesive, such as that caused by moisture in the adhesive during cure.
4. Failure at the interface of the adhesive with the honeycomb (Interface B). This type of failure is usually associated with contamination on the honeycomb surface prior to bonding.
5. Failure at the adhesive / fiberglass skin interface (Interface A).
Most peel test failures include a mixture of failure modes but this paper will focus on the mode that has been present in both low and high strength peel test failures, which is Failure Mode 5. This mode is the most interesting for the purpose of this paper because it represents the surface having been in direct contact with the peel ply prior to bonding. Upon removal of the peel ply, a three dimensional imprint is left on the fiberglass skin surface. Ideally, the surface is clean and possesses freshly fractured resin. It also has an increased surface area for bonding. If the surface left by the peel ply is effective, adhesion at this interface can be very good, producing either high peel values or forcing the failure to a different location. If the peel ply is ineffective and an additional surface preparation is not performed (e.g. sanding of the surface), very low peel strength is the inevitable result. Superficially, the failure modes that produce high and low value failures at this interface appear similar.

It is recognized the interfacial behavior of the adhesive to skin bond is only one of several factors that affect the final strength of an adhesively bonded honeycomb structure. For example, it is possible to change the peel failure locus from the subject interface to completely within the core while lowering the peel value. This was achieved by altering the physicochemical properties of the adhesive. However, the adhesive / skin interface is sufficiently complex that this paper will be confined to the effects of peel ply with only limited references to other factors. The other factors will be addressed in future publications.

**Specimen Configuration and Manufacture**

The panel configuration shown in FIG. 1 is similar to some of the parts manufactured by Sikorsky Aircraft. Some parts have machined angles over approximately 1 inch of the edge of the core block. These angles can be up to approximately 15° from perpendicular to the core cell walls. These parts have been included in this study. The test panels were fabricated by laying up a 3-ply fiberglass skin on a caul plate with mold release. Peel ply was applied to both surfaces followed by a bleeder layer and covered by a vacuum bag. This assembly was cured at approximately 90 psig, vented, at 350°F for 2 hours. The peel ply was then removed. In some cases, the peel ply was examined for damage along with the surface of the skin itself before bonding. The next step, in some cases, was abrasion of the skin for additional surface modification. Following the preparation of the skin for bonding, the adhesive was applied to the skins and a skin was bonded onto each side of the core using a 250°F / 30 psig, vented autoclave cure cycle (see FIG. 1a). After the cure was complete, specimens were cut from the panels and tested in peel, generating additional specimens for surface examination, along with peel strength failure loads. The peel test employed was similar to a simple 90° “T-peel” test using a modified “fish scale” force measurement (see FIG. 1b).

**Observations on Nylon Peel Ply Specimens**

FIG. 2 is a Scanning Electron Microscopy photomicrograph of a surface produced by non-release treated nylon peel ply after removal of the peel ply from the cured skin.
Notice the negative image pattern of the peel ply is replicated on the epoxy surface on the scale of the width of the individual fibers and larger. Another feature of importance is the relatively small amount of visible broken resin. The micrograph shows the epoxy resin of the skin penetrating the yarn. Where the fibers contact their neighbors, the resin penetrates deep within the fiber bundle. The result is a very small area of broken resin between the fibers within a yarn or bundle when the peel ply is removed. Only the top of each "knife-edged" resin feature between fiber imprints on the surface is broken resin. Additional broken resin can be seen at the intersection of the warp and fill. This broken resin is the remains of the resin that flowed to fill the void where the fibers of the intersecting yarns in the peel ply could not. These areas are generally the largest areas of broken resin visible on the surface of the skin after removal of the peel ply. Notice that the total area of broken resin is a small fraction of the total surface. Surface abrasion of peel-ply-produced surfaces is often used to increase the area of broken resin after removal of the peel ply. In the case of nylon peel ply the mechanical abrasion can increase the subsequent adhesion at the interface significantly [1].

Another interesting feature visible in FIG. 2 is smaller in dimension than the individual fiber diameter. That feature is enlarged in FIG. 3.
This feature appears to be a ductile residual from a fiber that previously resided in that channel. Such a ductile feature is not typical of the overloading of epoxy resin. Overloaded epoxy resin fractures in a more brittle mode, an example of which was previously indicated at the intersection of yarn imprints. The ductile feature is more typical of the nylon which makes up the peel ply itself. Notice, however, that larger scale features on the skin are good replicas of the peel ply cloth, so very little of the cloth material can remain after removal of the peel ply. An additional fact to be considered here is the relative difficulty in removing the peel ply from the skin. Nylon peel ply is much more difficult to remove than is polyester peel ply. The conclusion to be drawn from this information is that a very thin skin of each nylon fiber adheres to the surface of the epoxy. Fracturing the nylon fibers during removal of the peel ply leaves a thin skin of ductile Nylon material seen in FIG. 3. It also explains the amount of energy required to remove nylon peel ply, while apparently not fracturing larger amounts of resin in the epoxy skin material. It too explains why sanding, which can remove the layer of nylon in certain areas, promotes adhesion.

**Observations on Polyester Peel Ply Specimens**

*Type 1 Polyester Peel Ply ("optimal")*

FIG. 4 illustrates the fiberglass surface after removal of polyester peel ply Type 1.
Note the same negative image effect and epoxy channels seen using nylon peel ply. The main difference between the polyester and nylon produced channels is the absence of a polyester film layer and the presence of brittle epoxy fracture. Instead of the nylon layer, there is a clean surface of epoxy from the fiberglass prepreg. This surface is considered ideal to maximize bond strength. This type of polyester peel ply is considered “peel and bond”. Under some circumstances, no additional surface preparation is required before subsequent bonding.

FIG. 5 is a photomicrograph of the Type 1 peel ply surface after removal from the fiberglass. Note the limited evidence of frayed or broken fibers. This polyester peel ply did not leave much residue on the surface of the fiberglass. This could account for the relative ease with which this peel ply is removed (significantly easier than the nylon).

**Type 2 Polyester Peel Ply (“sub-optimal”)**

FIG. 6 shows the fiberglass surface produced by the Type 2 polyester peel ply which is not considered “peel and bond”. This micrograph shows fractured tendrils that have peeled off from the polyester fibers and remained on the surface. The tendrils are mainly following one direction, which corresponds to the direction of pulling when removing it from the fiberglass. These tendrils indicate the polyester fibers used to fabricate the peel ply did not possess enough strength to maintain their fiber integrity. This “contamination”, if not removed by additional surface preparation, can produce inferior bonding.
FIG. 5. Surface of optimal polyester peel ply after removal.

FIG. 6. Fiberglass surface after removal of sub-optimal polyester peel ply.
FIG. 7 shows the surface of the Type 2 peel ply after removal from the fiberglass. The tendrils can also be seen in these photos. Type 2 peel ply removes as easily from the fiberglass surface as Type 1. It is similar to nylon in that Type 2 leaves remains of itself on the fiberglass surface, and in order to bond, additional surface preparation of the fiberglass is required.

![Failed Peel Ply](image)

FIG. 7. Sub-optimal polyester peel ply surface after removal from fiberglass.

**Observations of Failed Bonds**

Load values obtained for 90° peel testing ranged from low (1-4 piw) to high (12-15 piw) where piw is pounds per inch width. The same adhesive to skin failure mode is seen for the low and high peel loads, identified as Failure Mode 5. In addition to the peel values, there is another subtle macroscopically viewable feature: crazing of the adhesive seen with high peel loads. Crazing is usually only seen with specimens exhibiting particularly strongly bonded adhesive which produce peel strengths higher than 15 piw. The crazing effect requires additional information to be understood.

*Sub-optimal Peel Ply Specimen*

FIG. 8 details a peel test specimen that produced a low peel load value. This specimen had sub-optimal polyester peel ply on the bond surface and had no further surface preparation before bonding. Notice the lack of core failure, and that most of the adhesive remains on the surface of the core. In the upper portion of the photo, the skin to adhesive failure occurred where the core was in contact with the fiberglass; the adhesive
in the center of the cells of the honeycomb remains on the fiberglass surface. In the lower portion of the photo the failure was different because there was an extra layer of adhesive. The extra adhesive reinforced the honeycomb cells, which prevented the failure of the core during peel. The weakest bond, between the adhesive and the fiberglass skin, failed instead of the honeycomb. This switch in failure mode will be discussed in a future publication.

The fracture surface where the adhesive was removed from the fiberglass skin has been enlarged and can be viewed in FIGS. 9 and 10. No fractured adhesive can be seen in the channels left by the polyester peel ply, nor in the areas where the warp and fill of the peel ply intersected.

FIGS. 11 and 12 show the surface of the adhesive remaining on the core for the peel test. The surface where the adhesive detached from the fiberglass has the same morphology that the peel ply imprint formed on the fiberglass surface, but it is a negative image to the fiberglass surface. There is also what appears to be partially peeled adhesive within some of the peel ply produced channels. This indicates the adhesive flowed into the channels on the fiberglass, but did not bond to the prepreg epoxy. Instead, the adhesive was prevented from bonding due to the polyester remaining in the channels (like nylon), which produced an inferior bond and allowed the peel specimen to fail at a low peel load. Pieces of the peel ply can be seen still attached to the adhesive surface, which was ripped away from the fiberglass surface. This effect clearly illustrates the need to control the properties of the polyester fibers used to fabricate peel ply.
FIG. 9. Fiberglass surface after peel test.

FIG. 10. Enlarged view of fiberglass channels after peel test showing polyester contamination.
FIG. 11. Adhesive that remained on the core surface.

FIG. 12. Enlarged view of adhesive that remained on the core surface.
**Optimal Peel Ply Specimen**

FIG. 13 shows peel test specimens that produced high peel load values. These specimens had optimal polyester on the bond surface with no additional surface preparation. They show a failure mode of skin/adhesive, where the adhesive appeared macroscopically to remove cleanly from the fiberglass skin during the peel test.

On closer examination of the fiberglass surface (FIG. 14), fractured adhesive can be seen in the channels left by the polyester peel ply, as well as the areas where the warp and fill intersected. Note the contrast between this failure mode and that illustrated in FIG. 10 and FIG. 12. This photo indicates a mechanical interlock mechanism of bonding is present. The apparent sequence of events is that the fiber of the peel ply pulls out of the epoxy of the fiberglass upon peeling of the peel ply. The separation of the epoxy from the polyester is reasonably complete, leaving channels in the epoxy where the fibers were imbedded. No residue from the peel ply has been detected on most of the channel surfaces.

Some of the channels in the skin surface formed by the removal of the peel ply fibers are actually larger below the surface than at the surface. The result is a trap (FIG. 15) where liquid adhesive can flow during the secondary bonding of the skin. After the solidification of the adhesive a mechanical lock forms between the adhesive and the precured skin at these entrapment points.

![FIG. 13. High peel value bond test using optimal peel ply.](image-url)
FIG. 14. Fiberglass surface after peel test.

FIG. 15. Illustration of trap formation and subsequent filling by adhesive bonding.

Conclusions

Adhesion to surfaces produced by peel ply appears to be dominated by mechanical interlocking. Nylon peel ply, which is more difficult to remove than polyester, leaves
residue appearing to interfere with subsequent adhesion. Optimal polyester peel ply can produce rather "clean" channels that contribute to mechanical interlocking at the laminate/adhesive interface. Less optimal peel ply produces fiber imprint channels that are “contaminated” with shattered fiber. It should be possible to improve peel ply performance by improving on the current products (e.g., stronger polyester fibers). With current peel ply products care must be taken in the choice of prepreg, adhesive, peel ply composition, and surface preparation.

Acknowledgements

We would like to acknowledge the following contributions to the effort to understand the role of peel ply in secondary adhesive bonding. John Houston, Joe Holder, and Ali Kahn of Precision Fabrics group have aided with samples and hours of discussion related to peel ply fabrication. Ray Krieger, John Hattayer, and Dr. Richard Mayhew of Cytec Engineered Materials have assisted with specimen examination and hours of discussion, particularly relating to interaction of the adhesive with the other components of the bonded honeycomb structure. This work could not have been performed without the help of these individuals.

References

SECTION II: ADHESIVELY BONDED REPAIR
Static and Dynamic Strength of Scarf-Repaired Thick-Section Composite Plates


ABSTRACT: Composite structural armor is typically a sandwich construction consisting of thick-section polymer composite, rubber, and ceramic layers, which are combined to provide an optimal balance of structural and ballistic performance at minimum weight. Design guidelines and repair techniques are needed for this unique class of multifunctional construction. Our focus is on the scarf repair of the thick-section composite backing plate subjected to static and dynamic loading. Composite backing plates of plain weave S2-glass fabric and SC15 epoxy resin were manufactured by the VARTM process. Deliberate damage to backing plates was repaired at elevated temperatures using induction heating and at room temperature. The static response of control and repaired plates was compared via four-point bend testing. The effect of three scarf angles (45°, 18.4°, and 11.3°) and four repair adhesives (two room-temperature and two elevated-temperature cure systems) was quantified. Using these repair techniques, renewal of stiffness was achievable, except for the case of the highly ductile, low stiffness adhesive. The renewal of moment capacity of the repair beams was highly dependent on the scarf angle for various adhesives, and a maximum renewal of strength was 60%. The dynamic strength of scarf patch repaired composite specimens was investigated through axial compression strength testing using a split Hopkinson bar. Under dynamic loading the axial strength was found to be dependent on the scarf angle and rate of loading. Loci of failure are reported for the various materials, scarf angles, and loading conditions.

KEYWORDS: repair, induction cure, thick-section composites, axial compression, Hopkinson bar, failure analysis

Introduction

Composite structural armor typically consists of different material layers stacked together to provide unique structural and ballistic properties, as well as satisfying other multifunctional requirements (e.g., fire, smoke and toxicity (FST) resistance,
electromagnetic shielding, etc.) [1-4]. This program used a simplified four-layer configuration that consists of a composite cover layer for durability, a layer of ceramic tile for ballistic protection, a rubber layer, and a thick composite backing plate for structural and ballistic performance (Figure 1). Fabrication of the composites used the vacuum assisted resin transfer molding (VARTM) process, which has been shown to provide superior mechanical properties in a single-step operation as compared to bonding individual layers in a multi-step process [5]. Additional details on structural behavior can be found in [5-7].

Under ballistic impact, both the cover layer and the backing plates could be completely perforated and the extent of damage in the ceramic tiles varied from a single tile to all of the tiles surrounding the impact site (see Figure 2). Multiple interfaces in the backing plate also had delaminations that were larger than the extent of damage to the ceramic tiles (i.e., 3 tiles in length). Extensive fiber damage also occurred at the impact site. Experiments have shown that such damage degrades the ballistic performance of the structural armor [8]. The compression strength after ballistic impact was also shown to drop to levels approaching 25% of the virgin strength [9]. Repair that is capable of renewing the structural and ballistic performance after a ballistic impact is a key issue for the use of composites for Future Combat Systems (FCS).

The extent of damage in different layers determines the level of repair to be performed. Different repair strategies and repair methods, that utilize conventional repair techniques or induction curing techniques, have been documented in previous studies [8,10]. Three levels of repair have been identified according to the level of damage through the thickness of the armor. Level I is concerned with the repair of the cover layer only, as in the case of damage due to a low velocity impact. Level II represents the case of the repair of both the cover layer and the ceramic strike face. Finally, Level III addresses the repair of all the layers that compose the structure, including the repair of the composite backing plate, as in the case of the high velocity ballistic impact damage shown in Figure 2. A repair by resin infusion was attempted [8], but it was shown to provide only moderate improvement in the ballistic performance of the repaired panels. Since, resin infusion does not effectively repair fiber damage, the present study evaluates a scarf repair that enables all damaged materials to be replaced for Level III repairs.
The repair should renew stiffness, strength, and ballistic performance to meet design requirements. The present effort on scarf repair builds on the procedures developed by Gama et al. [10] who demonstrated a Level III repair of structural armor. The repair involved the removal and replacement of all the layers shown in Figure 1, including that of the composite backing plate. Following the removal of the damaged cover layer, and the damaged ceramic tiles, the damaged region of the backing plate was removed using a wet grinding tool to form a circular hole. The edge of the hole was machined to produce a 45° scarf angle. The backing plate was then repaired with an adhesive bonded flush scarf plug of the same material and the ceramic and cover layer were replaced. The repair to the thin cover layer used a flush plug repair (i.e., butt joint). The repair of the entire cross-section was done in a single-step operation from one-side that used a vacuum bag for consolidation and induction heating for rapid heating and cure. Various susceptors were used at the various interfaces to locally heat the adhesive bond lines to cure temperatures without over-heating any single interface. A repaired demonstrator panel was manufactured to prove process viability, but was not tested. These repair techniques were used in the present study to fabricate test specimens to characterize the structural performance of the backing plate with scarf repair subjected to static and dynamic loading.

Composite structural armor is subjected to bending moments and shear forces due to terrain-induced loads and lateral impacts. Some understanding of the complex interaction between layers of structural armor is needed to establish simple test methods that evaluate Level III repair approaches and maintain some relevance to the application loads. In previous work [11], the finite element method was used to model the deformation behavior of the armor in bending. The through-thickness strain distribution deviates greatly from that of the linear classical analysis due to the compliant rubber layer that decouples the ceramic from the composite backing plate. These results show that it is possible to idealize the behavior of the backing plate in the structural armor as a beam subjected to bending loads. In addition, in-plane axial compression loads can also be used to evaluate the strength of bonded joints. The present study focused on developing simple test methods to quantify the performance of scarf-repaired thick-section composite backing plates of composite structural armor.
Test Methods

The four-point bending test was selected to characterize the static behavior of the virgin and repaired beams. The nominal dimension of the test beam was 889 mm x 30 mm x 13.2 mm. The support span was 762 mm, which provided a span-to-thickness ratio of 57.7. The span of the loading noses was sufficiently large (381 mm) to include the scarf repair. The specimen and test configurations are shown in Figure 3. An Instron 8562 (servo-hydraulic) machine equipped with a custom built four-point bend test fixture (support and loading nose diameter of 25.4 mm) was used to conduct the experiment at a constant crosshead displacement rate of 2.5 mm/min. Load and crosshead displacement data were collected using the Instron Series IX data acquisition software.

Dynamic axial compression tests of scarf-repaired composite joints were performed using the compression Split Hopkinson Pressure Bar (SHPB) technique [12] (Figure 4). The results generated were representative of the response of the scarf repair joints loaded in axial compression. The striker bar, the incident bar, and the transmitter bar were all made from Inconel 718 alloy (Young’s modulus – 200 GPa, bar velocity – 4920 m/s, Poisson’s ratio – 0.29, and diameter – 19.05 mm). Specimens of nominal cross-section (13.5 mm x 13.5 mm) were machined from scarf-repaired composite plates. The lengths of the specimens were 37.5, 70, and 90 mm for 45°, 18.4°, and 11.3° scarf angles, respectively. In order to load such long specimens for sufficient duration, a long striker bar (711 mm) was used to produce a long incident pulse. A rubber disk was used between the striker bar and the incident bar to shape the pulse and to uniformly load the specimen. The impact velocity of the striker bar was varied between 5 to 10 m/s, which produced a displacement rate of 2.5 to 5 m/s. Maximum force at failure was calculated using a ‘3-wave’ analysis from the incident, reflected, and transmitted stress pulses as obtained from each test. A minimum of two specimens was tested for each scarf ratio and adhesive used.
Figure 4 – Hopkinson bar procedure for dynamic axial compression of scarf repair composite specimens.

Materials

The backing plate considered in the present work consisted of 22 layers of Knytex plain weave S2-glass fabric of areal-density 0.81 kg/m². The lay-up orientation used was [0/90] (i.e., the fabric warp direction is at 0° and the weft direction is at 90°). The backing plates were fabricated by vacuum assisted resin transfer molding process (VARTM) [5], using an Applied Poleramic SC15 epoxy resin system, specifically developed for the VARTM process. SC15 has a tensile modulus of 2.7 GPa and strain to failure of 6%. The SC15 system gels at room temperature (8-12 hrs). Additionally, a four-hour post-cure at 149°C was used. The total thickness of the backing plate was nominally 13.2 mm with a volume fraction of about 50% and less than 1% void content.

The choice of a proper adhesive system for repair of structures is always crucial. The selected adhesive must offer good mechanical properties, high toughness, and meet service temperature requirements. It has to be polyvalent and adhere well to different materials. Also, a moderate viscosity is needed to fill the gaps in between the ceramic tiles and the adhesive should be easily spread. Finally, cure at a relatively low temperature in a short period of time is desirable.

Four adhesives were chosen for this study that offered a broad range of properties. The Dexter Hysol EA9359.3NA adhesive is a two-part system that cures at 82°C for 1 hour. According to the manufacturer’s datasheet [13], it has a bulk tensile modulus of 2.2 GPa, a tensile strength of 31 MPa, and an elongation at failure of 10%. The Dexter Hysol EA9394 adhesive is also a two-part system that cures at 66°C for 1 hour. According to the manufacturer’s datasheet [13], it has a bulk tensile modulus of 4 GPa, a tensile strength of 46 MPa, and a much lower elongation at failure of 1.7%. Both systems have been successfully cured using induction heating of structural armor. The use of a room-temperature cure adhesive was also considered. Plexus MA425 is a two-part methacrylate adhesive that cures in 30 minutes at room temperature. This adhesive provides good gap filling capability as well as a much higher elongation to failure (120%). This adhesive has a lower tensile modulus (345 MPa) and strength (17 MPa) compared to the elevated temperature cure adhesives. The fourth adhesive was also a methacrylate (Plexus AO420), which has similar mechanical properties, but a much lower curing time of six minutes.
Repair Procedure

The scarf patch repair concept shown in Figure 3 is a very efficient way of repairing highly loaded composite structures. Care must normally be taken to ensure that the scarf angle is low enough to allow for a smooth stress transfer between the two adherends. Scarf repairs of thin aerospace structures are commonly limited to angles ranging between 2° to 6°. This is not practical in thick-section laminates. In the present work on the repair of thick sections, the scarf patch was limited to a diameter of 350 mm (approximately three tile diameters, where extensive fiber damage occurs). Therefore, the maximum allowable scarf angle was about 11° (i.e., 1/5 scarf-ratio for a 13.2 mm thick adherend). Three scarf angles were investigated using our test methods. The angles included 45° (1/1-scarf), 18.4° (1/3-scarf), and 11.3° (1/5-scarf).

Twenty backing plates, 889 mm long and 30 mm large, were fabricated by the VARTM process [5] and used for the characterization of the static and ballistic performance of the repairs. Two backing plates were kept virgin and tested as control beams. Eighteen beams were machined to receive the scarf patch repairs, two beams for each repair adhesive and scarf angle combination. For the repair, a distance equal to 101.6 mm separates the lower tips of the scarf, as shown in Figure 3(b). The placement of the repair patch only required that the surfaces in contact be completely wetted with the repair adhesive. The repair stack was then placed in a vacuum bag for hardening of the repair adhesive.

The backing plates repaired using the elevated-temperature cure Hysol adhesives were cured by induction-heating technique as described in the next section. The Plexus repaired beams were cured at room temperature. The quality of the control and repaired beams was visually observed to be very good.

Induction Repair

Induction heating is a non-contact method by which electrically conductive materials (susceptors) are heated in an electromagnetic field. This technique has successfully heated multiple interfaces as shown in Figure 1 in a single-step operation, with the use of appropriately placed susceptors [14]. A stainless steel mesh susceptor, with a wire density of 5 by 5 per square cm and a wire diameter of 0.165 mm, was used in the present study. The stainless steel mesh was cut to the shape of the area to be bonded, cleaned with acetone, impregnated with adhesive, and then placed in the bond area between the parent and patch laminate (which was also covered with adhesive). The stack was then vacuum bagged and placed horizontally under the induction coil at an optimal stand-off distance. The power setting of the induction generator, the coil shape (see Figure 5), and the coil stand-off distance were selected for uniform heating of the bond line to the adhesive cure temperature. This relationship was established experimentally by using an actual repair backing plate that incorporated the susceptor mesh wrapped in a Kapton film to allow multiple heating cycles. A 2-turn rectangular coil was used for the fabrication of the 45° and 18.4° scarf repair as shown in Figure 5(a). A 4-turn spiral coil (shown in Figure 5(b)) was used for the fabrication of the 11.3° scarf repairs. The 11.3° scarf repairs were bonded in two passes, in order to assure a complete coverage of the bond area with the induction-heating field. The stack was placed underneath the induction coil at a selected
distance, and the power was increased from 50 to 100%, in 10% increments. Process temperatures were recorded with a thermal camera and an embedded thermocouple.

Figure 5 - Induction-heating of the elevated-temperature cure adhesives.

An AGEMA Thermo-vision 900 thermal camera was positioned in front of the set-up to capture the full field surface temperature of the bond area in real-time. An E-type thermocouple was placed at the susceptor/adherend interface to monitor the internal bond-line temperature. The bondline temperature was slightly higher than the surface temperature due to heat losses. Steady-state was reached in a few minutes for each increment in power. Once a susceptor steady state temperature of about 82°C (i.e., the cure temperature of Hysol EA9359.3) and 66°C (i.e., the cure temperature of Hysol EA9394) was achievable within the range of the power settings, the stand-off distance was recorded and this combination was used for further bonding trials. The steady-state surface temperatures, recorded from the thermal camera are presented in Figure 5 for the 18.4° scarf and the 11.3° scarf repairs.

Static Performance of Scarf-Repaired Composite Beams

The load-deflection behavior of all the specimens tested was observed to be linear elastic until failure. Since the span-to-thickness ratio of the beams was large (i.e., 57.7), classic beam theory was used to experimentally determine the bending stiffness per unit width of the beams under four-point bending.

\[ E_I I / b = (P / D_1) \cdot (E / 96) / b \]  

(1)

where \( E_I \) is the flexural modulus, \( I \) is the moment of inertia of the cross-section, \( P \), and \( D_1 \) is the instantaneous load and crosshead displacement in the linear-elastic region,
$L$ is the length of the support span, and $b$ is the width of the beam. A comparison of the bending stiffness of the control beams with that of the repaired beams is presented in Table 1. Results showed a 100% renewal of structural stiffness was attainable with the elevated-temperature cure repair adhesives. However, the room-temperature cure repaired backing plates were more compliant by approximately 15% due to the lower modulus of the Plexus adhesive (i.e., 0.345 GPa versus 2.2 GPa for EA9359.3NA). Since stiffness is typically the critical design parameter for composite armor structures, the lower stiffness of the Plexus repaired beams may prevent the use of the Plexus adhesive for repair of composite armor. However, it should be pointed out that the stiffness loss in a 3-D scarf repair is likely to be less than that measured in the 2-D beam specimen.

Table 1 – Bending stiffness (per unit width) of the repaired backing plates.

<table>
<thead>
<tr>
<th></th>
<th>Bending Stiffness, kN-m</th>
</tr>
</thead>
<tbody>
<tr>
<td>Undamaged</td>
<td>5.75</td>
</tr>
<tr>
<td>Elevated-temperature cure repaired</td>
<td>5.75</td>
</tr>
<tr>
<td>Room-temperature cure repaired</td>
<td>5.00</td>
</tr>
</tbody>
</table>

The strength renewal of repaired beams was evaluated using the ratio of moment capacity of a repaired beam to that of the control beam. The two baseline specimens (i.e., no repair) failed at an average bending moment of 8.79 kN-m/m (per unit width). The failure mode was observed to be a compressive failure of the fibers on the specimen surface layer. This failure mode did not cause catastrophic failure of the beam and only a minor change in compliance resulted. However, the moment capacity at the onset of damage was used as the baseline for quantifying the effectiveness of the repairs. The bending moment at failure of all the repaired backing plates tested is summarized in Table 2. The values given are an average value from two beams tested with minimal variation. In general, a major reduction in the moment capacity was observed and was strongly dependent on scarf angle and adhesive type. Furthermore, the repaired beams failed suddenly at the adhesive bond line (in contrast to the progressive failure of the baseline). The extent of strength renewal (i.e., renewal of bending moment capacity), with respect to scarf angles and repair adhesives, is shown in Figure 6(a) and (b) and ranges from 10 to 60%. The strength renewal of the induction-cured, Hysol adhesives is shown in Figure 6(a). The repair efficiency of the 45° scarf angle was low (approximately 20%) and was almost independent of the adhesive system used. The structural performance of the 18.4° and 11.3° scarf angles improved significantly as the scarf angle was reduced. It is observed in Figure 6(a) that the Hysol EA9359.3 repaired beams restored as much as 62% of the control strength of the backing plates, compared with only 43% for the beams repaired with the EA9394 adhesive. The improved performance of the Hysol EA9359.3 may be attributed to the higher elongation and toughness of this adhesive system. In a recent study [15], induction-cured EA9359.3 single-lap shear joints tested in tension and four-point bending were also shown to be stronger and tougher than similar EA9394 joints.
Table 2 – Bending moment at failure of the repaired baking plates.

<table>
<thead>
<tr>
<th>Bending moment at failure / unit width (kN-m/m)</th>
<th>Hysol EA9394</th>
<th>Plexus MA425</th>
<th>Plexus AO420</th>
</tr>
</thead>
<tbody>
<tr>
<td>45° scarf (1/1)</td>
<td>1.97</td>
<td>1.13</td>
<td>0.95</td>
</tr>
<tr>
<td>18.4° scarf (1/3)</td>
<td>4.60</td>
<td>3.36</td>
<td>4.05</td>
</tr>
<tr>
<td>11.3° scarf (1/5)</td>
<td>5.49</td>
<td>5.91</td>
<td>5.44</td>
</tr>
</tbody>
</table>

* To be compared with that of control beams, i.e., 8.79 kN-m/m.

The loci of failure of the repaired backing plates were, nevertheless, found to be similar for both elevated-temperature cure repair systems. The 45° scarf-repaired beams were seen to fail by an interfacial failure at the adhesive/metal mesh interface. Decreasing the scarf angle to 18.4°, resulted in a slight change in the failure mode, and the locus of failure was observed to be interfacial, but 50% between the metal mesh susceptor and the adhesive (top end of the scarf), and 50% between the adhesive and the parent (lower end of the scarf). The 11.3° scarf-repaired beams failed by an adhesive/parent interfacial failure. Some adherend failure was also observed to have occurred. The strength renewal with respect to scarf angle for the Plexus adhesives is shown in Figure 6(b). The structural performance was mainly dependent on the scarf angle, since the adhesive properties were similar. The repair efficiency was slightly lower (about 10%) for the 45° scarf angle compared to the Hysol adhesives. However, the repair efficiency increased to 40 and 65% for the 18.4° and 11.3° scarf-repaired backing plates, respectively. This level was comparable to that of the induction-cured EA9359.3NA repaired backing plates. The locus of failure of the 45° scarf was in the adhesive. Decreasing the scarf angle to 18.4°, promoted a mixed, adhesive/interfacial failure. The 11.3° scarf-repaired backing plates failed by interfacial failure. In summary, the 11.3° scarf angle enabled renewal of moment capacity approaching 65% of the static baseline.
This range of scarf angles is considered practical for the repair of thick-section backing plates used in structural armor.

**Dynamic Axial Compression of Scarf-Repaired Composite Plates**

Dynamic axial compression of scarf-repaired composite specimens was performed using the Hopkinson bar technique. The Hopkinson bar responses for Hysol EA9359.3 adhesive repaired specimens are presented in Figure 7. All of these specimens failed under dynamic loading. At lower impact velocity, a rubber pulse shaper generates an incident pulse that is almost triangular in shape. The incident pulse becomes trapezoidal in shape at higher impact velocities. The 11.3° scarf specimens did not fail when the pulse shaper was used, and thus these tests were performed without a pulse shaper.

![Figure 7 - Hopkinson bar responses of Hysol EA9359.3 induction cured scarf-repaired composite specimens under axial compression.](image)

The incident, reflected, and transmitted pulses were used to calculate the forces at the incident bar-specimen (IB-S) and specimen-transmitter bar (S-TB) interfaces, $F_1$ and $F_2$, respectively following the procedure described in Ref. [12]. The condition of stress equilibrium was checked using the non-equilibrium parameter, $R = 2(F_1 - F_2)/(F_1 + F_2)$ [15]. Figure 8 shows the bar-specimen interface forces and the non-equilibrium parameter calculated for the 45° and 11.3° specimens (bar responses presented in Figure 7). Stress equilibrium ($R = 0.09$) was achieved in the 45° specimen only at maximum/failure load. However, better stress equilibrium was achieved in the 11.3° specimen ($R = 0.06$ at failure). The average maximum force, $F_{\text{max}}$, was used to calculate the average axial strength of the specimen. Since the thickness of the adhesive bond is small as compared to the length of the incident bar, it is assumed that a volume element
in the adhesive layer is under stress equilibrium (see Figure 9). The axial strength was then transformed into normal and shear stresses along the failure plane (Figure 9b).

![Stress equilibrium in Hysol EA9359.3 scarf-repaired composite specimens.](image)

Figure 8 – Stress equilibrium in Hysol EA9359.3 scarf-repaired composite specimens.

![Stress analysis of scarf-repaired composite specimens under dynamic axial compressive load.](image)

Figure 9 – Stress analysis of scarf-repaired composite specimens under dynamic axial compressive load.

Figure 10 shows a plot of axial strength of the specimen as a function of scarf angle for quasi-static and dynamic loading cases for the various adhesives. One important observation from these tests is that the dynamic axial strengths of the scarf joints were higher than the quasi-static axial strength by a factor of 2-3 for each scarf angle. The influence of scarf angle was similar to the results shown in Figure 5 for the 4-point beam tests. The dynamic axial strength increased as the scarf angle decreased, for all adhesives under dynamic and quasi-static loading conditions (with the 11.3° scarf-repaired Plexus MA425 specimen being the exception).

The locus of failure was found to be 50% in the adhesive/metal mesh interface and 50% between the adhesive and parent material in the case of Hysol adhesives (Figure 11a), except for the 45° scarf for which the failure was 100% in the adhesive metal mesh interface. In the case of the Plexus adhesive joint, the locus of failure was found to be in the adhesive (Figure 11b). These failure patterns were very similar to those obtained from quasi-static four point bend tests.
Figure 10 – Axial compressive strength of scarf-repaired composite specimens under dynamic compressive load.

The axial compressive strength for the Hysol 9359.3 adhesive together with the transformed stresses in the scarf plane are presented in Figure 12(a). Results showed that the adhesive bond was subjected to normal compressive stress that varied significantly with the scarf angle. However, the shear stress was relatively constant in the range of 40-60 MPa. This suggested that the failure was governed by the shear stress component. In Figure 12(b), shear stresses of various adhesives are compared in case of static and dynamic loads. The results showed that the dynamic shear stresses at failure were significantly higher than those obtained under static loading. Furthermore, the Hysol adhesives offered higher dynamic shear strength compared to the Plexus adhesive.
Conclusions

The scarf patch repair scheme used in the present study is viable for Level III repair of composite structural armor. The scheme uses practical scarf geometries for thick-section applications, requires only one-sided access for repair and vacuum consolidation, and provides rapid heating via induction to cure adhesives at elevated temperatures. The repair procedure was demonstrated using four adhesive systems. Simple test methods were proposed to apply realistic static and dynamic loads to composite backing plates with scarf repairs.

The repair efficiency of scarf repairs, having scarf angles much greater than commonly used in aerospace applications, was assessed in a four-point bending test. The baseline behavior exhibited compression failure on the specimen surface followed by progressive failure. Scarf-repaired beams failed catastrophically in the adhesive bond, while the virgin specimens exhibited a progressive compressive failure of the fibers on the specimen surface layer. This difference in failure mode may indicate that the inherent energy-absorbing mechanisms of the composite were limited by the repair.

The results, however, have shown that complete renewal of stiffness is achievable for the elevated temperature cure adhesives (a slight reduction of 15% is measured for the low modulus adhesive). The degree of strength recovered (based on first damage in the baseline) from the repairs increased from about 10 to 20%, to 40%, and 60% for the 45°, 18.4° and 11.3° scarf repairs, respectively.

In the dynamic experiments scarf-repaired composite backing plates were subjected to compression loading using the Split Hopkinson Bar technique. The dynamic axial strength for all adhesives was higher than the corresponding quasi-static data by a factor of 2-3 for each scarf angle. The dynamic axial strength increased as the scarf angle decreased, consistent with the 4-point bend tests. The results showed that the dynamic shear strength in the scarf plane was also rate dependent and significantly greater than the static strengths. Higher rate impact testing is needed to fully characterize the strength and energy absorption capabilities of the scarf repair.
Based on the limited results generated for the adhesives considered in the present study, induction curing of Hysol EA9359.3 with a 11.3° scarf offered the best combination of structural and rate dependent properties.

Acknowledgments

The authors wish to acknowledge the financial support provided by the Strategic Environmental Research and Development Program (SERDP Grant No. DAAL01-98-K-0058) and the Composite Materials Technology (CMT) Collaborative Program sponsored by the U.S. Army Research Laboratory under Cooperative Agreement DAAD19-01-2-0005.

References


Installation of Adhesively Bonded Composites to Repair Carbon Steel Structure


ABSTRACT

In the past decade, an advanced composite repair technology has made great strides in commercial aviation use. Extensive testing and analysis, through joint programs between the Sandia Labs FAA Airworthiness Assurance Center and the aviation industry, have proven that composite materials can be used to repair damaged aluminum structure. Successful pilot programs have produced flight performance history to establish the viability and durability of bonded composite patches as a permanent repair on commercial aircraft structures. With this foundation in place, efforts are underway to adapt bonded composite repair technology to civil structures. This paper presents a study in the application of composite patches on large trucks and hydraulic shovels typically used in mining operations. Extreme fatigue, temperature, erosive, and corrosive environments induce an array of equipment damage. The current weld repair techniques for these structures provide a fatigue life that is inferior to that of the original plate. Subsequent cracking must be revisited on a regular basis. It is believed that the use of composite doublers, which do not have brittle fracture problems such as those inherent in welds, will help extend the structure’s fatigue life and reduce the equipment downtime. Two of the main issues for adapting aircraft composite repairs to civil applications are developing an installation technique for carbon steel structure and accommodating large repairs on extremely thick structures. This paper will focus on the first phase of this study which evaluated the performance of different mechanical and chemical surface preparation techniques. The factors influencing the durability of composite patches in severe field environments will be discussed along with related laminate design and installation issues.

KEYWORDS: composite doubler repairs, adhesive joint, Boron-Epoxy

Introduction

Advances in structural adhesives have permitted engineers to contemplate the use of bonded joints in areas that have long been dominated by mechanical fasteners and welds. Although strength, modulus, and toughness have been improved in modern adhesives, the typical concerns with using these polymers still exist. These include a strong sensitivity of mechanical properties to temperature change, concerns over durability in hot/wet environments, and an inability to

---

1Airworthiness Assurance Dept., Sandia National Labs, Albuquerque, NM 87185
Sandia is a multiprogram laboratory operated by Sandia Corporation, a Lockheed-Martin Company, for the United States Dept. of Energy under Contract DE-AC04-94AL85000.
quantify bond strength (i.e., identify weak bonds) in adhesive joints. As a result, deployment of bonded joints requires proper design as well as suitable surface preparation methods. This paper describes the first phase of a study into the use of bonded composite patches as a substitute for welding doubler plates to repair carbon steel structures.

Syncrude is one of only a few companies in the world that extracts oil from “oil sand” (bitumen) mineral deposits. The technique involves a mining operation followed by a mechanical and chemical process to produce crude oil. Current oil production is 260,000 barrels per day with a planned expansion to 465,000 barrels by 2008. Extreme fatigue, temperature, erosive, and corrosive environments induce an array of equipment damage. Shutdowns to facilitate repairs can cost in excess of $1M per day. Figure 1 shows several of the hydraulic and cable shovels along with the trucks used in the surface mining operation. The trucks hold from 240 to 380 tons of oil sand and can be filled in three to four passes of an excavation shovel. Oil sand recovery equipment can experience over a half million fatigue cycles in a single year. The unavoidable by-product of this use is that fatigue cracks are commonly found in the body, arms, frames, and buckets of this equipment. The current weld repair techniques for these structures only provide a temporary fix that must be revisited on a regular basis. Figure 1 shows one repair process that uses seam-welded fishplates (steel doublers) to repair boom cracks. Multiple plates were applied as the crack continued to grow across the width of the shovel boom.

Composite Doubler Repair Method

It is believed that the use of composite doublers, which do not have brittle fracture problems such as those inherent in welds, will help extend the fatigue life of Syncrude equipment and reduce the equipment downtime during mining operations. The bonded composite doubler repair technique was evolved for U.S. commercial aircraft repairs at the Sandia Labs Airworthiness Assurance Center [1-6]. The technique uses bonded composite laminates, or patches, in lieu of conventional, riveted or welded metallic doublers to repair cracks, corrosion or other damage in metallic structures. Two of the main issues for adapting composite repairs to Syncrude applications are developing an installation process for carbon steel structure and accommodating large repairs on structures up to 6 in. thick. Other impediments include the need to carry out the repairs in a field environment using a hot bonding process and dealing with complex and extreme three-dimensional stress fields in order to mitigate crack growth.

The repairs proposed in this study can best be described as a hybrid repair. First the fatigue crack will be gouged out and the material will be replaced in a fill weld process. Next, instead of welding doubler plates over the damaged region or leaving the fill-weld as the only defense against crack re-initiation, a composite doubler will be installed using a hot bonding process [4, 6]. It is believed that the fatigue performance of bonded composite doubler repairs will be superior to the equivalent welded connection in the high cycle, long life regime. This is due both to the lower stress concentrations and the minimization of the effects of inherent weld defects. Figure 2 shows a typical Boron-Epoxy composite doubler repair over a cracked metallic structure. The number of plies and fiber orientation are determined by the nature of the reinforcement required (i.e., stress field and configuration of original structure). The advantages of this approach include:

1. A bonded repair will help relieve the residual stresses brought on by traditional welding in thick structures.
2. Bonded repairs allow for rapid, coherent joining of materials. Load transfer occurs over the entire footprint of the doubler/adhesive (vs. discrete weld seams). The doubler can be tapered to gradually introduce the reinforcing effect. This greatly reduces stress concentrations
associated with discretely fastened or welded repairs. The more uniform load distribution provided by bonded joints improves fatigue life.

3. Adhesive joints provide localized support around the damaged region thus providing better crack mitigation than traditional repairs. This phenomenon is shown in Figure 3 where the short spring of a bonded joint performs better than the long spring represented by fasteners or fillet welds that are spaced far apart.

4. The directional stiffness of composite doublers can be tailored to address the critical loads so that reinforcement is produced only in the necessary directions.

5. This repair resists corrosion which may eliminate the onset of stress corrosion cracking.

6. Composite laminates are easily formed to fit the contour of tight radius areas.

The overall goal of this program is to establish the viability of using bonded composite doublers to repair equipment used in the oil recovery industry. This includes the development and validation of installation, design, analysis, inspection, and quality assurance processes and the comprehensive performance assessment of composite doublers in the selected applications. The high modulus of boron-epoxy composite material enables a doubler to effectively pick up load when bonded to a metal structure. Load transfer occurs by shear through the adhesive. Previous experience in aircraft applications [1-2, 6] has shown that the use of a single part toughened epoxy adhesive will produce optimum mechanical properties in the joint. All tests were
conducted with FM-73 adhesive which is cured with a hot bonding process. It is assumed that this film adhesive has sufficient gap filling properties to accommodate the small degree of surface irregularities expected in the Syncrude structures. The portion of the program presented here determined the optimum surface preparation in order to meet both the bond strength and bond durability requirements.

**FIG. 2: Schematic and isometric view of a bonded composite doubler repair**

**A. Joint Continuity of Bonded Doubler Reduces Stress Levels in Damaged Region**

**FIG. 3: Short spring representing the localized crack mitigation of bonded doublers versus the longer spring that allows cracks to open and grow more rapidly**
Surface Preparation

The prebond surface treatment to which steel is subjected can significantly influence the resultant initial strengths and long-term durability provided by such bonds. The key to structural bonding is that the adherend surfaces must be roughened and free from contamination and weak oxide layers. Proper surface preparation will produce these features on the material and will allow for a reliable joint with sufficient strength and durability [7]. In order to optimize surface preparation, the basic mechanisms of adhesion must be considered. The adhesive must be able to wet the entire surface of the adherend so that there is intimate molecular contact at the joint interface. There are two basic mechanisms of adhesion for structural adhesive bonding: 1) mechanical interlocking of the polymer with the adherend surface, and b) chemical enhanced bonding of the polymer molecules with the adherend surface layer. Both of these mechanisms were pursued using a tailored set of surface preparation options.

Since this study is aimed at field installation in industrial settings, it is necessary to use simple, repeatable, and economically viable processing procedures. Sophisticated chemical surface treatments are too expensive or complicated to produce repeatable results. Ultra high temperature and high pressure process are also not feasible in the field. To date, bonded composite doublers have been used on thin aluminum structure and the surface preparation uses a phosphoric acid anodize etch. This approach requires a tailored phosphoric acid containment system that is time consuming to apply and potentially hazardous in the repair applications for this program. The surface preparation options evaluated in this study were selected on the basis of strength and durability, as well as, their ability to be carried out in heavy industrial settings.

The matrix of surface preparation options is summarized in Table 1. The surface degrease step involved an application of acetone followed by residue removal using methyl alcohol. Surface roughening for mechanical interlocking was provided by grit blasting or a simple oxide removal and sanding process. The surface chemistry was changed in some of the options through the application of silane, Sol-Gel, or Pasa-Jell. Finally, different primers were applied to assess any durability improvements they might provide in extreme environments. A description of each of these processes follows along with some discussion on several other surface preparation options that were eliminated from consideration prior to any testing.

Surface Roughening

Roughening surfaces prior to bonding enhances the strength of adhesive joints. The abrasive process removes contaminated layers, including hard-to-remove oxide layers, and the roughened surface provides some degree of mechanical interlocking with the adhesive. The process also forms a larger effective surface area for the bond and can introduce physical/chemical changes which affect surface energy and wettability. All of these issues must be considered in light of the characteristics of the adhesive and its ability to spread on different surface textures.

Experience has shown that any relatively smooth surface bonded to structural epoxy adhesives will suffer rapid delamination during conditions of high stress and humidity. Rougher surface morphology allows for better adhesion and resistance to the damaging effects of moisture. The task of providing a fine scale, stable, repeatable, rough surface on low carbon steels is difficult due to the surface condition, heat treatment, and metallurgical complexity of the steel materials. For these reasons, along with field installation impediments, chemical etching was eliminated from consideration.
Prior to roughening the surface, Scotch-Brite abrasion was applied to remove any oxide coating and debris. Additional grease and surface residue was removed with an alcohol cleaning. The relationship between surface texture and adhesion is complex and the relative magnitude and interactions of these different mechanisms need to be understood to optimize abrasive surface treatment processes. The grit blast process is used extensively to provide a clean and uniformly roughened surface for high strength bonding applications [8]. In this study, various grit blast energy levels were applied to the carbon steel plates in order to determine the best surface roughening performance for the hardness of our material. Related information from bonded repairs on aircraft was used to select grit type (aluminum oxide) and grit size (50 μm). In addition, the degree of surface roughness (≈50 μm. RMS) was chosen to match the level that produced optimized adhesion tests in aluminum bonds that utilized the same adhesive. This roughness number represents the average departure of the surface profile from the mean plane.

**TABLE 1: Surface preparation options for adhesive bonding**

<table>
<thead>
<tr>
<th>Option</th>
<th>Scotch-Brite Abrade &amp; Degrease</th>
<th>Grit Blast</th>
<th>Blow Off Surface - Oil-Free Air</th>
<th>Post Blast Degrease</th>
<th>Chemical Treatment &amp; Heat Lamp Dry</th>
<th>Primer &amp; Air/Heat Lamp Dry</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>Sol-Gel</td>
<td>BR-6747</td>
</tr>
<tr>
<td>2</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>Sol-Gel *</td>
<td>BR-6747 *</td>
</tr>
<tr>
<td>3</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>Pasa-Jell *</td>
<td>BR-6747 *</td>
</tr>
<tr>
<td>4</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>-</td>
<td>BR-6747</td>
</tr>
<tr>
<td>5</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>6</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>Silane *</td>
<td>BR-6747 *</td>
</tr>
<tr>
<td>7</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>-</td>
<td>Silane</td>
<td>BR-127</td>
</tr>
<tr>
<td>8</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>✓</td>
<td>Silane</td>
<td>BR-6747</td>
</tr>
<tr>
<td>9</td>
<td>✓</td>
<td>SAND</td>
<td>✓</td>
<td>✓</td>
<td>-</td>
<td>BR-6747</td>
</tr>
</tbody>
</table>

* Corrosion assessment coupons; 2 at each condition indicated

The relationship between the grit blast delivery pressure (energy level) and the corresponding surface roughness was determined. It was found that a 60 psi grit blast on carbon steel (ASTM 572) produced a similar surface roughness as a 40 psi grit blast on aluminum (2024-T3) during aircraft repairs. Filtering and drying systems were added to the compressed air supply line to reduce contamination. Figure 4 shows the test specimen preparation using the portable grit blast system. After blasting, a jet of clean air was blown across the surface to remove loose grit particles. The chemicals listed in Table 1 were then applied generously using brush-on or spray applications such that any contaminating residues were rinsed from the surface. To determine the feasibility of a quicker surface roughening option, hand sanding was included in the test matrix. Zirconium oxide sanding paper was applied in two perpendicular directions to produce a scratched surface that was sufficiently roughened but much less uniform in appearance. Figures 5 and 6 compare two surfaces that were grit blasted and hand sanded.

**Chemical Interlocking**

**Coupling Agents** The use of silanes to promote adhesion is well documented [9-10]. Use of the silane with a chemically compatible primer provides enhanced durability in hot-wet conditions. In this study the silane chemical was mixed with distilled water in a 99:1 ratio and
then brushed directly onto the grit-blasted steel surface. The silane was applied as follows: 1) diluted mixture was brushed on with natural hair brush such that the liquid flow was sufficient to check for a water break-free surface, 2) the surface was wetted for 10 minutes with brush applications, 3) the surface silane was blown off with instrument quality air, and 4) excess moisture was driven off via a heat lamp dry (120 to 150°F for 20 minutes). If streaks are noted during the blowing process it indicates the presence of grit and the silane wetting process is repeated to remove excess grit. Aircraft repairs employ a complete cure of the silane coating at 200°F for 60 minutes. In order to cure the composite doubler and adhesive film, the structure must be heated to 225°F. Since heating the massive, steel structures more than one time is impractical, this effort evaluated a co-cure process wherein final cure of the silane was achieved simultaneously with the adhesive.

![FIG. 4: Grit blast of steel surface using 50 micron aluminum oxide grit applied at 60 PSI](image)

![FIG. 5: 50X microscopic view of grit blasted surface at 60 PSI (surface roughness = 52 μinch RMS)](image)

![FIG. 6: 50X microscopic view of hand sanded surface (nonuniform – no roughness value)](image)

Sol-Gel is another prebond surface treatment that facilitates adhesive bonding to metallic surfaces [17]. Sol-Gel formulations developed by Boeing were designed to work optimally with aluminum alloys; however, successful testing has already been completed on stainless steel alloys. This study attempted to extend Sol-Gel use to carbon steel materials. The water based
Sol-Gel chemistry develops thin coatings that produce a gradient from the metallic surface, through a hybrid inorganic/organic layer, to the organic epoxy resin (adhesive). Enhanced adhesion is produced by the chemical interaction at the interfaces between the metal and the Sol-Gel and the Sol-Gel and the primer. The Sol-Gel was applied in the same fashion as the silane chemical mixture. It was also co-cured with the epoxy adhesive in a single heat cycle.

**Conversion Coatings** A number of chemical conversion coatings have been researched as surface treatment alternatives to degreasing, grit blasting, or other coupling agents such as silane or Sol-Gel [9]. These conversion coatings offer the potential for high treatment rates. They produce varying degrees of surface organic material and complex oxide coatings. Zinc and iron phosphate treatment solutions can precipitate crystallites onto the steel surface.

Surface analysis and mechanical joint testing have demonstrated high strength and durability in bonded joints prepared with these chemicals. However, one of the main problems with these chemicals is the formation of a loosely adhering layer of iron oxide “smut” that is difficult to remove before bonding but easily pulls away with the adhesive when the joint is stressed. If the grain size is not carefully controlled, the coating thickness becomes so great that one phosphate crystal precipitates on another and a weak interface is produced. Heat cure epoxies, such as the FM-73 film adhesive used in this study, are especially susceptible to this problem as thermal stresses between the thick conversion coating and the steel substrate can create fractures in the coating. Thus, this steel treatment has a high potential to fail at the metal/metal oxide interface at lower stress levels than expected for the other surface preparation options.

Another major problem with chemical conversion coatings is that the nature of the resulting surface topology is a function of the metallurgy of the substrate. As a result they do not provide consistent results and, in general, offer no performance enhancement over the more easily monitored and uniform grit blast technique. Because of these application drawbacks and concerns over achieving repeatable results, chemical conversion coatings were eliminated from consideration in the surface preparation test matrix.

**Steel Chemical Etchants** Effective chemical etching treatments for carbon steels have not been found. Unlike aluminum and titanium, iron does not form coherent, adherent oxides, so it is difficult to replicate the fine microroughness needed for good adhesion found on other substrate materials. Although various chemical etchants have been tried for both low carbon and stainless steels for many years, none has been widely adopted or shown to be superior to grit blasting [12]. One traditional etching chemical for aluminum bonding was considered in this study: Pasa-Jell.

Pasa-Jell is formulated for treatment of aluminum and titanium materials and has been shown to improve bond strength in these alloys by cleaning and producing a surface microroughness. It has a high content of nitric and chromic acids and is used for corrosion removal and surface etching. It also contains chemical activators and inhibitors that can improve the hydrolytic stability of an interface. Pasa-Jell is inorganically thickened to permit application in localized areas and on vertical or overhead surfaces. In this study, the extrapolation of Pasa-Jell to carbon steel materials was evaluated by coating the adherend surface with this paste and removing the residue with a water rinse. However, exposure times well beyond the routine 10 to 15 minutes failed to produce any appreciable change in the carbon steel surface. In fact, much of the oxide layer was still intact after almost 30 minutes of exposure to Pasa-Jell. These bench top tests eliminated the Pasa-Jell surface treatment (Option 3) before any specimens were manufactured. This was not a highly desirable surface treatment option anyway since it involved the use and disposal of hazardous materials.
Primers

To achieve satisfactory bonding with metals, it is often necessary to use primers. Metal surfaces that have been freshly abraded are highly reactive and will undergo rapid interaction with water or organic contaminants. Primers protect surfaces from contamination and improve overall bond performance. Based on their performance in aircraft applications and for both aluminum and steel materials, two different waterborne, chromate primers were evaluated [4,9-12]: Cytec Fiberite BR6747 (40-72°F storage) and BR127 (0°F storage). The BR6747 primer is more easily applied in field applications because: 1) it can be stored at room temperature, and 2) it can be co-cured with the adhesive. The BR127 primer is designed to be cured immediately after application and prior to installing the adhesive. However, due to the reasons cited above the BR127 primer was co-cured simultaneously with the adhesive during this investigation.

Corrosion Assessment During Surface Preparation

At various stages of the surface preparation process, test coupons were produced to assess the effects of exposure to atmospheric conditions. The tests were used to determine if corrosion or other bond inhibiting coatings would form on the steel surfaces during any steps of the surface preparation process. Two coupons were produced for each of the six conditions indicated in Table 1. These conditions cover before and after chemical treatment and before and after primer applications. The coupons were exposed to medium (60%) and high (90%) relative humidity for two days after the steps marked by asterisks in Table 1. During field repairs, it is anticipated that there will be unexpected delays in moving from one installation step to another. These tests evaluated the possible deterioration in a surface that might occur during such delays. However, the humidity chamber tests revealed that there was no degradation in the prepared surfaces nor did any corrosion form in 48 hours of exposure.

Evaluation of Surface Preparation Options

The purpose of mechanical testing of adhesive bonds is three-fold: to provide engineering design data, to serve as a quality control procedure, and to evaluate the performance and relative merits of various bonding processes. The joint strength and durability tests described below provided information on all three of these items, however, the primary goal was to rank the viability and effectiveness of the candidate surface preparation options.

The tests were performed using a series of coupon test specimens that were designed to approximate essential features and test critical elements of our repair. The adherends were mild steel (ASTM 572) similar to the highly weldable steel used in large shovel and truck construction. The specimens were lap shear and wedge-loaded coupons as shown in Figures 7 and 8. The lap tests provided the ultimate shear strength of the adhesive bond as well as the modulus values for the adhesive. In the wedge test, a wedge is inserted 0.75 in. into a 1 in. wide by 6 in. long coupon. The resultant crack growth is measured over time as the bonded joint is exposed to various environmental conditions. Both test series used 3-4 specimens at each condition and the average results are reported.

A single part, heat cured film adhesive (FM-73) was used to bond all of the test specimens. This adhesive has been shown to provide high strength and durability in the extreme hot, cold, and wet conditions experienced by aircraft [4,7,11]. The adhesive was cured for 2 hours at 225°F and 8 psi (simulating vacuum pressures attainable in the field). Hot-wet conditioning was achieved by 30 day exposure to 100% RH at 140°F. In addition, zero degree conditioning was used to study the performance of the bonded joints during winter operation. These environmental exposures were followed by coupon testing at the hot (140°F) and cold (0°F) temperature
extremes. Ambient conditioning and testing was also conducted to complete the range of expected operating conditions and to provide baseline data.

**FIG. 7:** Lap joint shear strength test of bonded metal specimens by tension loading with secondary bending

**FIG 8:** Wedge test to measure durability of adhesive-bonded surface

**Wedge Test**

The wedge test specimen combines the effects of stress with aging environments of temperature and humidity. Figures 9 to 11 show the crack growth as a function of exposure time to the various environments. In most of the samples, a significant percentage of the total crack growth takes place in the first 48 hours when the driving force for the crack is at its maximum. The crack growth tends to stabilize towards a particular value after about 200 hours of exposure. The total crack growth, however, depends upon the surface preparation process used for the joint. The more extensive crack growth in Figure 11 (hot/wet), versus growth measured in the Figure 9 (room temperature) and 10 (cold) conditions, show that water absorption, represented by hot/wet conditioning, is the primary cause of performance reduction.

At ambient conditions, Figure 9 shows that the surface with no chemical treatment (Option 4) performed the best followed by the co-cured silane-primer options (Options 6 & 8). As expected, all of the results are tightly clustered with only a 12% separation between the shortest and longest crack growth. At zero degrees (Figure 10), Option 4 continued to perform the best followed by both the silane and Sol-Gel surface treatment options. It is here that the benefits of grit blasting begin to appear as the hand sanding Option 9 allows for almost twice the crack growth as the
other grit blast processes. The primary benefits of grit blasting and the use of primers are seen in the performance of the bonds in hot/wet conditions.

**FIG. 9:** Comparison of wedge crack extension for eight surface preparation options - carbon steel bonds at ambient conditions

**FIG. 10:** Comparison of wedge crack extension for eight surface preparation options - carbon steel bonds at zero degree conditions
Figure 11 contains the largest spread in results with both the silane (Options 6 and 8) and Sol-Gel (Option 2) processes performing the best. Interface durability is a major limitation in the adhesive system. The addition of silane clearly improved interfacial durability. Use of primers—coupling agents that effectively stabilize the iron surface against corrosion—also improved the long term durability of the joint. Surprisingly, Option 4 with no chemical treatment continued to compare well, however, the lap shear results will determine if this good crack mitigation ability is accompanied by sufficient adhesive strength. The hand sand (Option 9) and non-primer (Option 5) methods performed the worst with crack growth approximately 95% greater than the optimum processes.

**Lap Shear**

The lap shear results are presented as histograms in Figure 12. This test series used 3-4 specimens at each condition and the average results are reported. At ambient conditions, the Sol-Gel and silane processes were superior with less than a 10% spread in the ultimate shear stress levels ($\gamma_{\text{ambient}} \approx 5000$ psi). The zero degree exposure tests showed similar results while also revealing that the ultimate shear stress levels are increased at reduced temperatures ($\gamma_{\text{zero}} \approx 5500$ psi). While the hot/wet conditioning reduced the ultimate shear stress levels in all surface
preparation methods ($\gamma_{\text{hot/wet}} \approx 4000 \text{ psi}$), processes using silane and Sol-Gel retained most of their initial strength. Options 6 and 8 retained 82% of their shear strength vs. the ambient test specimens while Options 1 and 2 retained 78% of their full baseline strength. In all cases, the simple, primer-only approach that performed well in the wedge tests (Option 4), produced the lowest ultimate shear stress in the test specimens.

The most critical test for bonded joints is durability in humid, high stress environments. Moisture may affect adhesive joints by: 1) hydrolyzing the adhesive, 2) displacing the adhesive at the adhesive adherend interface and, 3) by promoting corrosion at the interface with the steel adherend. Thus the adhesive as well as the surface preparation process must be able to resist this type of degradation. The hot/wet condition tests revealed that mechanical preparations alone do not provide durable bonding substrates. Virgin grit-blasted steel surfaces produced joints with relatively little durability. Figure 11 shows that the untreated steel surfaces (Options 4, 5, and 9) experienced more rapid crack growth during humidity chamber exposure.

Clearly, the important parameters include surface roughening and chemical treatment/protection. Results from joints without chemical coupling agents and/or primer are inferior – especially when exposed to hot/wet conditions. Examination of the failure surfaces showed that undesirable adhesive failure (disbonds along bond line) occurred in surface
preparation Options 4, 5, and 9. Cohesive failure (fracture of the adhesive indicating that the full strength of the adhesive was achieved) was observed in the Sol-Gel and silane surface preparation methods.

Optimum behavior of the bonded joints would cause the failure mode to be cohesive fracture of the adhesive (as opposed to adhesive disbond between the adhesive and the substrate). This would allow the full strength of the adhesive to be utilized in the joint and repairs could be designed using the entire ultimate strength of the adhesive. Since it produces a reduction in processing time, co-curing the primers with the adhesive provides a tremendous advantage over precuring the primer followed by a separate adhesive cure cycle. However, the reduction in processing time is not a benefit if durability is lost. Cohesive failure of the specimens revealed that co-curing the primer did not reduce the performance of the bonded joint.

Cohesive fracture indicates that the bonding interface is stable under the test environment and that the adhesive polymer is the limiting factor in the joint performance. On the other hand, joint failures that are predominantly interfacial (disbond between adhesive and steel surface) indicate unpredictable joint failure which will be dependent on its stress field and environmental exposure history. These results reinforce the previously derived conclusions from the wedge tests about the efficacy of the grit blast-silane-primer surface treatment.

Conclusions

The variables studied in this effort relate to the three basic methods of surface treatment for good adhesion: 1) removal of contamination or weak boundary layers such as corrosive scale or oxide layers, 2) changing the surface chemistry, and 3) changing the surface texture. By comparing the hand sanded surfaces with the grit blast family of bonds it can be seen that the uniformity of the roughness provided by grit blasting improves both the ultimate shear strength and crack mitigation capability of the bond. The use of chemical coupling agents and protective primers are also necessary to optimize the performance of bonded joints in carbon steel structures. Conversely, use of timesaving but marginal methods such as hand sanding plus only solvent cleaning produces weak bonds with poor environmental durability. The chemical treatment and primer co-cure processes validated in this study eliminate the need for two elevated temperature cure cycles. This minimizes complexity and saves installation time.

It was determined that both the grit blast-Sol-Gel-primer and the grit blast-silane-primer processes produce the best overall results for bond strength and long-term durability. The results were very repeatable with less than 3% variations observed among common test specimens. When ease of installation and availability of materials are considered, the silane process (Option 6) becomes the surface preparation of choice for this composite doubler repair program. Fracture tests revealed that the failures did not occur at the steel/silane interface or at the silane/adhesive interface, but rather at the boundary layer region that is strengthened by the presence of the silane coupling agent. The addition of the silane clearly improved the interfacial durability so that cohesive fracture of the adhesive could be reliably produced. By avoiding adhesive disbond failures, the full strength of the adhesive could be achieved. The results presented here provide the first step in applying adhesively bonded composite doublers as an alternative to the fusion fillet welding of steel plates to repair trucks and shovels used in the Syncrude mining and oil recovery operation.
References

SECTION III:
BOLTED ATTACHMENTS
ABSTRACT: One purpose of this paper is to describe and explain some highlights in the history of the analysis and design of bolted or riveted joints in fiber-polymer composite laminates. The second is to project into the future, where it might become practical to replace today's empirical analytical tools, which need copious test data to enable them to be applied, with physics-based composite failure criteria of universal applicability that need very few intrinsic material properties. The paper focuses on the analysis tools and is not a compendium of available test data. The paper begins with a review of the governing geometric parameters, along with their influence on which failure mode will dominate. The concept of optimum joint geometries is introduced, to show how to maximize the gross-section laminate strength and, thereby, to minimize the weight of these structures. It progresses to the two widely used analysis tools, developed years ago at the Long Beach and St. Louis divisions of the former McDonnell Douglas Corporation. The Douglas model is applied via explicit formulae, in conjunction with an empirically established stress-concentration relief factor. The latest forms of these equations are presented here, having changed little since they were first proposed almost 30 years ago. The McDonnell model, identified by the code name BJSFM is also based on algebraic solutions, but is encoded to enable a more thorough assessment to be made of the entire stress field, rather than just the most critical two locations. This code also relies on empirically determined correlation factors, in the form of characteristic offset distances, sometimes mistakenly believed to be true material properties. These same factors can also be applied to modify the predictions of finite-element analyses. The importance of using simple comprehensible models is stressed. The former model can also be used with the non-linear multi-row computer code A4EJ, again with very little need of test data – provided that the fiber pattern does not differ excessively from the quasi-isotropic lay-up. Excessively orthotropic fiber patterns are shown to be unacceptably weak at bolt holes and in need of a disproportionately large number of experimental test data to cover the many additional failure modes that cannot occur for close-to-isotropic fiber patterns. The paper closes with a glimpse of what the new SIFT (Strain-Invariant Failure Theory) for fiber-polymer composites, developed within the Boeing Company and already accepted for use at many locations outside, might do for this technology in terms of finite element analyses that need only five intrinsic material properties for each fiber-polymer combination, regardless of fiber pattern and joint geometry. The opportunities for drastic reductions in the cost of test programs, and accelerated schedules through not having to wait for specific tests for each joint, are clearly very powerful.

KEYWORDS: Composites, bolted joints, riveted joints, analysis, design, test data, SIFT failure theory.
Introduction

Thirty years ago, when he was forty years younger, the author recognized that the most appropriate way to design aerospace structures was to design the joints first and to fill in the gaps in between (the basic structure) afterwards. This was in response to so many sub-optimum designs resulting from studies that had tried to “optimize” the basic structure first. Nothing he has learned since would have changed his assessment of the relative priorities, which may explain why so much of his career has been devoted to bonded and mechanically fastened joints, in both composite and metallic structures. Nevertheless, one thing has changed. He would now rank the absence of joints, in places where they needed to be to restore continuity of structural load paths, as an even greater priority. Stiffener run-outs, in both metallic and composite structures, have become just as great a challenge as traditional splices. This is of particular concern in unitized metallic structures and co-cured composite structures that have been compromised structurally to simplify the manufacturing process.

The focus in this paper is on mechanically fastened joints in fiber-polymer composite structures. Three aspects of this issue are addressed. The first is a physical understanding of the basic phenomena, which need to be understood at both the macro-level and that of the fiber and matrix constituent components. No progress can be made by regarding fiber-polymer composites as merely homogeneous anisotropic solids. The second is a historical review of two of the most used empirical analysis/design tools for bolted composite joints that are based on lamina-level analyses but require empirical correlation factors to make them work. These are not blind curve-fits. There is a physical explanation for these parameters which, if understood, would tend to restrict their application to joint geometries for which the parameters are applicable and to inhibit their misapplication when they are not. The paper closes with a third perspective, in which the possible future role of physics-based composite analysis tools, that are universally applicable and which do not need any correlation factors to compensate for omissions in their capabilities, is discussed.

Reliable analysis methods for designing highly loaded bonded composite joints, such as the carbon-epoxy-to-titanium stepped-lap splice at the root of the F/A-18 wings, preceded the development of comparable methods for highly loaded bolted composite joints. The bolted deeply countersunk wing-fold joint outboard on these same wings required far more testing before the joint achieved an acceptable life than the more heavily loaded bonded inboard joint. Far better design procedures were in place for the bolted joints in the composite wings of the AV-8B Harriers, but it should be noted that the spars and skins were continuous from tip to tip, with no highly loaded chord-wise splices. It was at this stage that the bearing-bypass interaction was first applied, although it had been anticipated earlier during NASA-funded bolted joint studies. This interaction, which is illustrated in Fig. 1, requires that the bearing stress be restricted if the basic structure is to be operated at a high gross-section efficiency (normally thought of as a strain level). Fig. 1 is plotted as a function of the total load, bearing plus bypass load (to be reacted at other bolts). The interaction between these load increments themselves is also linear.

One can, of course, use local build-ups in thickness to overcome the limit on structural efficiency set by bolted joints, but only at the expense of repairability everywhere else in the structure. The AV-8B wings were designed to permit repairs by
bolted patches, to make it possible for the customers (the U.S. Marines) to maintain their aircraft quickly without specialized equipment to achieve a higher operational readiness than other design philosophies would have permitted.

Later NASA-funded research at the former Douglas Aircraft Company (now Boeing Long Beach) focused on the validation of structurally efficient highly loaded chord-wise wing-skin splices typical of DC-9 commercial transport aircraft. More details are given later in the paper. It suffices here to cite the successful result of a gross-section strain of 0.005 at a load intensity of 50,000 lbs./in. in a 1-inch thick carbon-epoxy laminate. But it needed a four-row splice, in double shear, and tapered splice plates. This was no easy task. In a concurrent USAF-funded research program at Douglas, methods were developed that permitted the establishment of optimal joint proportions.

Probably the most highly loaded bolted composite joints ever built are those at the root end of a boron-epoxy horizontal tail built by the former Rockwell Corporation (now part of Boeing) as a USAF R&D contract. The highest joint loads were resisted by thick local skin pad-ups rather than by optimum proportioning. Today, the carbon-epoxy wing skins of the Lockheed (now part of Lockheed Martin) F-22 are the most highly loaded production applications of bolted composite joints.

The preceding applications confirm that it is possible to design reasonably efficient bolted composite structures, but experience generally has also shown that this technology is also extremely unforgiving with respect to design details. There is none of the ductile yielding that so many designers have relied on when designing traditional metallic aircraft structures. Nevertheless, taking no credit at all for the limited non-linear capability of well-designed bolted composites joints, in which the initial failures are non-catastrophic, would so penalize the materials that metallic structures would always be lighter. The prime purpose of this paper is to encourage an expansion of what is still an art into something more akin to science, so that the process will be available to more engineers.
The Basics for Single-Row Bolted Composite Joints

Apart from empirical point-design methods relying entirely on test data, all analyses of bolted composite joints start with an elastic analysis, in closed form or by finite elements. These are then modified by empirically established coefficients. The best methods can cover a great many joint geometries and load conditions with the minimum number of such coefficients. The hope for the future is that it may one day become possible to analyze all such joints with only universally applicable intrinsic material properties of the fibers and matrix. As will become evident later, we are not there yet.

The first thing to be understood about bolted composite joints is the multiplicity of possible failure modes, as a function of the joint geometry and the fiber patterns, as indicated in Fig. 2. While individual line loads can be transmitted by adding width as well as thickness to the laminate, distributed loads, as at a chord-wise splice in a wing skin or at a longitudinal splice in a fuselage, eliminate width as a variable and the joint must be designed to maximize the strength per unit width. This is the origin of the concept of joint efficiency, which is explained later.

The strongest possible failure mode, per unit laminate width, is always net-section tension, followed by bearing failures. (There is a common perception that, even so, bearing failures are to be preferred because they are more benign than tension failures. Unfortunately, this is possible only for single-row joints, which severely limits the load that can be transmitted through such joints, in comparison with multi-row joints in which the individual bearing stresses are lower.) The weaker shear-out failures occur only in badly selected fiber patterns with too many longitudinal fibers in the laminate. (The undesirable characteristic of this failure mode is that the strength cannot be raised significantly by increasing the distance between the bolt and the end of the laminate.) Bolts bend, or fail, only when their diameter is too small in comparison with the laminate thicknesses. This is the extreme case of designing to excessive bearing stresses. The mixed-mode failure shown is a warning of the need to carefully select joint proportions for test coupons, so that unambiguous data can be extracted.
There are actually even more failure modes associated with off-optimum fiber patterns. Purely cleavage failures can occur if there is an adequate number of both $0^\circ$ fibers, in the load direction, and $\pm 45^\circ$ fibers, in combination with a deficiency of $90^\circ$ fibers perpendicular to the load. In addition, failure of the $\pm 45^\circ$ fibers where they are tangential to the bolt hole can occur in laminates with too few $\pm 45^\circ$ fibers to efficiently resist the high $0^\circ$ loads of $0^\circ$-rich laminates.

The second important characteristic of bolted composite joints is the relationship between average bearing stress and the peak net-section tension stress. These are regarded as independent variables for traditional ductile metals that yield extensively long before ultimate failure. Classical analyses via Fourier series established [1] that, for elastic isotropic materials, the peak tensile stress alongside a loaded hole is on the same order as the average bearing stress, as shown in Fig. 3. In other words, the prevention of tension failures in the $0^\circ$ fibers alongside the hole is dependent on limiting the bearing stress caused by the load applied to that hole in the $0^\circ$ direction. This is the origin of the paradox whereby, for fiber-polymer composites, the net-section strength can be increased by reducing the net section to permit larger holes with lower bearing stresses, up to the point at which the maximum strength is attained, as is explained in detail later.

![Bearing and Tangential Stresses Diagram](image)

**FIG. 3 – Hoop- and bearing-stress distributions around loaded bolt hole in wide isotropic plate**

The next issue addressed here is a number of empirical formulae derived for loaded and unloaded holes that serve as the basis for predicting the elastic strength of bolted composite joints, using isotropic materials as the starting point. The first such formula is the well-known classical prediction of a stress concentration factor of 3 for a circular hole in an infinite isotropic plate, first derived in Ref. [2]. The next formula was proposed by Heywood [3], for an unloaded hole in the middle of a uniform strip of constant width.
With reference to the terminology in the inset in Fig. 3, this relation between the peak stress alongside the hole and the average net-section stress through the hole is

\[
k_{te} = \frac{\sigma_{\text{max}}}{\sigma_{\text{avg}}} = \frac{\sigma_{\text{max}}}{P/(w-d)} = 2 + \left(1 - \frac{d}{w}\right)^3.
\]  

(1)

The present author [4,5] proposed the first of a series of formulae for loaded holes, with the peak net-section stress expressed in terms of average net-section stress and average bearing stress. There is a slight difference between the formulae for individual coupons and for strips isolated mathematically from wide panels with regularly spaced holes in a single row perpendicular to the applied load. In their latest form [6], they are as follows.

The first such formula is for an unloaded hole in a unit strip of width \( w \) isolated mathematically from a very wide panel. It is similar to Equation (1), but has a different limit as the hole diameter \( d \) approaches the strip width \( w \) because of the absence of bending along a line of symmetry at the "edge" of the strip.

\[
k_{te} = \frac{\sigma_{\text{max}}}{\sigma_{\text{avg}}} = \frac{\sigma_{\text{max}}}{P/(w-d)} = 1 + 2\left(1 - \frac{d}{w}\right)^{1.5}.
\]  

(2)

For a finite-width strip of width \( w \), centrally loaded by a bearing load \( P \), with the center of the hole, of diameter \( d \), at a distance \( e \) from the edge (end) of the laminate,

\[
k_{te} = \frac{w}{d} + \frac{d}{w} + 0.5\left(1 - \frac{d}{w}\right)\Theta, \quad \approx \frac{w}{d} + \frac{d}{w},
\]  

(3)

in which

\[
\Theta = \left(\frac{w}{e} - 1\right) \quad \text{for} \quad \frac{e}{w} \leq 1,
\]  

(4)

\[
\Theta = 0 \quad \text{for} \quad \frac{e}{w} \geq 1.
\]  

(5)

For a row of loaded holes in an infinite panel, at a pitch \( p \),

\[
k_{te} = \frac{p}{d} + 0.5\left(1 - \frac{d}{p}\right)\Theta, \quad \approx \frac{p}{d},
\]  

(6)

in which

\[
\Theta = \left(\frac{p}{e} - 1\right) \quad \text{for} \quad \frac{e}{p} \leq 1,
\]  

(7)

\[
\Theta = 0 \quad \text{for} \quad \frac{e}{p} \geq 1.
\]  

(8)

These formulae enabled the first predictions to be made of optimum joint proportions, in the form of ideal \( d/w \) ratios, first for single-row joints and then for multi-row joints. Surprisingly, to this day, the significance of this concept has apparently not been widely
appreciated, since so many design procedures for bolted composite joints are still limited to considerations of net-section strength and bearing strength – as if they really were independent quantities.

Equations (3) and (6) can be re-arranged to relate the peak stress on the net section to the average bearing stress instead of the average net section stress, using the relationship

\[ \sigma_{brg} = \sigma_{avg} \times \left( \frac{w - d}{d} \right) \]
whence \[ k_{tb} = \frac{k_{te}}{\left( \frac{w}{d} - 1 \right)} \]  

where \( k_{tb} \) is the ratio of peak tension stress alongside the hole to the average bearing stress. According to equation (3), therefore,

\[ k_{tb} = \left[ 1 + \left( \frac{d}{w} \right)^2 \right] for \quad e \geq w \]

This last formula is consistent with the wide-plate limit of unity as \( d/w \rightarrow 0 \), expressed in Fig. 3, with the ratio asymptoting to infinity as \( d \rightarrow w \).

Equations (3) and (6) can also be re-arranged as formulae to express not the peak stress, but the net-section strength. From these relations, it becomes possible to identify optimum \( d/w \) and to a lesser extent \( e/w \), values to maximize the net-section strength. The effect of the \( d/w \) ratio is found to be dominant, with that of \( e/w \) of significance only when it is too small. The joint efficiency is defined here by

\[ \eta = \frac{\text{net-section strength}}{\text{gross-section strength}} = \frac{1 - \left( \frac{d}{w} \right)}{k_{te}} \]  

In the case of the finite-width strip with a single fastener, Equations (11) and (3) would then predict that, for large edge distances \( e \),

\[ \eta = \left( \frac{d}{w} \right) \left[ \frac{1 - \left( \frac{d}{w} \right)}{1 + \left( \frac{d}{w} \right)^2} \right] for \quad \frac{e}{w} \geq 1 \]

This joint efficiency is zero both for \( d/w = 0 \), for which the bearing stress would be infinite, and at \( d/w = 1 \), for which there is no net section left, with a maximum efficiency barely greater than 20 percent at a \( d/w \) ratio of about 0.4. This formula is plotted in Fig. 4.

This very low maximum-possible structural efficiency for single-row bolted joints in brittle linearly elastic materials is not very encouraging for the prospects of bolted composite joints. However, Fig. 4 also includes a typical efficiency curve for real fiber-polymer composites, with a maximum joint efficiency nearly twice as high – with a more
practical \( w/d \) ratio as well. The reason for this difference between real fiber-polymer composites and the idealized linearly elastic model will be explained shortly. Fig. 4 also indicates the challenge faced by composites when competing against ductile aluminum alloys and the like, with static structural efficiencies twice as high again. However, that attribute must be balanced against the superior fatigue resistance of typical fiber-polymer composites.

![Graph showing joint structural efficiency](image)

**FIG. 4** - Relative efficiencies of bolted joints in ductile, fibrous composite, and brittle materials

**Accounting for the Post-Linear Behavior of Bolted Composite Joints**

The difference between the observed composite behavior in Fig. 4 and the linearly elastic predictions is not the result of material non-linear behavior of either the fibers or the resin matrix. It is the result of local non-catastrophic failures of one or both of the matrix and fiber constituents and of separations between the two. The so-called “composite material” is, by that stage, no longer a continuum. This, obviously, complicates both closed-form analyses and finite-element models considerably. So far, to the best of the author’s knowledge, most such analyses are either empirical curve fits to a multitude of test data, which are limited in applicability to the point designs those data characterized. The others are modifications of linearly elastic solutions achieved by incorporating some empirically determined correlation factor of far greater applicability. But even the latter approaches have limits beyond which the methods break down. Two of these latter methods are discussed here.

\[
k_{tc} - 1 = C(k_{te} - 1) ,
\]

(13)
Here, $C$ is the empirical correlation coefficient determined from many tests, of different geometries, but all failing by tension-through-the-hole. Fig. 5 explains the meaning of this formula and the limits on its applicability. The effective stress concentration factor at failure in the composite laminate is determined from the measured joint strength via the standard formula

$$k_{tc} = \frac{F_{tu}(w-d)}{P},$$

in which $P$ is the failing load and $F_{tu}$ the unnotched ultimate tensile strength of the laminate. By definition, this formula is inapplicable to bearing failures, since the net section would not have failed. It is equally inapplicable to other modes of possible failure, but it is the formula for all of the highest strengths that can be achieved.

**FIG. 5 – Relation between stress concentration factors observed at failure of composite laminates and predicted for perfectly elastic isotropic materials**

With these formulae, together with the bearing strength cutoff, it now becomes possible to even predict the behavior of each fastener in a multi-row joint, without needing to test such more complex joints to generate data. This capability was verified experimentally [7,8].

The first, developed by the author under contract to NASA Langley [4,5], involves an empirically calibrated stress-concentration relief factor, which has the effect of increasing the joint efficiency defined by Equation (11) as the effective stress concentration is reduced. The manner in which this relief is characterized is via Equation (13), which is valid only for the many situations in which the failure mode is tension-through-the-hole. It is not valid for bearing failures, which are associated with higher values of $k_{tc}$, or for shear-out or cleavage failures which, if they occur at all, will precede the tension failures.

Fig. 6 presents an actual set of test data [4] on bolted joints of many geometries, from which it was deduced that the value of the coefficient $C$ in Equation (13) was 0.25 for quarter-inch bolts in one particular quasi-isotropic laminate. The same value was also established for two different quasi-isotropic carbon-epoxy laminates.
FIG. 6 - Actual test data used to establish empirical stress-concentration relief factor for quarter-inch bolt holes in carbon-epoxy laminates

It may seem that this method could not be applied to non-quasi-isotropic laminates. Yet tests of bolted joints in orthotropic laminates suggest that this is not so, at least for tension-through-the-hole failures. The horizontal axes in Figs. 5 and 6 refer to isotropic materials. The stress concentrations would be higher in bolted joints of the same geometries but in orthotropic laminates. If this effect of orthotropy could be represented as a single proportionality factor for each fiber pattern, the relationship in Equation (13) could be applied to orthotropic fiber patterns equally well as with isotropic laminates. Changes in slope of the correlating line, via adjustments of the value of C would be equivalent to expansions of the horizontal axes.

So far, the efforts to see if all such test data could be reduced to a single universal relationship via an orthotropic reference have not succeeded. Nevertheless, this method has been applied effectively in terms of different factors C for different degrees of orthotropy (different fiber patterns) via the formula

$$C \approx \frac{\text{Percentage of } 0^\circ \text{ plies}}{100}.$$  \hspace{1cm} (15)

This increase in applicability of the method is recorded in Fig. 7.

The linear relationship for fiber patterns containing no more than 50-percent $0^\circ$ plies loses its applicability for more orthotropic fiber patterns because the mode of failure changes from tension to shear-out. However, it does appear to be reasonable for the close-to-isotropic laminates in which the strongest bolted joints are made. The problem with the laminates more rich in $0^\circ$ plies than about 25 percent is that the effective stress-concentration factors at the bolt holes increase in intensity almost as fast as the unnotched laminate strengths rise.
It must also be remembered that the assessment here refers to unidirectional loads. Most real structures are also subjected to loads in different directions. For aircraft wing skins, for example, there is a need for a substantial number of ±45° fibers to resist torsion loads. According to the author's Ten-Percent Rule [9-11] for predicting the strength of unnotched laminates, unidirectional plies are so highly orthotropic that the transverse strength and stiffness is only about one tenth as high as for the longitudinal properties. Consequently, fibers are needed in each direction for which there is some applied load. Indeed, unlike the case of homogeneous isotropic metals, the need for fibers in multiple directions remains even when the various loads are not applied concurrently. In a metallic wing, the same skin that resists bending loads under one situation can resist torsion loads at some other time.

All the data condensed into Fig. 7 apply to a single fastener diameter, 0.25 inch. There is also an absolute size effect; the joint strengths decrease as the fastener size increases. An empirical formula proposed by the author for quasi-isotropic laminates is

$$C = 1 - e^{-d}, \quad \text{for quasi-isotropic laminates}, \quad (16)$$

in which the fastener diameter $d$ is in inches. This formula is plotted in Fig. 8.

This formula asymptotes towards an absence of stress-concentration relief for very large holes, in conjunction with a complete absence of stress-concentration for extremely small holes. So far the efforts to find an equivalent formula to account for hole-size effects in orthotropic laminates has not succeeded. Equation (16) has all of the correct trends when multiplied by the coefficient $C$ in Equation (15), but some inconsistencies appear when several different levels of orthotropy are plotted simultaneously. Consequently, the author has yet to propose formulae sufficiently comprehensive to cover all combinations of hole size and fiber patterns. His methods are, therefore, incomplete, in the general case, even though they appear to work well for the quasi-isotropic fiber patterns associated with most of the best bolted composite joints.
The other widely used method of analyzing bolted composite joints was developed at about the same time as the author's method (above) by colleagues in the former McDonnell Aircraft Company (now Boeing St. Louis). It is the key to the BJSFM method [12,13] in which supposedly universal characteristic dimensions were used to predict failure not where it actually happened but at a location nearby, as indicated in Fig. 9.

This method, also, is not of universal applicability. Nevertheless, it did prove to be very useful in the sense that many of the bolted joints in the composite structures on the F/A-18 and AV-8B aircraft were analyzed using a single offset distance of 0.020 inch (0.5 mm). From the start, this single value accounted for a range of bolt diameters and
fiber patterns, although finite-width effects were not introduced until later. However, others without access to the McDonnell test data have tried to misapply this method far beyond its limited range of applicability. A comparison between the two methods is included in Ref. [14].

Note that the “characteristic dimension” needed to reconcile test data and predicted strengths varies around the circumference of the hole. The different locations are primarily a function of fiber pattern, but can sometimes change with geometry as well.

The BJSFM worked well when applied by the McDonnell engineers only because all their applications involved a low bearing stress, in the preferred design region shown in Fig. 1. Their coupon testing revealed that very different offset dimensions would have been needed for bearing failures, or of the $\pm 45^\circ$ plies instead of the $0^\circ$ plies. Despite assertions by the originator of this technique, Ralph Nuismer [15], that the characteristic offset was merely an empirical correction, like the author’s C factor, some researchers have bestowed upon the offset distance the title of material “property”. No matter how many textbooks refer to it as such, the characteristic offset dimension is NOT a material property and should not be regarded as such. The BJSFM method does not assume that it is – only that it is constant for the critical tension-through-the-hole failure mode. Indeed, the McDonnell engineers introduced further simplifications to enhance the appeal of their method. For example, when this method is applied in the really useful form shown in Fig. 1, there is no indication that there is actually a family of such lines – one each for each $w/d$ ratio [6]. The typical chart actually used is for a $w/d$ ratio of 4, which is appropriate for both fastener seams along spars and for seams of fasteners in chord-wise splices. The simpler the method is, the more it is likely to be used successfully, provided that one remembers that no single offset dimension covers every bolted composite joint. The study in which some 125 different offset dimensions were deduced from 125 tests of different joint geometries and mixtures of bearing and bypass loads was really not a contribution to the greater use of this method in the actual design process. The existence of this method is reported here because it points the way to possible future analysis methods in which truly universal methods could be based on the very small number of actual intrinsic fiber and matrix properties.

**The Design and Analysis of Multi-Row Bolted Composite Joints**

The preceding discussions have focused on single-row bolted composite joints, but the analysis methods are equally applicable to multi-row joints, even though the point-design methods based exclusively on test would not – unless the specific geometry had already been tested. Multi-row joints are needed to raise the joint strengths above those achievable by optimal single-row joint geometries, even though the increase is surprisingly small because, otherwise, metallic structures would prove to be lighter than composite ones, unless the glass softening-strip technique [16] were employed and there were a willingness to make unrepairable composite structures with zero damage tolerance. (There are some applications for which this is appropriate because of the great weight savings and added performance that comes with it, but there are many others for which such extremes would not be tolerated.) What the analysis methods can do that the point-design techniques cannot is to make reliable projections beyond the data on which they were based. This has already been done in earlier documents. Better yet, some such projections have been validated by tests and by inference, the other projections made with the same methods and coefficients have also been validated. One seemingly surprising
result is that, for the optimum \( w/d \) ratio, two identical bolts in tandem create a joint only 10 percent stronger than one bolt on its own. This has been predicted by the present analysis method over 25 years ago, and confirmed by test numerous times. Even the very best multi-row joints have exceeded the strength of the optimal single-row joints by less than 20 percent. Greater improvements have been achieved only by starting from a far-from-optimal single-row joint. In the author’s opinion, the reason why such limits are not better known is that the concept of the joint efficiency chart, Fig. 4, has yet to be widely applied. Too many designers still think in terms of independent bearing and net-section tension strengths, because that is the way joints in metallic structures have been designed and analyzed since the infancy of the aviation industry.

Fig. 10 is a more comprehensive joint-efficiency chart, covering the whole range of bearing stresses from full load transfer in a single row of fasteners, at the bottom the set of curves, to the zero-bearing-stress unloaded holes defining the top curve. The carbon-epoxy material on which these calculations are based is AS-4/3501-6. Slightly higher strains to failure, but much the same structural efficiencies, would be predicted for the newer high-strain fibers like IM-6.

![Fig. 10](image)

**FIG. 10** - Influence of bolted joint design on structural efficiency of carbon-epoxy composite structures

Fig. 10 confirms the existence of a true optimal joint geometry at about \( w/d = 3 \) for single-row joints. The failure mode is by tension through the hole, at a strength higher than can be achieved by *any* geometry for which bearing failures can be achieved, whether in a single row or by multiple rows. This point is clarified in Fig. 11, which shows how the bolts in each row must be moved further apart, as additional rows are added, to avoid tension failures.
Fig. 10 makes it clear that, to attain a joint strength greater than for the best single-row joint, it is necessary to use multiple rows and that, in the most critical row, the fasteners must be moved further apart, i.e., the columns separated, at the same time as the bearing stress is reduced. Normally, these two requirements would be mutually exclusive but can be achieved by using tapered splice plates and varying the w/d ratio between the rows, in conjunction with the A4EJ computer code developed as part of the USAF research contract [17] to establish how much load is transferred by each bolt. The reliability of such predictions was confirmed by NASA-funded tests [8]. The joint geometry for the strongest joints is shown schematically in Fig. 12.

Fig. 11 – Multi-row bolted joints failing in bearing

The splice plates were tapered, and the skin wasn’t, because the bearing stress in the skin at the most critical outermost (first) rows at the right side of Fig. 12 could be minimized only by maximizing the load transferred through the last row in each skin, where there is no bypass load to be considered. The most critical location was
deliberately located at the thin end of the splice plates, where the first failure would be visible and would involve the smallest possible decrease in total load capacity. The splice plates were actually made from aluminum alloy so that the holes could be counterbored. Such recesses would cause delaminations in composite splice plates, which would need tapered washers instead. The bolt spacing was constant at all rows of the joint, with the smallest diameter (3/8" inch) for the first row, \( w/d = 5 \), an intermediate diameter (1/2 inch) for the next two rows, \( w/d = 4 \), and the largest bolts (5/8" inch) for the last row, \( w/d = 3 \). This level of complexity in the design is typical of what is needed to achieve the highest possible efficiency. The investigation included analyses of simpler geometries that were far weaker because too high a bearing stress was developed in the skin at the first row. Intermediate tests on uniformly thick splice plates confirmed how much less efficient they were.

There is a common misconception amongst aerospace designers that the metric for the most efficient splice design is the number of pounds of load transferred per pound of spliced weight. That is not true even for ductile metals, let alone brittle composites. The weight of the splice plates is so insignificant in comparison with that of the basic structure that, while the true goal is that of minimum TOTAL weight consistent with safety, maintainability, etc., a good approximate metric is minimizing the weight of the skins by maximizing their operating stress levels. In the case of composite structures, it is customary to design in terms of gross-section strain level. This is obviously satisfactory for a common fiber pattern, but it should be questioned whenever different patterns are involved. For example, the highest strains achievable are associated with 0° loads on a ±45° laminate, which is matrix dominated. Yet the modulus of a ±45° laminate is so low that the associated stress (and load carried) is therefore very low. Stress is actually a more reliable metric.

Physical Explanation of Sources of Enhanced Measured Strengths Beyond Predictions by Linear Elastic Analyses

As a prelude to a discussion of the possibilities of establishing bolted composite joint analysis capability with no limits on applicability, it is appropriate to explain the physical phenomena, or at least what is known of them, that new analysis tools will be required to account for. The key reason why existing analyses for composites have only limited capability is simply that real composites of materials are heterogeneous combinations of fibers and matrix, while the primary analyses can cope with only those aspects of behavior that can be approximated by the models that contain no greater detail than is permitted by analyses of homogeneous "materials". This is why the empirical modifications described above have been needed – and will continue to be until better methods are developed. (Nevertheless, it must be appreciated that the capabilities of the best current methods are better described as a situation in which good answers are available for many of the situations, with no acceptable answer for others, rather than that poor answers could be found for all problems.)

Typical of the effects of heterogeneity is the macro-level size effect whereby the strength of composite laminates with very large holes, loaded or not, is reliably estimated by perfectly elastic models at the homogenized level, while the predictions of strength for quarter-inch (6.35 mm) holes underestimated the measured strengths by about 50 percent. It is obvious to any physicist that the size of the hole cannot change the properties of either the fibers or the matrix. So where does the undeniable "size effect" come from?
The strongest clue comes from old tests by the former Lockheed Sunnyvale (now part of Lockheed Martin) in which the measured strengths of quarter-inch (6.35 mm) bolted joints at 350 °F (180 °C) showed strengths far lower than at room temperature [18]; indeed, the measured strengths at high temperature agreed with the predicted room-temperature strengths based on assumed homogenized material. All of these phenomena can be explained by the hypothesis that the added strengths occur only when the matrix fails before the fibers and the agreement with unmodified analyses is achieved only when the fibers fail first.

This dichotomy is easiest to explain in terms of the performance of the quarter-inch (6.35 mm) bolted joints at the two temperatures. Consider, first, the behavior at room temperature. In order to increase the remote load level, it is necessary that there be non-catastrophic local damage at the first location to reach critical conditions. For a quasi-isotropic laminate loaded in the 0° direction, the location is along the 90° axis through the hole. The most critically loaded fibers are those in the 0° direction. If these were to fail first, the failure process would be unstable, as it is at 350 °F (180 °C) so, by inference, the initial failures must be in the matrix or at the interface between the fibers and matrix. This was confirmed decades ago by dye penetrant inspections made at NASA Langley. As explained in Fig. 13, this initial damage, by locally isolating the 0° fibers in the critical location from either increases or decreases in load, enables the predicted very high stress to be averaged with adjacent lower stresses. This allows the remote stress to be increased, as shown. The process continues until the stresses in the undamaged matrix outside the separated region are reduced to a level at which they will no longer break. After that, the fibers fail, at a remote load level elevated appreciably above the predictions based on a model of an undamaged continuum.

![Stress-concentration relief in fibrous composite laminates by local matrix cracking](www.polycomposite.ir)
The event triggering the end of the load-redistribution process that increased the joint strength would appear to be associated with a reduction to sub-critical conditions in the matrix immediately outside the zone that was softened by separations between the constituents of the fiber-polymer composite. This is consistent with the thinking behind the characteristic-offset model of analysis whereby the homogenized material reaches critical conditions at some small distance remote from the originally predicted most critical location which, in reality, is still the location at which the final failure process commences. The problem is that traditional analyses cannot predict these actual conditions, at ultimate failure, because the "composite material" is no longer a continuum there, while the analysis model still is. Hence the need for some form of empirical modification factor like those described earlier.

The reason why this stress-concentration-relief phenomenon is not observed for the quarter-inch (6.35 mm) bolts at high temperature, for which the matrix is both softer and more ductile – and would therefore be predicted to be LESS likely to fail than at room temperature, when it would be stiffer and more brittle – is that increasing the test temperature completely eliminated the massive residual thermal stresses in the matrix that existed at room temperature. Most of these thermal stresses were excluded from the analysis by the artificial homogenization of the fibers and matrix into an "equivalent" anisotropic solid, at the ply level (unidirectional or bi-directional). Failure of the matrix at room temperature is actually the result of the combination of mechanical and thermal stresses. This is one of the key errors in the traditional modeling of composites as homogeneous anisotropic solids. They are not! The belief that any related errors are compensated for because these effects were automatically present in any such tests used to establish lamina properties is invalid because the effect of the omitted influences is equivalent to some offset of the origin for the stress-free state, and not a proportional change in reference properties. The reason why the bolted composite joint performance at high temperature is akin to that of perfectly brittle materials, when the matrix would be expected to be at its softest, is simply that critical conditions are then developed in the fibers BEFORE the state of deformation in the matrix becomes critical. The linear behavior, at the macro level, of the composite material at failure is simply a consequence of the linear elastic failure of the fibers to failure. There is no mystery to explain, unless one persists in thinking of composite materials as homogeneous continua instead of as discrete fiber and matrix constituents.

Having explained the anomalies associated with test temperature, it remains to use the very same model to explain the equivalent anomaly associated with hole size. This may be more difficult, since it requires thinking in terms of strain gradients that do not exist in any of the standard uniform test coupons used to validate both realistic and implausible models of "composite materials". Consider two supposedly mathematically similar (proportional) models of laminates made from the same plies of composite materials arranged in the same pattern. One has a hole one quarter of an inch (6.35 mm) in diameter, while the other laminate, 24 times as long, as wide, and as thick, has a 6-inch (12.7 cm) hole. To complete the similarity, further assume that the ply thicknesses have been scaled as well, so that there is exactly the same number of interfaces through the thickness at which the adjacent fibers change direction. One could even scale the thicknesses of the interfaces but, since they are omitted from traditional analyses, this would have no effect. These coupons are to be as mathematically similar as possible, to
ensure that there will be no argument that the homogenized level of analyses, whether by closed-form or finite-element analysis, will fail to predict any difference between the two test coupons. It does not matter whether the widths are finite, or effectively infinite, provided that they are the same in each case. Based on past experience, only the analyses would be expected to be the same; the laminate with the small hole would be expected to be far stronger. But why? The true material properties do not change with hole size, or any other aspect of joint geometry, so why should the apparent macro-level properties change, particularly when such effort was devoted to creating mathematically equivalent problems? In truth, the only thing not to have been scaled is the fiber diameter. (There is a direct analogy of this problem with adhesively bonded joints in which every dimension is scaled except for the thickness of the adhesive layer; the joint strength decreases as the size is increased – both by analysis and test.) Is the non-scalable fiber diameter responsible for the difference in strengths between composite laminates with holes of different sizes, or is there some other explanation? And how can this effect be accounted for in analysis?

Explaining this phenomenon is complicated because, in reality, the ply thicknesses could not be scaled without causing edge delaminations in the larger coupon, so the greatest similarity that could be achieved would have common effective ply thicknesses. At that point, it becomes immaterial if the total thickness is scaled or not, provided that every detail in the lay-up sequence is matched.

An assessment of what differences might exist between the two laminates of different hole sizes, with every other equivalence as possible maintained will show that the distance over which the final 10 percent of the load in the most critically loaded fibers is built up is 24 times shorter for the quarter-inch (6.35 mm) hole than for the 6-inch (12.7 cm) hole. In other words, while the analyses will show the same state of deformation in the fibers, and in the matrix, at every point of equivalence, there actually must be a difference that is not accounted for at this level of analysis. If this difference could be accounted for, in terms of true constituent properties, the matrix could be predicted to fail first in the laminate with the smaller hole and the fibers to fail first around the larger hole. Clearly, this will require some level of analysis beyond the standards employed today. The question of what might work, or at least a minimum improvement that might be needed, will be resumed after an explanation of the first science-based model for predicting the strength of fiber-polymer composites that has ever been developed.

The Sift Failure Model for Fiber-Polymer Composites

The merger of The Boeing Company and the former McDonnell Douglas Corporation in 1997 brought together two researchers who, between them, held the keys to the development of what the author considers to be an original physics-based model with which to characterize the strength of fiber-polymer composites. Jon Gosse, in Seattle, had modeled the failure of the resin, and adhesive layers, in terms of the first strain invariant \( J_1 \), being the sum of the three orthogonal strains. In Long Beach, Hart-Smith, if we pretend that the fiber diameter could be scaled as well, the load in the fiber would be increased in proportion to the square of fiber diameter. This would be changed, along the length of the fiber, over a length proportional to the hole diameter, through an area further increased in proportion to the greater circumference of the larger fibers. In other words, similitude could be satisfied only if every dimension were scaled, including fiber diameter and interfacial thickness. But then it would be, and this discrepancy between test and theory would disappear.
the present author, had developed a model for failure of carbon fibers based on a shear mechanism, that is now recognized as being defined by a further invariant, the von Mises' equivalent shear strain. Their collaboration has produced a model based on universal true material properties at the fiber and matrix constituent level, each of which can fail only by dilatation or distortion, as described in Fig. 14 for failure of the matrix. The failure envelope for fiber failures is of the same form, but the brittle-fracture failure possibility is apparently further removed from the origin. No other mechanisms are possible, other than for interfacial failures. (The intense contraction of the resin matrix cured at high temperature as it cools down and is resisted by stiff fibers seems to have effectively suppressed this possible mechanism.

This so-called Strain Invariant Failure Theory for fiber-polymer composites is described in Ref.'s [19] to [21]. A most significant feature of the model is that the intralaminar residual thermal stresses are accounted for. It has been their omission from all previous models, other than in terms of micromechanics, that has doomed them to be incapable of properly characterizing most matrix failures. The development of the theory is now complete for all uniform states of deformation, as in unnotched laminates and structures. Some progress has also been made in matching the measured strengths of laminates with holes in, under both tensile and compressive loads, but it is too early to claim that this aspect of the model is anywhere near complete.

The SIFT model is more than a mere physical description of the phenomena. Gosse and a colleague in Boeing St. Louis, Jeff Wollschlager, have developed different, and therefore confirmatory, finite-element programs to enable the model to be applied, using different kinds of elements. Without these codes, the capabilities of this model could not have been demonstrated. The basis of this new analysis method is explained in Figs. 15

![FIG. 14 - SIFT matrix failure criteria for fiber-polymer composites](image-url)
It is traditional that the primary large-scale analyses of composite structures, in terms of finite-element models, be first conducted at the homogenized laminate level, to reduce the complexity of the analysis to a manageable level. At this level of analysis, only deflections can be predicted, not the internal stresses. This is then followed by an equally traditional decomposition of the laminate-level analysis into and equivalent ply-by-ply (lamina) level analysis, in the belief that it was then possible to apply failure criteria and predict the strength of the structure. Where the SIFT approach breaks with tradition is that the decomposition process is continued one step further, into the stresses and strains in the fiber and matrix constituents. Only then does it become possible to reliably predict...
matrix failures under arbitrary combinations of biaxial loads. Stopping the process one step short, as is customary, restricts the predictions of strength to situations in which the fibers fail first and, even then, it is necessary to avoid most of the published failure models. The maximum-strain model for glass-fiber-reinforced laminates and the truncated maximum-strain failure model for carbon-fiber-reinforced laminates are the best available for use at the lamina level.

Fig. 15 also explains the importance of the level at which the various reference properties are established. Too high, or too low, a level and the measurements will not be correlatable with the intrinsic properties needed to predict situations for which no tests have been performed.

The decomposition of lamina-level stresses and strains into those in the fiber and matrix constituents requires the use of micromechanical models like that shown in Fig. 16. These models take a lot of work to prepare, all by Gosse or Wollschlager, and it is necessary to validate that the grid size is such that the predictions have converged. But, once prepared, they are universal in applicability and do not need to be repeated for each new structure being analyzed.

These models account separately, for the thermal and mechanical contributions to the strains in the constituents, with respect to the stress-free states to which each would contract were it not for the enforced compatibility of deformations (except at free edges). Each such model represents the conditions that prevail in the interior of laminates as well as on stress-free surfaces. The contours shown in Fig. 16 are for constant $J_1$ strain invariants in the matrix. These, and the other three invariants are compared against the four failure criteria. Experience has identified preferred failure locations, so that, in practice, the search for the critical conditions is not as onerous as it might seem.

---

3 The models do not yet account for chemical shrinkage, which is known to be significant for thick filament-wound structures, or for swelling from moisture absorption, which may or may not be important. In any event, the same absorbed moisture that might alleviate some thermal stresses at room temperature, could aggravate them during any rapid decrease in temperature. Absorbed moisture might even turn into steam during rapid heating or ice during rapid freezing. But these additional effects could be added to the model, if one knew how to, without invalidating the treatment of the thermal effects, which are now known to be necessary.

4 Although failure of fibers by brittle fracture, and by compressive instability are theoretically possible, and provision is made for them in the coding, none of the huge number of multi-directional test specimens and subcomponents has yet indicated any laminate strengths associated with these mechanisms for either AS-4 or IM-6 carbon fibers. So far, all fiber failures have been governed by distortion, which is the model proposed by the author, and independently by researchers at DERA, in the UK, more than 20 years ago. This mode of failure has recently been confirmed experimentally by Gosse and Christensen [20] by tensile tests of ±10° laminates in which the fibers broke, and the load dropped, at a strain level far lower than for 0° tests, while the tough matrix remained intact. All fibers had tensile axial loads, but were also subjected to intense transverse compression from the very high Poisson's ratio of this laminate. This time, however, there is no opportunity for those who insist that carbon fibers cannot possibly fail by shear to blame the reduced failure strain on premature axial compression in half of the fibers, as there was for the earlier demonstration of this interaction between longitudinal and transverse stresses on the fibers for pure shear tests of a ±45° carbon-epoxy laminate, almost 30 years ago. (Significantly, those results still seem to be the highest such strengths ever measured for such a laminate.) Also, this time, the broken fibers remained visible in the undamaged matrix, with none of the customary splintering. So, it is no longer possible to question the fact that transverse stresses of the opposite sign do decrease the longitudinal fiber strain at failure, in a manner that only a distortional failure mechanism appears to be able to explain.
No other representation of fiber-polymer composites has even come close to matching the demonstrated capabilities of the SIFT model, and none can match the features incorporated in this model. This is not to say that no prior models have ever contributed to the design and analysis of good composite structures. Quite the opposite—but those empirical models that have been useful have been unable to solve all of the problems, and have been capable of being misapplied to make faulty predictions. There really is a desperate need for a fiber-polymer composite failure model that is reliable in the hands of novices and experts alike. The originators of the SIFT model are well aware that much more remains to be done in regard to non-uniform deformation fields before that day has arrived.

**Possible Application of the Sift Model to Bolted Joints in Fiber-Polymer Composite Laminates**

The following explanation of load transfer and potential failure sites in bolted composite joints is formulated around the popular family of orientations $0^\circ$, $\pm45^\circ$, $90^\circ$, but the same logic could be developed for any other orientations, too. The necessary features for this discussion are shown in Fig. 17. Particular attention must be paid to the interruptions to the $0^\circ$ fibers, parallel to the prescribed load direction, by the loaded or unloaded hole. The arguments are presented first in relation to the non-bearing side of the hole and therefore apply equally to loaded and unloaded holes.

**FIG. 17 – Basic references for modeling of load transfer around bolt hole**

The load in the $0^\circ$ fibers that are interrupted by the hole has only two alternative load paths available. It can be transferred sideways into the adjacent $0^\circ$ fibers in the same ply, at least until a tangential vertical split develops in the matrix, starting just inside the first continuous $0^\circ$ fiber outside the hole. Such splits have been observed often in laminates, particularly in $0^\circ/90^\circ$ fiberglass-epoxy laminates. The splits stop spreading when they are...
so many hole diameters away from the hole that the stress field is close to uniform. Such splits effectively eliminate the stress concentration at the holes, so that the failing load is very close to the unnotched strength of the net section. However, because the matrix provides so much softer a load path than the fibers at ±45°, this interrupted load is more likely to be sheared above or below the stack of 0° fibers into whichever of the ±45° fiber directions would carry the load out past the hole, the 45° fibers in the other direction terminate against the hole and cannot contribute to such load transfer. The limit to the amount of load that could be sheared sideways in this manner is defined by the strength of matrix interface at the change in fiber direction. This would obviously fail early whenever too many parallel plies were blocked together; this is a common design error made in the name of reducing cost by designers who think of composites as homogeneous solids instead of in terms of their discrete constituents. Conversely, if the plies were too thin, like the 0.025 mm (0.001-inch) layers sometimes used in stiffness critical space structures to prevent delaminations in unnotched composite laminates subject to the intense thermal cycles of deep space, the interface may never fail. Tests of bolted joints in such laminates have confirmed the low strengths achieved whenever the matrix is prevented from failing first. But neither of these possible matrix failures would necessarily cause immediate failures of the continuous 0° fibers just outside the hole. Consequently, after either or both such matrix failures, the remote 0° stress at the top of Fig. 17 could be increased.

The next possible failure, if there were too few ±45° fibers in proportion to the number of 0° fibers, would be in the continuous ±45° fibers tangential to the hole. However, in the case of unloaded holes, the macro-level analysis of stresses around a circular hole in an isotropic plate would identify virtually no tangential stresses at the ±45° orientations. Only on the back side of the hole, in response to the bearing load on a loaded bolt, would serious loads develop in the fibers in those orientations. Their failure under bearing loads is to be expected, has been predicted by the BJSFM analyses, and has been observed by test when there have been too few ±45° fibers, something that happens frequently when the fiber pattern is “optimized” before the bolted joints are designed. Failure of these ±45° fibers on the back side of a loaded bolt could be instantly catastrophic, but it is not clear that this always needs to be so.

The 90° fibers are expected to be most highly loaded directly above or below the hole in Fig. 17, but the stresses will be small and compressive for unloaded holes and at the top of loaded bolt holes. Cleavage failures, like those shown in Fig. 2 result whenever laminates contain too few 90° fibers for loaded fasteners.

It is for the reasons just explained that there are preferred fiber patterns for bolted or riveted composite laminates. These are identified in Fig. 18; they seem to contradict the popular notion of optimizing fiber patterns in accordance with the basic structure away from the fasteners — and they do. It is most important, when designing structures, to design the most critical locations first and to fill in the gaps in between later. If the as-manufactured design does not require adherence to this advice, the ability to repair the structure after it was damaged in service, or even before delivery, will usually be too expensive. The consequence of this will result in the structure being scrapped and replaced.

Most of the failures discussed above would be identified by any sufficiently detailed finite-element that checked against the SIFT failure criteria for failure, at least for the
first exceedence of one of the four invariants. The ones of real interest here are those that
do not result in immediate catastrophic failure, but permit local load redistribution to
accompany an increase in the remote load level. These require that the initial damage be
localized, so that the laminate is behaving like a continuum *outside* that region. The
same concept that enabled the characteristic-offset method to be applied might be adapted
to the new SIFT failure model, based on the understanding generated above.

If, for whatever reason, the fibers in some direction are predicted to be critically
loaded *before* there is any indication of having exceeded the capabilities of the matrix,
one could reasonably assume an instantaneously catastrophic failure. Of course, it must
be understood that these predictions have yet to be made and therefore cannot be
compared with existing test data. The predictions would need to agree with both loaded
and unloaded hole data, over the full range of hole sizes. However, the author is
confident that this portion of the analysis method *will* be confirmed, on the basis of the
test data and other analyses he is already aware of. One caveat is needed, however, in
regard to fiber failures. It is possible to create fiber patterns around bolt holes that are so
far from the real optimum fiber pattern that fibers in the ±45° direction, for example,
could fail first under a 0° load, if there were too few of them, well before the fibers in the
0° direction failed, particularly if there were too many of them.

The real challenge concerns situations in which the matrix is predicted to fail by the
standard SIFT models *before* any fibers become critical. The standard SIFT models
appear to have no capability to respond to what has been described earlier as the hole-size
effect, because of the problems with scaling fiber diameter. Proving that it works will
require validation for both very small and very large holes, with a net-section strength
asymptotically approaching the unnotched strength for very small holes and approaching

---

**FIG. 18** – *Preferred fiber patterns for structural fiber-polymer composite laminates*

If, for whatever reason, the fibers in some direction are predicted to be critically
loaded *before* there is any indication of having exceeded the capabilities of the matrix,
one could reasonably assume an instantaneously catastrophic failure. Of course, it must
be understood that these predictions have yet to be made and therefore cannot be
compared with existing test data. The predictions would need to agree with both loaded
and unloaded hole data, over the full range of hole sizes. However, the author is
confident that this portion of the analysis method *will* be confirmed, on the basis of the
test data and other analyses he is already aware of. One caveat is needed, however, in
regard to fiber failures. It is possible to create fiber patterns around bolt holes that are so
far from the real optimum fiber pattern that fibers in the ±45° direction, for example,
could fail first under a 0° load, if there were too few of them, well before the fibers in the
0° direction failed, particularly if there were too many of them.

The real challenge concerns situations in which the matrix is predicted to fail by the
standard SIFT models *before* any fibers become critical. The standard SIFT models
appear to have no capability to respond to what has been described earlier as the hole-size
effect, because of the problems with scaling fiber diameter. Proving that it works will
require validation for both very small and very large holes, with a net-section strength
asymptotically approaching the unnotched strength for very small holes and approaching
the homogeneous solution with no stress-concentration relief at all for very large holes.\(^5\)

The existing method for unnotched laminates will, however, identify both where the
matrix is predicted to fail and which fiber direction, and where, is predicted to be the
most severely loaded and, by inference, the first to fail when the load was increased.
This is important, since, as his colleague has pointed out to the author, any analysis
method to be used in production, as distinct from research, must be capable of being
applied blindly (with no special skills) as well as accurately.

The following proposal is made for analyzing bolted composite joints using the SIFT
model in those situations in which the matrix is predicted to fail first. The process starts
with the conventional homogeneous finite-element model. The remote load level should
be scaled to make the most critical fiber or resin location just on the point of failure
according to the existing SIFT post-processing via the micromechanical modification
factors to relate the strains in the laminae to those in the fibers and matrix constituents.
Both mechanical and thermal loads should be included, but experience has shown that the
thermal effect on the fibers is insignificant for fibers embedded in the typically soft resin
matrices. It is necessary to differentiate between cases in which the interfaces between
changes in fiber direction are predicted to be critical and those when they are not. It is
also necessary to distinguish between failures predicted to occur by tension around the
perimeter of the hole and predicted compressive failures in the region of high bearing
stresses at loaded bolt holes. In this last case, there is a known sensitivity to through-the-
thickness compression resulting from clamp-up between clevis plates or large washers
that can enhance the bearing strength far beyond the uniaxial longitudinal compression
strength of the fibers. Typical bearing strengths associated with protruding-head
fasteners can be almost twice the strength developed by a simple pin with no through-the-
thickness constraint. Deliberately over-torquing the bolts can raise the strength by a
further factor of two. Tests on unequal tri-axial compression of unidirectional 0° carbon-
epoxy rods have shown an enhanced strength consistent with shear failures of the fibers
[22], in which the enhanced strength could not be ascribed to friction that might occur in
mechanically fastened joints. Indeed, there are many cases in which it has been found by
tests at NASA Langley [23] that through-the-thickness clamp-up can be so effective that
the failure site is removed from a location adjacent to the fastener to a different location
outside the washer, as shown in Fig. 19. Disassembly after the tests then showed
absolutely no damage in the area where bearing failures would otherwise have been
anticipated. It should be clear that there are many possibilities for bolted composite
joints to fail at loads far higher than those predicted by linear elastic analysis of
homogeneous solids. The response to each of these, in terms of selective post-processing
of the primary analysis, will now be explained.

\(^5\) At the time of writing, the SIFT method has been shown to match a considerable amount of open-hole
data for 6.35 and 12.7 mm (0.25- and 0.50-inch) holes in a variety of fiber patterns, by use of what are
called damage functionals. No modification of the finite-element analysis has been made to allow for
possible progressive damage, because of the difficulties for re-establishing convergence of the model after
each such modification. But the very difficult finite-element models for very small and very large holes in
very wide panels have yet to be constructed. So the question of complete validation is not yet resolved.
The author has no doubt that the SIFT team will explain all of the data on bolted composite joints;
otherwise he would not be suggesting in public a technique to be evaluated without first confirming that it
works. His confidence is based on the relatively enormous progress already made and the historic
willingness to add new features as they prove to be necessary.
At this point, it is necessary to introduce some practical constraints on the analysis procedures to be used in production work, as distinct from research and development. If the first critical condition found is not instantaneously catastrophic, one could remesh the problem after accounting for any sub-critical damage that might modify the internal load paths. This would be followed by a complete re-analysis at the laminate level. The answer would be meaningless in the case of the first failure being a delamination, because this can be modeled only in a ply-by-ply model. So the complexity of the primary analysis would thereby be raised considerably. Assuming that these obstacles were overcome, one would then identify what failed next. If that, too, were not instantaneously catastrophic, further modifications could be made to the primary analysis model to account for a second source of modification in the load transfer between and within the plies. The process would continue to be repeated until a real catastrophe was predicted. It could, and with the correct constituent properties should, be a very accurate prediction. If there were sufficient non-catastrophic initial failures, the process should even give guidance to selecting a better and stronger laminate. But this process would be time consuming, expensive, and would require an expert in both finite-element analyses – to ensure that convergence has been achieved after each modification – and in composite material behavior at the constituent level – to know what modifications in include in the model at each stage. And, as his colleague, Jon Gosse, frequently reminds the author, this would mean that the analysis process would become so complicated that potential customers would choose to make structures from other materials that were easier to analyze. What Gosse advocates very strongly is simple upper and lower bounds on the true answer that can be established by post-processing alone, with no modification ever of the primary analysis. His concern is that the matrix inversions in the primary analysis will go unstable if there are progressive refinements of the large model to reflect any progression of damage. At this stage, it remains to be seen if this goal can always be satisfied, by having the bounds sufficiently close together. But, with no real experience...
to go on at this stage, the author will propose techniques that might satisfy such a goal. Even so, it must be acknowledged that more precise modified analyses by R&D engineers may be needed to identify which simplifications are acceptable for production use – and which ones aren't.

**Tension-Through-the-Hole Failures**

For well balanced laminates inside the area of preferred fiber patterns in Fig. 18, the failures finally observed after failure are seen to have initiated alongside the holes, along the 90° axis shown in Fig. 17. It has also been found, with only one exception, that the ultimate failure load was higher than is predicted by linear analysis of homogenized "equivalent" composite material. Careful examination has also revealed damage to the matrix that is presumably responsible for the enhanced strength of the laminates. The one exception was a series of tests by Lockheed Sunnyvale, at 350 °F (180 °C), in which it was found that there was no enhancement of the strength, presumably because the fibers failed first. If one were to plot the in-plane shear stresses, at the homogenized laminate level, along the 0° lines tangential to the holes shown in Fig. 17, it would be found that these stresses decay rapidly away from the hole. This has to be so because there can be no such stresses at that level remote from the holes whenever the remote stresses are uniform, as is usually the case. The integral of these shear stresses is mirrored in the 0° stresses found along a parallel axis just outside the hole. This represents the load transferred from the interrupted fibers into those that are continuous. (At this coarse level of analysis, it is not possible to distinguish between the shear stresses that are in the ±45° fibers that are not interrupted by the hole and the shear stresses in the matrix within the 0° plies, but it obvious that the greater stiffness of the 45° fibers will make that the preferred load path unless there are too few of them.) These two stress distributions are plotted in Fig. 20.

![Diagram](image)

**FIG. 20 – Relative severity of potential critical conditions in 0° fibers and associated in-plane shear loads**
It will be seen that it is theoretically possible for either stress field to become critical first, when the overall internal loads in the laminate have been decomposed into the individual contributions with in each ply, and further, to the fiber and matrix constituents within each ply.

(Woven layers must be separated into their two equivalent unidirectional plies if there is to be any differentiation between failures in the matrix and fibers.)

If the 0° fibers were predicted to become critical first, at this location, catastrophic failure would be expected to follow. However, if the matrix were predicted to fail first, before any of the fibers had reached a critical load, one would expect initial non-catastrophic failures to develop in the matrix. These could spread, as the load was increased, until the fibers became critical too. Such matrix failures would spread only so far as the matrix was predicted to be more critically loaded than the fibers. Normally, it would be arrested, as indicated in Fig. 20.

The proposal suggested here for estimating the load at which final failure would occur, without modifying the global analysis model, is as follows. If the 0° fibers are predicted to fail, under a 0° load, before the matrix becomes critical, there is no mechanism to allow any load redistribution. Therefore, such a prediction would be assumed to be an instant failure at ultimate load, with no preceding first-ply failure. The lower pair of curves in Fig. 20 signifies this condition. If the matrix were predicted to fail first, as indicated by the upper pair of curves, the load at which the matrix was first predicted to fail would customarily be identified as the first-ply failure, even though it should be noted that no ply has actually failed since the fibers that carry the load are still intact. (This is one of the chronic shortcomings of trying to predict failure of composite laminae with only homogenized models.) An obvious lower-bound estimate for the ultimate strength would be that achieved by simply scaling up this same state of deformation until the first 0° fibers are predicted to fail, alongside the hole. This takes no credit for any load redistribution to alleviate the most critical stress concentrations. A better estimate of the ultimate strength, from this same global analysis, would be found by increasing all of the remote loads proportionally and progressively by the factor $A/B$ shown in Fig. 20. The logic behind the justification for this estimate is as follows. If the matrix has failed throughout the range shown in Fig. 20, the load in the intact fibers within that region cannot be altered. If they hadn’t already broken, they could withstand more load. The factor cannot be allowed to be so high that the stresses and strains in the fibers ever exceed their values for unnotched laminates free from stress concentrations. The matrix failure could either be within the 0° plies or, more likely, in the interface between the 0° fibers and the +45° or -45° fibers from which the loads were being transferred. Interfacial delaminations can be modeled without knowledge of fiber diameter provided that each block of parallel fibers and the interface thickness and properties are right. The reason why this estimate, based only on modifying the micromechanical models to compensate for matrix failures, or over-riding predictions of such failures, is likely to be an under-estimate rather than an over-estimate, is that the average strain in the 0° fibers throughout the region of matrix damage would be predicted to exceed the failing strain level. Consequently, those fibers would either be less critically loaded, or the softened zone immediately adjacent to the critical location next to the hole would be larger than from this estimate. Fortunately, there are sufficient bolted
joint test data available to assess this hypothesis once the necessary finite-element models (local/global and micromechanical) have been created.

For quarter-inch bolts at room temperature, the measured bolted joint strengths are typically at least 50 percent higher than those predicted by linear analyses of homogenized equivalent composite material, so there are clearly significant load redistributions after initial failures with which to evaluate hypotheses with which to explain them. In particular, there is a well-established absolute size effect whereby the relief disappears for all very large holes and becomes greater and greater for small holes. There are trends for the hypotheses to explain as well as specific individual data.

The concept of being able to predict post-initial damage without having to reformulate the global analysis model to account for load redistribution relies on the fact that the initial damage is usually confined to such a small area that the load redistribution is so localized that its effects would not be seen in a global laminate-level analysis. This is certainly true of the micro-damage that frequently occurs around fasteners. Conversely, if the initial damage were not so localized, as sometimes happens in fiber patterns that should be avoided, it is likely that the approach Jon Gosse wishes to follow will not work. Even so, the decompositions of the coarse output that follow the global analysis preceding the SIFT assessment will probably reveal very widespread damage. This may be sufficient to discourage the use of such laminates in real structures, so the analysis may have achieved its primary mission – which is to improve the design – without actually needing to predict all of the strengths.

With some fiber patterns around bolt holes, the first critical location might not be in the $0^\circ$ fibers alongside the bolt hole, but in the $\pm45^\circ$ fibers perpendicular to the $\pm45^\circ$ axes in Fig. 17. This failure mode has been observed experimentally and revealed in many analyses of badly designed laminates at the former McDonnell Aircraft Company. If there simply are insufficient $\pm45^\circ$ fibers to transfer the load from the interrupted $0^\circ$ fibers or to withstand the hoop stresses developed around the back of a loaded fastener, they will become critical before the $0^\circ$ fibers. Their failure too, like the resin interfaces at changes in fiber directions, may not be instantaneously critical. However, there is less experimental evidence about how much such damage can be tolerated because it arises only in fiber patterns that are unsuitable for bolted joints or even large cutouts. What usually happens with laminates with a deficiency of fibers in one or more directions is that there is no way to break the fibers in the directions for which there is an excess. The next failure is often very long splitting between the $0^\circ$ fibers, tangential to the holes, extending over many hole diameters, as explained in Fig. 21. This limits the stress concentration in those fibers, but they still fail at a load level far less than they can develop in well-designed laminates. This kind of failure is prevalent with high concentrations of $0^\circ$ plies and is aggravated by the total absence of either $\pm45^\circ$ plies or $90^\circ$ plies.

Only application of different failure scenarios, using physics-based composite failure models will ever tell if the art of predicting such failures can ever progress beyond empirical correlation with test data. But at least the numerical experimentation can be far less expensive than laboratory experiments. Much the same comment can be made about laminates with too few $90^\circ$ fibers to withstand the hoop stresses developed around the back of bolt hole. Again, the $0^\circ$ fibers are incapable of being loaded to their true capacity (for $0^\circ$ loads). But even if it transpires that analyses of the kind proposed here never
work outside the realm of well-designed laminates, the ability to inexpensively identify inferior fiber patterns, in regard to both direction and stacking sequence, would be priceless.

- Shearout failures for large edge distances
- Characteristic neat parallel plug sheared out at low load
- Failure load insensitive to edge distance
- Prevalent failure mode for fiber patterns containing too many 0° plies

FIG. 21 – Shear-out failures for bolted joints in excessively orthotropic composite laminates

Bearing Failures

Even with good fiber patterns there are some joint geometries for which failure occurs away from the location at which tension-through-the-hole failures occur alongside the holes. In the case of loaded bolt holes, if the fastener spacing (w/d ratio as defined in Fig. 17) is high enough, failure will occur near the 0° axis in Fig. 17, on the side of high bearing pressure. The empirical analyses developed in Ref. [18] have been able to differentiate between tension and bearing failures, as a function of joint geometry, for many years now. These same simple analyses were also able to quantify how much stronger the tension-through-the-hole failures were than bearing failures, with very little test data after the basic model had once been validated. However, they are unable to quantify the beneficial effects of the through-the-thickness clamp-up shown in Fig. 19. The SIFT failure model may be able to, however, since the von Mises criteria for both the matrix and the fibers shows that tri-axial compression enhances the material strength. Indeed, if there is no difference at all between the strains along, transversely, and normal to the fiber axis, the strength is predicted to be infinite, just as it is for the resin matrix under equal tri-axial compression. The author is unaware of any other composite failure theory that could explain such phenomena.

It has been found that the bearing strength of laminates loaded by a pin with absolutely no constraint in the through-the-thickness direction is on the order of the axial compression strength of that laminate under a state of uniform deformation. Even fingertight nuts and bolts can almost double that strength, however, and additional torque is known to increase the strength even more, as shown in Fig. 22.
The author looks forward to the day when the SIFT failure model for fiber-polymer composites is applied to this class of problems, too. It has already successfully explained so much data relating to uniform strain fields and laminates with small holes loaded in both tension and compression that there is reason to be optimistic about further applications to non-uniform stress fields around loaded bolt holes.

Concluding Remarks

This paper has attempted to survey some of the useful empirical analysis and design techniques developed for mechanically fastened joints in fiber-polymer composite laminates and to give some indication of what improvements may be expected in future, based on this original physics-based composite failure model.

The contemporary analysis/design methods have proved to be both effective and capable of generalization far beyond the test data used to develop them. However, they have been limited in the sense that some form of empirical correction factor is needed for all such models. Linear closed-form and finite-element analyses are far too conservative for reasonable designs, even though they form the starting point for all of today’s analysis methods.

It needs to be stated that the characteristic dimension used in some of the bolted-joint analysis tools is not a material property, because there is no such thing as a composite material, only a composite of materials. Not all researchers and textbook authors have understood that it is simply an empirical correction factor of limited applicability, which changes with each mode of failure. Its usefulness will be diminished by the pretense that it has any greater significance.

There is no one empirical model that encompasses all of the capabilities of other models. Only a non-empirical model could offer that level of applicability.

One of the huge attributes of even approximate closed-form analyses is the opportunity to perform parametric studies and, in this case, to identify optimum joint proportions for maximizing the gross-section operating strain level, for instance. The C-factor method has permitted this for many years. It has proved to be a great asset that no amount of finite-element analyses will ever supplant.
The new SIFT (Strain Invariant Failure Theory) for fiber-polymer composites offers hope that bolted joints and other stress concentrations may some day be analyzed in terms of truly universal constituent properties. But even so, the practicalities of finite-element analyses of progressive damage may preclude the use of idealized step-by-step remodeling of the problems. We may still be limited to some level of approximation, but hopefully a lesser level than in the past. The more accurate the model, the less the need for empiricism. And, more importantly, the more accurate the model, the less will be the need for extensive test data before each model can be applied.

Perhaps the most powerful motivation for developing even better analysis tools for bolted composite joints than are used today is that design operating strain levels will then be less limited by the shortcomings of the theories, making composite structures more competitive with the equivalent metallic ones.  

References


---

6 No amount of better analysis will ever overcome the lack of ductility of fibrous composites or the discreteness of the fibers that inhibits the spreading of cracks perpendicular to the dominant load direction, but conservatisms introduced because of incomplete characterization of the properties could then be removed.


James R. Eisenmann¹ and Carl Q. Rousseau¹

IBOLT: A Composite Bolted Joint Static Strength Prediction Tool


ABSTRACT: The objective of this paper is to describe the bolted joint analysis method developed and used by LM Aeronautics (named IBOLT) including the theoretical basis, required input, and coupon-level strength-prediction validation. IBOLT performs a fracture-mechanics-based static strength prediction for a rectangular composite joint element subjected to any combination of biaxial membrane loads, shear loads, and an off-axis bolt load. Out-of-plane bending moments are also treated. Joint configuration effects (single- or double-shear) are modeled by a beam-on-elastic-foundation analysis to account for the thickness and stiffness of the joint members as well as the bolt bending and shear stiffness. Empirical equations are included to account for fastener head geometry, effects of filled versus open holes, fastener/hole clearance, and fastener torque. Required input, in addition to the usual geometry, loads, fastener, and substrate elastic properties, includes a variety of special laminate-level joint properties. Uni-axial notched, un-notched, and bolted-joint coupon tests are conducted to develop these properties. In addition to these uniaxial coupon tests, bearing/bypass tension and bearing/bypass compression tests have been performed to validate the accuracy of IBOLT strength predictions. An overview of these validation results is provided, along with a brief summary of IBOLT’s strengths and weaknesses as a practical bolted joint stress analysis tool.

KEYWORDS: composite material, failure, failure criteria, strength, structural requirements, failure modes, allowables, stress analysis, failure analysis, fracture mechanics

Introduction

Bolted joints are the most common form of joint in composite aircraft structure. Such joints are also often the limiting factor for structural efficiency and load-carrying capability. As such, composite bolted joint static stress analysis methods were the subject of much early analytical development in the field of advanced composite mechanics [1-3, 6, 8-10]. While Garbo and Ongowski’s [7] Bolted Joint Stress Field Model (BJSFM), and purely empirical bearing/by-pass interaction curve-fits [4] are widely used throughout the aerospace industry, a proprietary method generally referred to by one of its FORTRAN program names, IBOLT (Interactive BOLTed joint analysis), has always been the method of choice at what is now the Lockheed Martin Aeronautics’ Fort Worth facility.²

The objective of this paper is to describe the IBOLT analysis, including the

¹ Senior Staff Engineer, Lockheed Martin Aeronautics Company, P.O. Box 748, Fort Worth, TX 76101.
theoretical basis, required input, and coupon-level strength-prediction validation. The following three sections of this paper address these points. The final section of the paper summarizes these results.

Theoretical Basis of IBOLT

Joint Problem Definition

Composite bolted joints exhibit two distinct types of load vs. deflection behavior when loaded to static failure. The load vs. deflection response for combined bearing and bypass loading is nearly linear to failure. The load vs. deflection behavior for bearing load only, however, displays significant non-linearity very similar to that of a metal-to-metal joint. The IBOLT program described here performs separate static failure analyses for these two types of loading.

The following steps shown in Figure 1 are performed when employing IBOLT to predict the static strength of a composite bolted joint, a task that is accomplished on a fastener-by-fastener basis:

1. The load transferred by each fastener is determined (magnitude and direction). This task is usually performed with the aid of a finite element model (FEM), but can be accomplished by spring models or by hand calculation for simple joints.
2. The membrane loads and bending moments present in the composite laminate for a fastener location of interest are defined.
3. A rectangular element of composite laminate of appropriate size is selected, and a freebody diagram satisfying static equilibrium of this "joint element" is defined.
4. The IBOLT program then provides the margin of safety for static strength for this joint element.

Steps 2 through 4 are repeated for each joint element and load case of interest. The margin of safety for the joint is the minimum margin of safety obtained from any of the joint elements that were analyzed.

Analysis Overview

IBOLT performs a fracture-mechanics-based static strength prediction for a rectangular composite joint element subjected to a combination of biaxial membrane loads, shear loads, and an off-axis bolt load as shown in Figure 2. Out-of-plane bending moments are also treated. The membrane loads and shear loads are assumed to be constant along a given edge and are defined as tractions (lb./in.). The bending moments are also assumed to be constant along a given edge (in.-lb./in.). The bolt load is defined by a magnitude (lb.) and a direction (deg.).

In addition to the loading, other effects are treated in the IBOLT analysis. Joint configuration effects (single- or double-shear) are modeled by a beam-on-
elastic-foundation analysis to account for the thickness and stiffness of the joint members as well as the bolt bending and shear stiffness. Empirical equations are included to account for fastener head geometry, effects of filled versus open holes, fastener/hole clearance, and fastener torque.

Fracture-mechanics-based bolted joint static strength prediction for a rectangular composite element

FIG. 1—Composite bolted joint analysis steps.

Fracture-Based Failure Theory for Bypass Loading

One failure characteristic of composite laminates containing holes is a sensitivity to hole diameter. For a constant width-to-diameter ratio, the strength of notched
laminates varies with hole size as shown in Figure 3. As the hole becomes very small, the strength approaches the unnotched laminate strength. This variation of strength with hole size is also observed for compression loading and for bearing loading as shown in Figure 4.

![FIG. 3—Laminate bypass strength varies with hole size [6].](image1)

Conventional strength-based failure theories are not applicable for this type of behavior. Two common strength-based failure models are shown in Figure 5. For ductile materials, the net section stress at failure is equal to the unnotched strength. For a constant width-to-diameter ratio, the predicted value of remote stress at failure is then a constant value as indicated by the upper horizontal line.

![FIG. 4—Bearing strength variation due to hole.](image2)
For brittle materials, the remote stress at failure would be equal to the unnotched strength divided by the elastic stress concentration factor at the critical location – on the hole boundary at the narrowest section. This also provides a constant value of predicted remote stress at failure as indicated by the lower horizontal line in Figure 5. Neither of these criteria shows the variation in strength with hole size observed in the test data.

In searching for a suitable failure criterion, we found that one of the linear elastic fracture mechanics (LEFM) solutions utilized a curve that had a shape similar to the composite failure stress versus hole size curve. This solution, developed by Bowie [5], treats the case of a tensile coupon with a central hole having symmetric edge cracks on the hole boundary as shown in Figure 6. This solution provides values of the function $f(a/r)$ as a function of $a/r$ where $a$ is the length of each edge crack, $r$ is the radius of the hole, and $K_i$ is the stress intensity factor.

$$K_i = \frac{\alpha}{\sqrt{a} f\left(\frac{a}{r}\right)}$$

Figure 6—Bowie solution has same shape.
It was shown in [6] that for a composite laminate having an elastic stress concentration factor near the isotropic value of 3.0 and assuming that the laminate behaved as if there were small symmetric edge cracks on the hole boundary at the next section as in Bowie’s solution, that the notched strength could be predicted by the following equation:

\[
\sigma_c = \frac{F_u}{f\left(\frac{a}{r}\right)}
\]

FIG. 7—Fracture mechanics models the strength behavior for circular holes [6].

Here \(\sigma_c\) is the notched failing stress, \(F_u\) is the unnotched failing stress, and Bowie’s \(f(a/r)\) function serves as an effective stress concentration factor that incorporates the effects of both hole size \((r)\) and a fixed effective flaw size \((a)\). Since all test data reported in [6] were conducted on the same laminate, a single value of “a” was chosen to provide the best-fit curve shown in the Figure 7. As shown, the equation matches the experimental behavior very well. A more detailed derivation of Equation 1 is given in [6].

Generalization of LEFM Solution for Combined Loads

Up to this point, the LEFM solution has been applied only for by-pass tensile loading of an unloaded hole. To analyze joint elements taken from complex joints containing multiple fasteners, the type of loading shown in Figure 8 must be addressed.

Nine Basic Loading Conditions - Figure 9 shows a set of nine basic load cases that can be combined to treat a realistic joint element loading condition. These include five types of bypass loading: X-direction tensile (NX), Y-direction tensile (NY), shear (NXY), and membrane loads reacted in shear in the X-direction (NXX) and Y-direction (NYY). Bearing loads reacted in tension in the X-direction (PNX) and Y-direction (PY) are included as well as bearing loads reacted in shear in both
Bowie solved this problem

Realistic bolted joints have more complex combinations of loading

FIG. 8—Complex Bolted Joint Loading.

FIG. 9—Basic bolted Joint Load Cases.

the X-direction (PSX) and Y-direction (PSY). The sign of the loading may be reversed for any of the nine basic load cases. Bowie-type LEFM solutions were generated numerically using the boundary element method [7] to treat each of these nine basic load cases so that superposition could be used to analyze complex loading situations formed by combinations of the basic load cases.

In the example of Figure 10, three of the nine basic load cases are combined to form the complex load state representing a joint element at the edge of a structural panel having a single free edge. Loads acting on the element boundaries are given as constant tractions with units of lb/in. Fastener loads acting on the hole-boundary are expressed in pounds. For simplicity, the example shown assumes the element dimensions are 1.0 inch by 1.0 inch to allow the edge tractions (lb/in) and bolt load (lb) to be summed directly. Since each basic load case is in equilibrium, any combination of basic load cases will result in a balanced element freebody diagram.
Eight Failure Locations - For the open hole tensile coupon, the failure location is expected to be on the hole boundary at the net section of the coupon. For more complicated loading conditions, the failure location is not as easy to determine. Ideally all locations on the hole boundary would be evaluated for possible failure, and the most critical location would be selected. After observation of many failed composite coupons, a simplifying assumption was made to reduce the scope of the required analysis. It was observed that for the [0/±45/90]_c family of laminates, failure occurred at or near one of the eight locations shown in the Figure 11. These are the eight locations where fibers are tangent to the hole boundary. Each failure appeared to initiate as a through-the-thickness radial crack. The coupons evaluated had been loaded to failure with bypass loading, bearing loading, and combined bearing/bypass loading. These eight failure positions are numbered sequentially counterclockwise at 45-degree intervals. Position 1 is located at the intersection of the hole boundary with the X-axis. The local element XY coordinate system has its origin at the center of the fastener hole. The X-axis is defined to be parallel to the 0-degree ply direction.
The unnotched laminate strength in the tangential direction is not constant at the eight failure positions around the hole. Figure 12 shows how the percentages of tangential, off-axis, and radial plies vary with position for a (50/40/10) laminate (i.e., one containing 50% 0°-plies, 40% ±45°-plies, and 10% 90°-plies). Tangential plies are those plies oriented tangent to the hole-boundary at a given position. Radial plies are those oriented in a direction normal to the tangential plies (radial with respect to the center of the hole). Off-axis plies are those plies oriented at ±45 degrees to the tangential direction.

Since the ply content of the laminate in the tangential direction varies from one failure position to another, the value of the effective flaw size “a” also changes with position. Given the thickness fractions of the tangential, off-axis, and radial plies at the eight failure positions, the value of “a” can be calculated as follows:

\[ a = \left( \frac{t_{\text{tang}}}{t_{\text{total}}} \cdot G_0 + \frac{t_{\text{oa}}}{t_{\text{total}}} \cdot G_{45} + \frac{t_r}{t_{\text{total}}} \cdot G_{90} \right) / \left( C \cdot \pi \cdot F_{tu}^2 \right) \]  

Here, C is the constant defined in Paris and Sih [8] relating critical strain energy release rate to fracture toughness:

\[ C = \sqrt{\frac{S_{11} S_{12}}{2} \left( \frac{S_{11}}{S_{22}} + \frac{2 S_{12} + S_{66}}{2 S_{22}} \right)} \]  

\( S_{11}, S_{22}, S_{12}, \) and \( S_{66} \) are the laminate compliance values oriented with respect to the tangential ply direction. In equation (2), \( G_0, G_{45}, \) and \( G_{90} \) are the critical strain energy release rates for 0°, ±45°, and 90° plies and are summed according to their thickness fractions (also with respect to the tangential ply direction) to provide a laminate critical strain energy release rate according to [9]. \( F_{tu} \) is the unnotched tensile strength in the tangential direction. \( G_0, G_{45}, \) and \( G_{90} \) are defined early in the model calibration process using open hole tension (OHT) data from three or more
laminates as discussed later. Figure 13 shows the typical variation in the value of "a" with position on the hole boundary for a (50/40/10) laminate. The Bowie-type LEFM solutions that were developed for the nine basic load cases included treatment of through-the-thickness radial cracks at each of the eight failure positions. The ability for the effective flaw size "a" to be a function of laminate ply percentages, rather than a single fixed value, provides this strength model with the ability to better reflect the physics of damage development over a wide range of laminate configurations. This flexibility thus provides a better prediction of joint strength over the full extent of the design space.

\[ a = f(\text{material properties, ply}) \]

**FIG. 13**—Circumferential variation in crack length.

Element boundaries are normally defined midway between adjacent fasteners.

**FIG. 14**—Element boundary definition.
Finite Geometry Effects

Appropriate values of joint element length and width must be defined in order to perform the static strength analysis. A convenient approach is to define the element boundaries to lie midway between adjacent fastener lines as shown in Figure 14. IBOLT treats each joint element as an isolated rectangle of length $A$ and width $B$ with constant loads applied on the element boundaries. For bypass loads, this approach is conservative as it ignores the shadowing effect caused by the proximity of adjacent fastener holes which tends to divert load away from the center of the joint element.

Each IBOLT solution is based on infinite plate equations which are then corrected for the finite dimensions of the joint element ($A$ and $B$). These finite geometry correction factors were generated numerically using the boundary element method [7] for all nine load cases and eight failure positions for the usable range of laminate ply percentages and element geometries as summarized in Figure 15. These finite geometry factors are stored within IBOLT in a large lookup table that allows rapid retrieval of the values appropriate for each analysis.

Joint Element Orientation

Proper alignment of the IBOLT joint element is a key step of the analysis process. Figure 16 illustrates orientation of the joint element with respect to the laminate ply directions. Correct orientation of the IBOLT element is achieved by aligning the element X-axis with one of the four fiber directions for the $[0/\pm45/90]$ laminate. Figure 16 shows four elements that are correctly aligned and one element that is incorrectly aligned. As discussed previously, the eight failure locations evaluated by IBOLT are located at the eight points where the plies are tangent to the
hole boundary. Unless the element is aligned with one of the four laminate ply directions, these failure locations will not correspond with the tangency points, and the analysis will be invalid. Along structural boundaries that are not parallel to one of the four ply directions, one corner of the IBOLT element may extend beyond the physical edge of the structure as shown in the lower right corner of the structure.

FIG. 16—Proper IBOLT element alignment.

The element may be aligned with any one of the four fiber directions as appropriate. If the element X-axis is aligned with a fiber direction other than the 0° direction, the ply percentages will have to be redefined. For example, the element shown in the lower left of Figure 16 is aligned with the -45° fiber direction. If the laminate in the panel coordinate system is a (50/20/20/10) laminate, the IBOLT ply percentages will be defined as (20/50/10/20) for this particular joint element. In all cases, the hole must be located in the center of the IBOLT element and aligned with one of the four fiber directions.

Margin of Safety Calculation

The failing strength for an infinite plate with an open hole, loaded in axial tension can be predicted by dividing the unnotched laminate strength $F_{lu}$ by the Bowie function $f(a/r)$:

$$\sigma_c = \frac{F_{lu}}{f(a/r)}$$  \(4\)

The Bowie function $f(a/r)$ serves as an effective stress concentration factor that includes the effect of hole diameter (hole radius “r”) and an effective flaw size (crack length “a”). The open hole coupon fails at locations 3 and 7 (at the net section), and $F_{lu}$ is the unnotched strength of the laminate in the tangential direction at locations 3 and 7 shown in Figure 11. Each of the eight locations on the hole
boundary has a value of unnotched strength \( F_u(i) \) and effective flaw size \( a(i) \).

Equation (4) can thus be rewritten to apply to each of the 8 locations on the hole boundary (denoted by the index "i") and corrected for finite geometry effects using the correction factor \( LTX(i) \). The resulting equation shows the remote applied axial X-direction tensile stress required to initiate failure at location i:

\[
\sigma_c(i) = \frac{F_u(i)}{FTX(i) \left( \frac{a(i)}{r} \right) \cdot LTX(i)}
\]

(5)

Where, at location "i":
- \( F_u(i) \) is the tangential strength;
- \( FTCX(i)(a(i)/r) \) is the Bowie function;
- \( LTX(i) \) is the finite geometry correction factor; and
- \( \sigma_c(i) \) is the applied remote stress required to initiate failure at location "i."

Rearranging equation (5) gives:

\[
Ftu(i) = \sigma_c(i) \cdot LTX(i) \cdot FTCX(i) \left[ \frac{a(i)}{r} \right]
\]

(6)

For the purpose of combining stresses contributed by each of the nine load cases that we wish to include, equation (6) can be expanded to:

\[
F(i) = + SIGTX \cdot LTX(i) \cdot FTCX(i) \left[ \frac{a(i)}{r} \right] \cdot FTF \quad \text{Tension, X-direction}
+ SIGXX \cdot LTXX(i) \cdot FTCXX(i) \left[ \frac{a(i)}{r} \right] \cdot FTF \quad \text{Membrane/shear, X-direction}
+ SIGTY \cdot LTY(i) \cdot FTCY(i) \left[ \frac{a(i)}{r} \right] \cdot FTF \quad \text{Tension, Y-direction}
+ SIGYY \cdot LTYY(i) \cdot FTYY(i) \left[ \frac{a(i)}{r} \right] \cdot FTF \quad \text{Membrane/shear, Y-direction}
+ SIGXY \cdot LTXY(i) \cdot FTCXY(i) \left[ \frac{a(i)}{r} \right] \cdot FTF \quad \text{Shear, X-Y}
+ SIGBX \cdot LBX(i) \cdot FBX(i) \left[ \frac{a(i)}{r} \right] \cdot FBF \quad \text{Bearing/tension, X-direction}
+ SIGXX \cdot LBXX(i) \cdot FBXX(i) \left[ \frac{a(i)}{r} \right] \cdot FBF \quad \text{Bearing/shear, X-direction}
+ SIGBY \cdot LBYY(i) \cdot FBYY(i) \left[ \frac{a(i)}{r} \right] \cdot FBF \quad \text{Bearing/tension, Y-direction}
+ SIGYY \cdot LBYY(i) \cdot FBYY(i) \left[ \frac{a(i)}{r} \right] \cdot FBF \quad \text{Bearing/shear, Y-direction}
\]

(7)

This equation shows the total effective tangential stress at location "i" resulting from the contributions of all nine load cases. The first line in the equation is the contribution resulting from an applied tensile stress in the X-direction (SIGTX) and is equal to equation (6) multiplied by a joint configuration factor. Each successive line represents the contribution to the effective tangential stress at location "i" resulting from a different load case for a total of nine load cases. Each line in the equation is the product of the applied load, the finite geometry correction factor, the effective stress concentration factor (the appropriate Bowie function for load type and location on hole boundary), and the empirical factor FTF or FBF. The FTF term represents the cumulative effect of additional factors that affect bypass loading. These include filled hole effects, countersink effects, torque, and clearance effects. The FBF term
represents the cumulative effect of additional factors that affect bearing loading such as single/double shear effects, countersink effects, and torque effects. The functional form of these terms is:

\[
FTF = FCNTT * FFILL * FTTORK * FCLEAR
\]

(8)

\[
FBF = FSSDS * FCSKBC * FBTORK
\]

(9)

Where:

- \( FCNTT \) and \( FCSKBC \) are by-pass and bearing countersink correction factors,
- \( FFILL \) is a filled-hole by-pass correction factor,
- \( FTTORK \) and \( FBTORK \) are fastener off-nominal torque correction factors,
- \( FCLEAR \) is a hole-clearance factor, and
- \( FSSDS \) a single-shear bearing factor.

All such empirical correction factors, as well as the bearing cut-off factors discussed below, are derived from multi-variate regression analysis of open- and filled-hole tension and compression test data; and tension-, compression-, and shear-loaded bearing test data (see following discussion on “required IBOLT input properties”). Finally, the by-pass margin of safety at location “i” is given by

\[
M.S. = F_{tu(i)}/F(i) - 1.0
\]

(10)

A margin of safety is calculated at each of the eight locations on the hole boundary. The lowest of these eight values is the critical margin of safety for bearing/bypass loading. This margin is compared with the bearing cutoff margin of safety to determine the critical margin of safety for the joint element.

**Bearing Cutoff**

The basic strength model used in IBOLT only deals with a single failure mode--progression of a radial, through-the-thickness crack from the edge of the hole. For the pure bearing case (no bypass loading), additional failure modes may be involved. A separate check on bearing strength for the no-bypass condition called the “bearing cutoff” calculation has been included in IBOLT.

The starting point for the bearing cutoff check is the measured bearing value in the material data file that is appropriate for the type of loading. The material data file contains test results for BCT (bearing with tension reaction), BCC (bearing with compression reaction), and BCS (bearing with shear reaction). Each of these bearing configurations, seen in Figure 17, is unique and has a different bearing strength value. The single-shear bearing test values BCT, BCC, and BCS are obtained from a (25/50/25) laminate attached with a 0.25-inch-diameter, protruding head, high-strength steel fastener to a rigid steel fixture approximately 0.80 inch thick. Values of \( e/D \) and \( W/D \) are approximately 2.5 and 5.0, respectively. Loading for the test is in the \( 0^\circ \)-ply direction. The goal of the bearing cutoff check is to provide a bearing value appropriate for the actual structural joint being analyzed. To provide this, it is necessary to correct the value from the data file for fastener diameter and type,
laminate ply percentages and thickness, e/D, W/D, and substructure modulus and thickness.

![Bearing Cutoff with Tension Reaction (BCT)](image1)

![Bearing Cutoff with Compression Reaction (BCC)](image2)

![Bearing Cutoff with Shear Reaction (BCS)](image3)

**FIG. 17—Three types of bearing reactions.**

Both a primary and a secondary method are used to provide the bearing cutoff value as outlined in Figure 18. The lower value obtained from these two methods is used to calculate the bearing cutoff margin of safety.

![Two Methods Are Used To Calculate Bearing Cutoff](image4)

**FIG. 18—Two bearing cut-off criteria.**

**Beam on Elastic Foundation** - The primary method of calculation uses a beam-on-elastic-foundation analysis to provide a bearing stress profile through the thickness of the laminate. As shown in the Figure 19, this analysis models the fastener, the
The skin and substructure serve as elastic foundations with appropriate moduli and thicknesses. The shim is modeled as an unsupported length of fastener between the two foundations. Fixity provided at the ends of the beam (fastener) by the head and nut are modeled as springs that resist rotation at these two points. Values for the spring constants are obtained by comparing predictions with test results for a given fastener system and adjusting spring constants until correlation is obtained. The analysis is based on a paper by William Barrois [10] and modified to add an unsupported length of beam between the skin and substructure representing a shimmed gap. This method accounts for laminate and substructure modulus and thickness, fixity provided by the head and nut, and provides a factor FPKBC that is the ratio of peak bearing stress-to-average bearing stress. Other factors, as shown in Figure 18, are used to account for fastener diameter (FDIABC), e/D and W/D (FGEOBC), %0° and %90° plies (FPCTBC), and bolt load direction (FTHBC). The peak bearing stress provided by this analysis is compared to the reference bearing stress contained in the material data file (minimal peaking in bearing stress) to determine the margin of safety. Output includes shear and bending moment values along the length of the fastener.

![Beam-on-elastic-foundation fastener model](image)

**FIG. 19—Beam-on-elastic-foundation fastener model.**

**Knockdown Factor Approach** - The secondary method of calculation applies a series of empirical factors to the bearing value in the data file (BCT, BCC, BCS) to account for diameter (FDIABC), countersunk fastener head (FCSKBC), e/D and W/D (FGEOBC), %0° and %90° plies (FPCTBC), and bolt load direction (FTHBC). These reductions are applied in both the X- and Y- directions since FGEOBC and FPCTBC are functions of ply percentages, edge distance, and width and have different values in these two directions. A final correction is applied for bolt load angle (FTHBC) assuming a cosine 20 variation in bearing strength with angle. The secondary analysis method applies to the rigid substructure case as no correction is
made for substructure thickness and modulus. For rigid substructure, the primary method can over-predict the bearing strength. Use of the secondary analysis (when it predicts a lower bearing cutoff value) insures that bearing strength predictions will not be unconservative.

Theoretical Limitations

Because of the approach taken in defining the basic IBOLT strength model, the program has several limitations as summarized in Figure 20. The boundary loads are assumed to have a constant magnitude along any given element boundary. The laminate being analyzed must be a member of the [0/±45/90] family; no other ply angles are permitted. The percentage of plies in any one of the four permissible ply directions must be between 7 and 60 percent. The IBOLT element edges must be aligned with one of the four ply directions and will not necessarily be aligned with the edges of the structural boundary. Since the pure bearing case has additional failure modes not treated in the basic by-pass strength model, the prediction for the pure bearing (no bypass) case is largely empirical. Of course, the by-pass strength model is semi-empirical, since the strain energy release rate properties and various correlation factors are derived from notched laminate test data. Finally, the IBOLT element must be rectangular with length and width ranging from at least two to no more than ten fastener diameters, with the fastener hole located at the center of the element. Within these limits, any combination of A and B values is permitted.

Fig. 20—IBOLT theoretical limitations.

Required IBOLT Input Properties

For a given material system and environment, the IBOLT program must be calibrated with a series of lamina and laminate coupon tests as outlined in Figure 21.
The calibration process is sequential in nature and begins by comparing lamination theory predictions of laminate modulus (average of tension and compression) with experimental values provided by strain-gaged unnotched tension (UNT) and unnotched compression coupons (UNC). The lamina modulus value of $E_{11}$ in the material data file is adjusted to provide the best correlation between measured and predicted laminate moduli.

Predicted unnotched laminate tension strength values for three or more laminates are then compared with test results to define an appropriate single value of laminate failing tensile strain $\varepsilon_{11T}$. The tested laminates exhibit fiber-dominated behavior having $0^\circ$ ply percentages typically ranging from 15 to 60 percent. Since the $0^\circ$ plies are the major load carrying component of the laminate, the theoretical laminate failing strain should equal the failure strain of the $0^\circ$ plies – a constant value. This approach differs from the common industry practice of assigning higher failing strains to soft laminates and lower failing strains to hard laminates, but provides a sufficiently accurate estimate of the unnotched strength for use in predicting bolted joint strength. Following this, IBOLT-predicted open hole tension (OHT) strengths for three or more laminates are compared with test results, and values of critical ply strain energy release rate $G_0$, $G_{45}$, and $G_{90}$ are adjusted to match predictions with experimental values. Next, IBOLT-predicted values of filled hole tension strength for the same set of laminates are compared with test results. Parameters defining the fill factor for tension (FFILLT) are adjusted to provide the best correlation.

A similar path is followed for correlation of compression test results with IBOLT predictions using UNC coupon data to define the best single value of laminate...
failing compression strain EPS11C, open hole compression (OHC) data to define compression correlation parameters (FTFCS), and filled hole compression data to specify parameters for the compression fill factor FFILLC.

The effect of ply percentage on bearing strengths involves creating biquadratic surface fits to the available test results for BCT, BCC, and BCS. The effects of fastener torque, countersink fastener head geometry and countersink depth, width, and edge distance are typically not sensitive to material type and thus can often be determined from legacy test results from previous material systems.

In the final step, notched flexure coupons containing open, filled, and filled countersunk holes are used to set parameters controlling prediction of out-of-plane moment effects (FOHBND, FFHBND, FFCBND).

This process is repeated for each material system and environment of interest to define the required set of material property values. For design purposes, linear interpolation of property values for intermediate temperatures is permitted.

**Experimental Validation**

Figure 22 is a correlation plot of measured versus predicted bearing/by-pass failure loads. Predictions were made using IBOLT and the previously-noted semi-empirical input data development. None of the bearing/by-pass tests were used to develop the IBOLT input data. The bearing/by-pass tests shown in Figure 22 included two material forms (one tape, one fabric), three environmental conditions (CTD, RTD, ETW), three by-pass ratios, seven tape-, and three fabric lay-ups. The material system in all cases was a carbon fiber/bismaleimide, all fasteners were 0.250-inch diameter protruding head bolts, and the joint configurations were all double shear. Each point in Figure 22 is the average of three-to-five individual tests.

Additional validation studies are available in [11] for four bearing/bypass load ratios as well as three complex applied loading states. The complex loading states were obtained by orienting the test laminate at an angle with respect to the applied loads to create a state of biaxial in-plane load, plus shear, combined with an off-axis bolt load.

**Summary**

The objective of the body of this paper was to describe the theoretical basis, required input, and coupon-level strength-prediction validation for the IBOLT bolted joint static strength model. In addition to the preceding technical discussion, a brief summary of IBOLT’s strengths and weaknesses as a practical bolted joint stress analysis tool is also now provided.

IBOLT has the following general features:

- Applies to the [0/±45/90]s laminate family;
- Percent of +45° plies may be different from percent of −45° plies;
- Treats rectangular, finite-geometry joint element;
- Predicts strength as a function of hole diameter;
- Employs fracture-mechanics-based failure theory;
• Allows combinations of nine basic types of loading;
• Defines effective flaw size as a function of ply percentages;
• Analyzes for possible failure at eight positions on hole boundary;
• Models both single and double-shear joint configurations using a beam-on-elastic foundation analysis to capture bolt bending and shear effects; and
• Empirically includes tensile and compressive bearing cut-offs, as well as a variety of fastener/geometry effects, such as countersinks, head/tail fixity, hole-clearance, and fastener torque.

![Graph](image)

**FIG. 22—Correlation of IBOLT with bearing/by-pass tests.**

Technical and operational strengths of IBOLT relative to other bolted joint analysis methods and empirical bearing/by-pass interaction curves include:

• Correctly capturing all relevant permutations of freebody loading configurations for a single bolt in a rectangular plate element;
• Handling a wider variety of joint configurations and fastener types/effects; and
• Running quickly enough as a subprogram to be included in automated optimization routines and multi-site streamed analysis methods.

Some of the relative weaknesses of IBOLT include:

• The theoretical limitations noted in Figure 20, particularly the (0/45/90) laminate-family restriction;
• The cost of generating the G0, G45, and G90 B-basis values from notched laminate testing; and
• The cost of generating empirical bearing cut-offs from laminate-level bearing testing over a range of laminates and environmental conditions.

Overall, IBOLT is deemed by LM Aero to be the most accurate and capable
bolted joint analysis method available today.

Acknowledgements

This work was performed under LM Aero internal funding.

References


**Hui Bau**

**Damage and Failure Mechanisms in Composite Bolted Joints**


**ABSTRACT:** This study’s overall goal is to develop methods to accurately predict the strength of notched composite laminates. Empirical analyses of notched composite test data led to an understanding of failure mechanisms. Trends in ultimate strength versus layup parameters were extracted from several DoD test programs. A test program with incrementally loaded, notched coupons was conducted to track sub-critical damage. Damage states at near-ultimate loads were correlated to the ultimate strength trends. These trends were then attributed to specific matrix damage types.

Detailed finite element analyses of composite bolted joint configurations were conducted using solid element models with nonlinear material properties. Sensitivity studies were conducted to determine the mesh refinement needed for converged peak stresses. At the loads corresponding to initial damage in the incremental notched tests, linear and nonlinear finite element analysis consistently predicted peak ply stresses which were significantly higher than ply strengths derived from unidirectional coupon tests.

**KEYWORDS:** bolted joints, composite material, laminate, notched strength, open hole strength, failure modes, finite element analysis, nonlinear, matrix cracking

**Introduction**

Many methods to predict the strength in composite materials have been proposed over the last 30 years [1,2]. None of those methods have been shown to give accurate strength predictions over a wide range of materials, laminates, and test configurations. Most previous researchers who have worked towards predicting composite strength have tended to focus on developing failure criteria. However, it is necessary to resolve all of the issues shown in Figure 1 before a composites strength prediction method can be validated enough for use in all structural designs. Since each of these issues has its inherent complexities, several steps are needed to resolve the problem.

![Diagram](fig1.png)

**FIG. 1 — Predicting composites strength – issues to resolve**

\(^1\) Self-employed, HubCo, HC68 Box 15G, Taos, NM, 87571, huibau@taosnet.com
The author takes an approach to narrow and bound the problem by working towards predicting strength in notched tape composites (open and filled holes under uniaxial tension and compression) and by 1) detailed understanding of failure mechanisms and damage progression from test data, 2) obtaining accurate predictions of stress fields around the hole, 3) understanding and obtaining accurate ply strengths and finally 4) backing out the failure criteria from the test data, accurate stresses and accurate ply strengths. This paper presents results from continuing studies of initial damage and ultimate strength from notched composites test data and finite element analysis (FEA) of initial damage in notched composites [3-7]. Future work will include the study of in-situ effects on strength, evaluation of failure criteria, and damage progression modeling.

A major reason for the lack of accurate predictive methods is that failure in composites is a result of a highly nonlinear process which includes material nonlinearities and different modes of damage progression. As damage occurs in the composite, stress concentrations change locally and globally, which affect the damage progression as well as ultimate failure strength.

**Ultimate Strength Failure Mechanisms**

An empirical study was conducted towards improving the understanding of ultimate strength behavior in notched composite laminates. This study was based on the concept that the complex ultimate strength behavior in notched composites can be attributed to different damage states just prior to ultimate failure, and configurations which have the same damage state just prior to failure will show a consistent ultimate strength trend over that configurations’ parameters. Thus, the main reason that notched composites strength data appears so inconsistent is that several failure mechanisms may be present for one material system over a wide range of layups, environments, and test configurations. While the ultimate failure mode for notched coupons in uniaxial tension or compression is net section, there are different failure mechanisms which result in variations in ultimate strength. All of the failure mechanisms in this report led to net section failures.

There were three tasks to the empirical failure mechanisms study: 1) a study of ultimate strength vs. layup parameters to determine ultimate strength trends, 2) a study of damage just prior to ultimate failure, and 3) correlating ultimate strength trends to the damage state at near-ultimate loads. All of the laminates in this study were of the (0°±45°/90°) layup family, which is commonly used in aircraft structures.

**Ultimate Strength Trends**

To determine the ultimate strength trends, a database of notched composites tests from several DoD aircraft allowables test programs was compiled [10-16]. There were more than 5000 notched composites tests conducted for these programs, which included various carbon fiber composite materials with layup, geometry (width), and environment variables as well as bolt variables such as open and filled holes, diameter, countersink, etc. Generally, these test programs had baseline geometry and bolt configurations which were tested with several layups.

To reduce the confusion from the effect of confounded variables, this study focused on evaluating layup variables for the baseline notched tension and compression configurations in carbon fiber tape laminates [6, 7]. Data from over 1500 notched composite baseline tests were used in this study. There were over 40 baseline...
material/environment/geometry configurations which included 5 material systems in up to 5 thermal/moisture environments, with filled or open holes under uniaxial tension or compression. Each baseline configuration consisted of 4-8 tested layups with 1-5 material batches for each layup and 3-6 replicates for each batch. For each baseline configuration, ultimate strengths were plotted over various layup parameters such as 0° plies, modulus, etc.

The ultimate strain trends over the layup parameters listed in Table 1 were identified from the plots. For some configurations, there were obvious trends. Each trend had material, layup, and environment characteristics that were consistent between the configurations. For example, the constant strain trend which typically occurred in softer layups, was also more common in configurations tested at elevated temperatures and in materials with softer matrices. The characteristics and the behavior of the identified trends were used to attribute ultimate strain trends to test data in plots which did not have obvious trends.

Over 98% of the test data could be categorized under one of the identified strain trends listed in Table 1. In some cases, the characteristics of an identified trend had been previously noted in literature and were associated with a stacking sequence parameter, such as the delaminations with 0°/90° ply clusters [1] and 0° tangential matrix cracking in very stiff laminates [9]. However, the basic trend and constant strain trend had not been previously explicitly identified and associated with failure mechanisms, since these trends are difficult to discern and understand from an individual test program. Using a large body of test data where configurations tested at various environments with different materials can be directly compared, the five identified trends and their characteristics could be differentiated with confidence.

<table>
<thead>
<tr>
<th>Identified Trends</th>
<th>Characteristics</th>
</tr>
</thead>
<tbody>
<tr>
<td>Basic</td>
<td>generally decreasing ultimate strain with increasing stiffness; the most common trend, occurs over all material systems and configurations</td>
</tr>
<tr>
<td>Constant Strain</td>
<td>constant ultimate strain which tends to be lower than what would be expected from the basic trend; tends to occur in softer configurations such as high temperatures, soft layups, less constraint at the hole</td>
</tr>
<tr>
<td>Delamination</td>
<td>increased ultimate strain for layups with adjacent 0° and 90° plies</td>
</tr>
<tr>
<td>0° Tangential Matrix Cracking</td>
<td>increased ultimate strain with increasing 0° plies and thicker stacks of 0° plies, also increased scatter which appears to be related to high variability in the 0° tangential matrix crack lengths where longer cracks result in increased ultimate strains</td>
</tr>
<tr>
<td>Very Soft Layups</td>
<td>jump or drop in ultimate strain compared to the extrapolation of the trends in more moderate layups; attributed to change in failure mechanism; some ultimate strain drops and jumps appear to be due to a change from the basic trend to the constant strain trend; but there is insufficient data to determine the failure mechanism for other configurations with strain drops or strain jumps</td>
</tr>
</tbody>
</table>

The basic trend was the most common trend and occurred over a wide range of tests. Interestingly, the basic trend is nearly parallel to strength predictions generated
using peak axial stresses from 2D anisotropic stress fields [17] with a simple axial ply strength failure criteria. In this report, these predicted strains are referred to as e2DKt. This trend is the basis for the characteristic dimension approach for notched laminate strength predictions [18] which is commonly used for aircraft structural analyses.

Figures 2 and 3 show comparisons between tests identified with the basic trend and the e2DKt predictions. Figure 3 has very high scatter in the open hole compression test of a stiff layup which is attributed to replicates with different failure mechanisms. Figures 4 and 5 show four of the identified ultimate strain trends.

**FIG. 2 — Filled hole tension test data with the basic ultimate strain trend**

**FIG. 3 — Open hole compression strain at room temp./ambient — ultimate strain trends**
JOINING AND REPAIR OF COMPOSITE STRUCTURES

- IM6/3501-6 grade 190 tape, Open Hole Tension, cold/ambient environment

- 0° tangential matrix crack trend
  - these stiff layups have thick stacks of 0° plies

- basic trend with delaminations
  - these layups have adjacent 0°/90° plies

- OHT, IM6/3501-6, -75°F/ambient moisture, dia=.312 in, width/dia=5
- OHT, IM6/3501-6 tape, -65°F/ambient moisture, dia=.25 in, width/dia=6

FIG. 4 — Open hole tension at cold/ambient – ultimate strain trends

- IM6/3501-6 tape, Open Hole Tension, room temperature/ambient moisture

- constant strain trend
  - layup A
  - layup B

- OHT, IM6/3501-6 tape, room temperature, ambient moisture, dia=.312in, width/dia=5
- OHT, IM6/3501-6, room temperature, ambient moisture, dia=.25in, width/dia=6

FIG. 5 — Open hole tension at room temperature – ultimate strain trends
Incremental Damage Studies

The second task involved evaluating damage information from a notched composites test program which was conducted at University of Dayton Research Institute (UDRI) on IM6/3501-6 coupons [19]. Four layups were tested in three configurations: open hole, filled hole with low-torqued fasteners, and filled hole with fasteners installed at full torque. Monotonic and incrementally loaded tension and compression tests were conducted. The monotonically loaded tests were instrumented with strain gages at the side of the hole. The filled hole configurations included load washers to determine clamp-up loads.

Figures 6 - 8 show the notched tension and compression monotonic test data and the ultimate strength trend attributed to each test. These attributed trends were determined by comparing the considerable amount of IM6/3501-6 test data from the empirical ultimate strength study with the UDRI test data. Four of the ultimate strain trends from the empirical trend study were represented in the notched tension tests. Two ultimate strain trends were represented in the open hole compression tests. The failures in the filled hole compression tests appeared to initiate in the composite at the outer edge of the washers.

FIG. 6 — IM6/3501-6 open hole and low torque tension – ultimate strain trends
FIG. 7 — IM6/3501-6 full torque tension – ultimate strain trends

FIG. 8 — IM6/3501-6 notched compression tests – ultimate strain trends
For the incremental tests, each coupon was loaded in ten increments to try to capture the onset of damage and the damage state just prior to ultimate failure. After each load increment, the coupon was unloaded, then the hole and coupon surface near the hole were swabbed with dye-penetrant and X-ray radiographs were taken. In the filled hole incremental tests, the bolts were removed for each X-ray, then re-installed.

The radiographs were digitized so they could be visually evaluated with computer enhancements. The different sub-critical damage types around the hole which were observed in the radiographs are shown in Figure 9. Most of the sub-critical damage types were intra-ply matrix cracks which emanated from the hole edge and followed the fiber directions. These cracks often initiated in locations which were tangential or radial to the hole edge. Small delaminations were difficult to detect with this method. Also, damage which did not allow a dye-penetrant path to the hole edge or coupon surface could not be detected, such as damage which initiated near the edge of the washer in filled hole compression specimens. The damage types were recorded for each load increment.

FIG. 9 — Sub-critical damage types

Figures 10 and 11 show sketches of the damage states in the radiographs from the load increment just prior to failure. Since there were usually three replicates, these sketches show the damage combined from all of the replicates. The highest ratio of the next to last load increment vs. the ultimate load for the replicates is listed for each layup configuration.
OHT = tension coupon with open hole

LHT = tension with low bolt torque

THT = tension with fully torqued bolt

note: in the layup code above, 62/29/9 = 62% 0° plies, 29% ±45° plies and 9% 90° plies

FIG. 10 — Sub-critical damage at near-ultimate loads for notched tension
Correlation Between Strength Trends and Damage Types

The damage types identified from the near-ultimate radiographs were tabulated with each of the ultimate strength trends. Damage types which were specific to the ultimate strength trends were isolated. From this, the three ultimate strength trends in the UDRI tests were attributed to the specific matrix damage types, as shown in Figures 12-14.
The basic failure mechanism was attributed to intra-ply matrix cracks in the \( \pm 45^\circ \) and \( 90^\circ \) plies which are at the side of the hole, but not directly at the critical location. These types of cracks would relieve the \( 0^\circ \) ply fiber stresses at the critical location.

The tension coupons with \( 0^\circ \) tangential matrix cracks also had basic matrix cracks, so these cracks combine to result in the \( 0^\circ \) tangential matrix crack trend for notched tension. The compression coupons with \( 0^\circ \) tangential matrix cracks were associated with the basic trend, so it appears that \( 0^\circ \) tangential matrix cracks result in stress relief for the fibers in the \( 0^\circ \) plies at the critical location in notched tension, but not in compression.

The constant strain trend was attributed to intra-ply cracks in the \( \pm 45^\circ \) or \( 90^\circ \) plies which initiate at the critical location, near the \( 90^\circ \) position at the side of the hole. It appears that the constant strain trend is due to a crack propagation failure mechanism which initiates from the associated cracks.

In the empirical trend study, the delamination ultimate strength trend was associated with side of hole delaminations which provide a stress relief to the \( 0^\circ \) ply at the critical location. There was inadequate data in this study to ascertain the failure mechanism(s) for the very soft layup trend. Since this trend exists for layups which have a very large percentage of \( \pm 45^\circ \) and/or \( 90^\circ \) plies, it is likely that the ultimate failures in these layups are not dominated by \( 0^\circ \) fiber failures at the critical side of hole location.

**FIG. 12 — Damage associated with the basic failure mechanism**

**FIG. 13 — Damage associated with the \( 0^\circ \) tangential matrix crack plus basic failure mechanism (notched tension)**
The failure mechanism information can be used qualitatively to assess various methods to predict notched composite strength. The strength trends can also be used to empirically "predict" ultimate strength with empirically derived "characteristic equations". For example, the constant strain trend can be described with a constant and the basic trend can be described with the $\varepsilon_2DK_1$ prediction plus a constant. However, since the characteristic equations require empirically derived constants, a significant amount of testing may be needed to define the required parameters. Further study of the characteristic equation constants may result in reduced requirements for testing.

**Empirical and Analytical Study of Initial Damage**

This study [3-5], which is still in progress, aims toward developing and validating methods to accurately predict damage and ultimate failure strength in notched composites through analytical techniques. Previous work using linear stress analysis techniques to determine stresses at the hole edge with ply strength based failure criteria have consistently resulted in ultimate strength predictions which were excessively conservative [1]. Various methods were developed which modify the predicted stresses and/or failure criteria, to obtain better correlation to test data. For example, the characteristic dimension approach reduces the predicted stresses by using ply stresses away from the hole edge [18]. Nonlinear FEA have been developed which can track intra-ply damage progression [8,9] and recent 3D damage progression methods [20] have been developed which can also predict delaminations. Most researchers continue to emphasize the stress analysis and/or failure criteria aspects of the problem and have not successfully predicted strength over a broad range of layups and materials.

The ability to predict initial matrix damage is the first step in the analytical prediction of ultimate strength through progressive damage modeling. The failure mechanism study showed that notched ultimate strength trends depend on the damage state just prior to failure, and that some types of damage have a stronger effect on ultimate strength than others. In multiaxial loading conditions, a benign damage type may become critical when load changes cause the location of the peak fiber stresses to shift. Therefore, to be able to accurately predict ultimate strength under general loading, it is necessary to predict all of the various types of sub-critical matrix damage.

The bounded approach was used to evaluate initial damage in this study. Initial damage types and loads were determined from test results and nonlinear 3D FEA was used to obtain accurate stress states at those loads. Then the predicted ply stresses at the
initial damage locations were compared to the ply strengths for a preliminary assessment of ply strength based failure criteria.

Test data for initial damage types and loads were obtained from the UDRI incremental x-ray tests [6, 7, 19]. For all tension and open hole compression specimens, matrix cracks appeared to initiate at the hole edge, although this may be a characteristic of the dye-penetrant/X-ray approach. For most configurations, there was a wide variation in initial damage loads. A few configurations also had variation in the types of initial damage. The initial damage data which were consistent between replicates were considered robust. Only a few tested configurations in the stiffer laminates had robust initial damage types and loads. Although there was considerable variation in the matrix damage initiation loads and damage progression sequences, there was much less scatter in the ultimate failure strengths.

A detailed 3D FEA study was conducted with the goal of obtaining an accurate stress state around the hole at initial damage loads for open hole configurations, since all of the initial matrix damage types initiated at the hole edge. Mesh refinement with linear FEA and linear and quadratic solid elements in a quasi-isotropic layup under axial tension was evaluated to determine the size of the solid elements at the hole edge necessary to produce converged peak stresses (within <10%). The meshes were increasingly coarser away from the hole edge. The most detailed model analyzed had seven elements through the thickness in each ply and 1 matrix interlayer element per ply. The analyses with quadratic elements were not more effective than the linear elements in reaching convergence, considering the computer run times. Table 2 shows the results of the sensitivity study for linear elements.

For convergence of peak in-plane ply stresses and peak out-of-plane shear stresses, the mesh at the side of the hole should have a few elements per ply through-the-thickness which includes a separate element for the matrix interlayer, and the elements should be thin in the radial direction (on the order of one fiber diameter). The angular dimension should be within the recommended aspect ratios considering the matrix interlayer and radial dimensions. More elements through-the-thickness or alternate methods are necessary to obtain convergence for peak through-the-thickness axial stresses. The ply thickness of IM6/3501-6 grade 190 tape studied was 0.19 mm. Most of the analysis results in this report were from models with four composite elements+matrix element/ply. For these models, the element size at the hole edge was 0.046 mm for the composite and 0.007 mm for the matrix in the thickness direction, 0.055 mm tangentially (one degree), and 0.005 mm in the radial direction. Based on this study, the use of 1 element/ply FE models for notched strength analyses [20] results in an inherent element size dependent "characteristic dimension" effect for some of the ply stress components.
BAU ON COMPOSITE BOLTED JOINTS

TABLE 2 — Mesh refinement at hole edge for converged peak ply stresses in notched tension

<table>
<thead>
<tr>
<th>Peak Ply Stress</th>
<th>Location</th>
<th>Mesh Required for Convergence</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\sigma_{11}$ tension</td>
<td>$0^\circ$ ply, at critical side of hole location (Fig 9)</td>
<td>one element/ply is adequate, not very sensitive to mesh refinement over 1 element/ply</td>
</tr>
<tr>
<td>$\sigma_{22}$ tension</td>
<td>$90^\circ$ ply at critical side of hole location</td>
<td>matrix interlayer is necessary</td>
</tr>
<tr>
<td>$\tau_{12}$</td>
<td>$0^\circ$ ply, ~12° above and below critical side of hole location</td>
<td>element should be ~ one fiber thickness in the radial direction</td>
</tr>
<tr>
<td>$\sigma_{33}$ tension</td>
<td>$90^\circ$ ply at critical side of hole location</td>
<td>did not obtain convergence with FEA model with 8 elements per ply</td>
</tr>
<tr>
<td>$\tau_{13}$</td>
<td>matrix interlayer at ±45°/0° ply interfaces ~12° above &amp; below critical side of hole loc</td>
<td>should model matrix interlayer + &gt;2 elements through-the-thickness of each composite ply</td>
</tr>
<tr>
<td>$\tau_{23}$</td>
<td>matrix interlayer at ±45° ply interfaces ~30° above &amp; below critical side of hole loc</td>
<td>should model matrix interlayer + &gt;2 elements through-the-thickness of each composite ply</td>
</tr>
</tbody>
</table>

Linear and nonlinear FEA were conducted for the UDRI open hole tension and compression configurations using mesh refinements determined from the sensitivity study. A simple failure criteria was applied to FEA results: the axial and shear ply stress components (without interaction) were compared to the appropriate ply strengths to obtain predictions of matrix failure locations and the critical coupon loads. Ply strengths were obtained from uniaxial ply level tests using the same batch of material as the notched tests. The predicted matrix failures from the linear and nonlinear FEA were compared to the initial damage types and loads from the incremental notched tests.

The linear FEA with ply strength criteria provided poor correlation with test data. At initial damage loads, this method predicted many locations around the hole which had ply stresses which were higher than the corresponding ply strengths. This may be due to nonlinear behavior at initial damage loads and/or in-situ strengths which are higher than the ply strengths from standard coupon tests. While the linear analyses and ply strength criteria predicted matrix damage in many erroneous locations, in some cases, the predicted ply stresses were very low in all directions at the actual initial damage locations.

The nonlinear FEA included nonlinear 3D shear properties in the composite elements, where stiffnesses in the fiber direction were linear, but transverse and shear stiffnesses were nonlinear. The nonlinear material properties were based on stress-strain results from ±45° axial tension tests. Nonlinear isotropic properties were used for the matrix interlayer elements. Residual thermal effects were included in the nonlinear models. Nonlinear geometric effects were determined to be insignificant at initial damage loads, so were not included in the analyses. Ply stresses were evaluated instead of ply strains for all nonlinear results to correctly account for the thermal effects. The Abaqus FE code was used for the nonlinear analyses.

In the linear FEA at initial damage loads, there were many locations where the predicted ply shear stresses were much higher than the ply shear strengths. The nonlinear material properties resulted in substantial stress redistribution at these locations. The nature of the stress redistribution was complex, with changes in stress primarily near the regions which had nonlinear behavior. Depending on the ply stress
component and the layup, the peak ply stresses from the nonlinear models were either higher or lower than peak stresses from the linear models.

Although there was substantial stress redistribution, the nonlinear analyses consistently predicted in-plane ply transverse and shear stresses which were higher than the ply strengths at various locations around the hole. However, with the non-interactive ply stress criteria, the nonlinear analyses resulted in fewer erroneous predicted damage locations than the linear FE analyses and better correlation between peak stresses and matrix crack initiation locations.

The phenomenon where predicted ply stresses are higher than the ply strengths at initial damage loads is similar to the conservatism encountered with predicting ultimate strengths in notched composites. Although there are many possible causes for the conservatism in the nonlinear initial damage predictions, substantial effort was made in this study to obtain robust initial damage loads from tests and accurate nonlinear stress fields at the hole edge. The simple ply strength failure criteria used in this study is inherently less conservative than any of the interactive ply-stress-based failure criteria which are typically used by researchers. Therefore, it was concluded that the ply strength values determined from the unidirectional coupon tests may be too conservative.

Evidence in composites literature tends to support this conclusion. Test data \[21\] show that there are in-situ effects on axial compression and transverse tension ply strengths which are related to thickness and stacking sequence which cannot be explained through detailed stress analysis. Hole size effects on notched strength have been attributed to larger volumes of highly stressed materials increasing the probability of flaws \[1\]. This type of information suggests that the ply strengths derived from unidirectional unnotched tests may be too conservative to be used to accurately predict the initiation of matrix damage with 3D FEA at stress concentrations. However, these effects have not yet been adequately quantified. Further investigation of these effects and the other possible causes will be necessary before failure criteria can be backed out of the nonlinear FE analyses and incremental test data.

Conclusions

Although uniaxially loaded notched composites generally have net section failure modes, various failure mechanisms result in different ultimate strength trends. Four ultimate strength trends were identified in open hole specimens with 0° fiber failure dominated laminates: delamination, 0° tangential matrix cracking, constant strain, and a basic Kt related trend. Each trend is associated with a different type of sub-critical matrix damage which effects the critical 0° ply fiber stress.

Nonlinear FEA with non-interactive ply strength criteria and ply strengths derived from unidirectional coupon tests consistently resulted in conservative initial damage predictions. Since the nonlinear analyses and the initial damage loads from tests were considered to be reasonably accurate, and the failure criteria were considered to be unconservative, the conservatism was attributed to inaccuracies in the ply strength values. Investigation of in-situ and scaling effects on ply strengths is required to resolve the initial damage prediction discrepancies.

Acknowledgments

The author would like to thank Stephen Ward of SW Composites and D. M. Hoyt of NSE Composites for their help in this work, and also Drs. Eric Nottorf, Sohan Singh,
Kathy Ferrie, and Carl Rousseau (formerly) of Bell Helicopter for their support of this continuing project.

References


Development of Compression Design Allowables for Composite Bolted Joints Using ASTM Standard D 6742


Abstract: Compression design allowables for carbon/epoxy laminates could be increased by taking advantage of hole-filling effects, if reliable design data were available. Previous research had demonstrated that filled hole compression strengths range between 5% to 35% higher than open hole strengths, due to variability in fastener-hole clearance and clamp-up torque. The importance of measuring these parameters for proper data interpretation resulted in the development of ASTM Practice for Filled Hole Tension and Compression Testing of Polymer Matrix Composite Materials (D 6742/D 6742M). Specific instructions regarding measurement, recording, and calculation of critical parameters for filled hole specimens are provided. The standard was implemented at Boeing-Philadelphia in the development of compression allowables for high-temperature materials used on the RAH-66 airframe. Enhanced compression strengths were demonstrated for several material systems tested in a variety of environments. A semi-empirical analysis technique permitted strain allowables to be generated using fewer laminate configurations and specimens than in previous programs.

Keywords: composites, bolted joints, compression loading, hole tolerance, finite-element analysis, strength prediction, design allowables

Introduction

Historically, allowable strengths for compression-loaded carbon/epoxy laminates used in airframe structures have been generated using open hole specimens [1]. This is done because open hole coupon specimens provide a repeatable stress concentration, are simple and inexpensive to manufacture, and demonstrate conservative strengths relative to those for the filled hole condition. However, in the sizing of bolted joints, the assumption of the complete loss of a fastener leaving an open hole may be too conservative. Filled hole-based compression allowables could be used to increase structural efficiency, if reliable design data were available.

As shown in Figure 1, the filled hole strength enhancement (relative to open hole strength) results from the addition of a load path through the fastener. This additional load path reduces the load which must be carried around the hole, and subsequently reduces the stress concentration local to the hole edge. For a through-fastener load path to exist, the laminate must deform such that any clearance between the fastener and the hole surfaces is eliminated. The load share carried through the fastener, and subsequently the laminate filled hole compression strength, will vary depending upon the initial hole clearance and the ability of the laminate to deform and eliminate this clearance [2].
Filled hole compression allowables have not been widely used in airframe design, due to a lack of confidence in our understanding of geometry-based effects and failure mechanisms. Also, design/test criteria for the development of such allowables were lacking consensus. This paper summarizes a multi-year National Rotorcraft Technology Center (NRTC)/Rotorcraft Industry Technology Association (RITA) research program conducted at Boeing-Philadelphia which addressed these issues. The paper then discusses the subsequent development of ASTM Practice for Filled Hole Tension and Compression Testing of Polymer Matrix Composite Materials (D 6742/D 6742M). Additionally, the application of the new standard on the RAH-66 Comanche composite allowables program will be addressed.

**Filled Hole Compression Research Program**

**Static Filled Hole Compression Strength**

The initial research program (1996–1998) consisted of an assessment of factors presumed to affect static filled hole compression strength by influencing the formation of through-fastener load paths [3]. These included initial fastener-hole clearance, laminate stiffness, through-thickness restraint, and fastener torque.

Initial clearance results from hole manufacturing tolerances and elongation resulting from repeated bearing loads. Clearance fit fastener holes used in composite primary structures are typically larger in diameter than the bolt diameter; for example Class 1 holes of 6.35 mm (0.250 in.) nominal diameter may range in actual diameter between 6.35 mm to 6.43 mm (0.253 in.). Thus, Class 1 structural holes may be up to 1.2% larger in diameter than the fastener at installation. As filled hole-based allowables for primary structures must account for hole clearances and/or elongation levels of this magnitude, hole diameters spanning this range were tested during the investigation.
The relative influence of the aforementioned parameters upon static compression strength was assessed using a matrix of 100 coupon-level open and filled hole compression specimens. Three laminate configurations (A, B and C, shown in Table 1) were used in the experiments. Specimens were manufactured using standard Boeing equipment and procedures, except that holes were drilled to precise diameters using drill reamers. Hole diameters were nominally 6.350, 6.375 or 6.426 mm (0.2500, 0.2510 or 0.2530 in.) with ±0.0076 mm (0.0003 in.) tolerance; the precise diameters permitted a meaningful examination of clearance effects. Filled hole specimens were tested using three fastener installation conditions, namely pure pin, finger tight (0.3 to 0.7 N-m, or 3 to 6 in.-lb, torque), or normal installation torque (9.7 N-m, or 85 in.-lb, torque).

TABLE 1 -- Laminate configurations.

<table>
<thead>
<tr>
<th>Laminate ID</th>
<th>Material</th>
<th>Stacking Sequence&lt;sup&gt;1&lt;/sup&gt;</th>
<th>% 0/45/90</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>T800/3900-2 Tape</td>
<td>[45/90/-45/03/±45/03/±45 ]&lt;sub&gt;s&lt;/sub&gt;</td>
<td>50/42/08</td>
</tr>
<tr>
<td>B</td>
<td>T800/3900-2 Tape</td>
<td>[45/90/-45/04/45/05/-45 ]&lt;sub&gt;s&lt;/sub&gt;</td>
<td>62/29/09</td>
</tr>
<tr>
<td>C</td>
<td>T800/3900-2 Tape</td>
<td>[45/90/±45/03/±45 ]&lt;sub&gt;s&lt;/sub&gt;</td>
<td>33/56/11</td>
</tr>
<tr>
<td>D</td>
<td>IM6/3501-6 Tape</td>
<td>[45/90/-45/03/±45/03/±45 ]&lt;sub&gt;s&lt;/sub&gt;</td>
<td>50/42/08</td>
</tr>
</tbody>
</table>

<sup>1</sup> Overscore indicates plies are not included in laminate symmetry.

Open and filled hole coupon specimens were manufactured to the configurations shown in Figure 2. Specimens were fabricated with two holes to permit a post-failure examination of the damage state at the hole where ultimate failure did not occur, such that non-catastrophic damage modes could be characterized. Tests were conducted in room temperature, ambient humidity conditions using fixtures and procedures analogous to those specified in ASTM Test Method for Open-Hole Compressive Strength of Polymer Matrix Composite Laminates (D 6484/D 6484M).

Hole diameters are 6.350 mm (0.2500 in.), 6.375 mm (0.2510 in.), or 6.426 mm (0.2530 in.) ± 0.0076 mm (0.0003 in.) as specified. Filled hole specimens use a Boeing BACB30VT8K pin and BACC30CC8 collar at each hole.

FIG. 2 -- Open and filled hole specimen configurations.
Mean filled hole failure strains for the various hole conditions and diameters tested are compared with mean open hole strains in Table 2. A representative graph presenting the effects of the through-fastener load path, initial bolt-hole clearance, through-thickness deformation restraint, and fastener torque upon far-field compression failure strain is shown in Figure 3. The height of the shaded bars represents the mean failure strain for a given set of specimens (normalized relative to the open hole strain), and the error bars represent ± one standard deviation.

Mean failure strains for 6.350 mm and 6.375 mm (0.2500 in. and 0.2510 in.) diameter filled hole specimens were very similar, regardless of the fastener condition or torque. Conversely, mean failure strains for specimens containing 6.426 mm (0.2530 in.) diameter filled holes were consistently lower than those for 6.350 mm and 6.375 mm diameter filled hole specimens. Failure strain enhancement caused by the presence of the fasteners ranged between 9–26% for specimens with 6.426 mm diameter holes (relative to open hole failure strains), versus 18–37% for specimens containing 6.350 mm and 6.375 mm diameter holes. Initial clearance had the greatest influence upon filled hole-related strength enhancement.

### TABLE 2 -- *Open and filled hole compression test results.*

<table>
<thead>
<tr>
<th>Laminate ID</th>
<th>Hole Condition</th>
<th>Nominal Hole Diameter [mm]</th>
<th>Percent of Mean Open Hole Strain [%]</th>
</tr>
</thead>
<tbody>
<tr>
<td>A</td>
<td>Open Hole</td>
<td>6.350</td>
<td>100</td>
</tr>
<tr>
<td></td>
<td>Filled Hole, Pin</td>
<td>6.350</td>
<td>122</td>
</tr>
<tr>
<td></td>
<td></td>
<td>6.375</td>
<td>124</td>
</tr>
<tr>
<td></td>
<td></td>
<td>6.426</td>
<td>116</td>
</tr>
<tr>
<td></td>
<td>Filled Hole, Finger Tight</td>
<td>6.350</td>
<td>134</td>
</tr>
<tr>
<td></td>
<td></td>
<td>6.375</td>
<td>137</td>
</tr>
<tr>
<td></td>
<td></td>
<td>6.426</td>
<td>122</td>
</tr>
<tr>
<td></td>
<td>Filled Hole, Normal Installation</td>
<td>6.350</td>
<td>133</td>
</tr>
<tr>
<td></td>
<td></td>
<td>6.375</td>
<td>135</td>
</tr>
<tr>
<td></td>
<td></td>
<td>6.426</td>
<td>126</td>
</tr>
<tr>
<td>B</td>
<td>Open Hole</td>
<td>6.350</td>
<td>100</td>
</tr>
<tr>
<td></td>
<td>Filled Hole, Pin</td>
<td>6.375</td>
<td>122</td>
</tr>
<tr>
<td></td>
<td></td>
<td>6.426</td>
<td>111</td>
</tr>
<tr>
<td></td>
<td>Filled Hole, Norm. Installation</td>
<td>6.375</td>
<td>118</td>
</tr>
<tr>
<td></td>
<td></td>
<td>6.426</td>
<td>109</td>
</tr>
<tr>
<td>C</td>
<td>Open Hole</td>
<td>6.350</td>
<td>100</td>
</tr>
<tr>
<td></td>
<td>Filled Hole, Pin</td>
<td>6.375</td>
<td>119</td>
</tr>
<tr>
<td></td>
<td></td>
<td>6.426</td>
<td>110</td>
</tr>
<tr>
<td></td>
<td>Filled Hole, Norm. Installation</td>
<td>6.375</td>
<td>130</td>
</tr>
<tr>
<td></td>
<td></td>
<td>6.426</td>
<td>123</td>
</tr>
</tbody>
</table>
Failure strains were similar when fasteners were installed either finger tight or at normal installation torque. This indicates that through-thickness preload had relatively little influence upon the failure mechanisms, at least in room temperature conditions. Conversely, mean filled hole failure strains increased dramatically when finger tight or normal torque conditions were present (between 6–13%), compared to the pure pin data. Therefore, it was concluded that restraint of laminate through-thickness deformation (caused by the presence of the fastener head and collar) had a significant influence upon filled hole compression failure. This effect was not noted for laminate configuration B, which indicated a change in deformation behavior or failure mode for this relatively stiff laminate.

Representative strain gage readings taken local to the open and filled holes are presented in Figure 4. The figure compares local strains near open or filled (pin) holes of 6.350, 6.375 or 6.426 mm (0.2500, 0.2510 or 0.2530 in.) nominal drilled diameter measured throughout the tests. Specimens containing 6.350 mm diameter filled holes exhibited reduced local strains at all far-field strain levels, indicated that a through-fastener load path existed at the onset of loading. Measured strains local to 6.375 mm and 6.426 mm diameter filled holes diverged from open hole behavior at 2000–2500 με far field strain for the 6.375 mm diameter specimens, and at 3000–4000 με for the 6.426 mm diameter specimens.

It should be noted that the laminate configurations tested were unsymmetric local to the laminate midplane. The use of such configurations did not appear to significantly affect the test results, based upon a comparison of the observed failure modes with those historically observed for symmetric layups.
During the second phase of the program (1997–98), the influence of clearance upon the compression bearing-bypass strength of bolted joints was assessed using a matrix of 39 coupon-level specimens [4]. One laminate configuration (D, shown in Table 1) was used in the experiments, which were conducted in room temperature, ambient humidity conditions. Specimen configurations and hole diameters were similar to those used in the first phase, except that the specimens were 305 mm (12.0 in.) long and contained one centrally located hole. Fasteners were installed in a “finger tight” condition (0.3 to 0.7 N-m, or 3 to 6 in.-lb, torque).

Bearing-bypass loads were applied using the test system shown in Figure 5. This apparatus was previously used in structural allowables testing for the Boeing 777 aircraft. Bearing loads were introduced to the specimen through the bearing-reaction plates by differentiating the deflection (and thus the loading) of the hydraulic actuators. The presence of the bearing-reaction plates stabilized the specimen under compression loads. Once installed in the test apparatus, each specimen was loaded in longitudinal compression until final failure, under a constant percentage of load transferred at the fastener.

Representative bypass-dominated specimen compression failure modes are shown in Figure 6. The majority of bypass-dominated specimens (0–25% load transfer) typically failed in offset net section compression modes (characterized by through-section fractures emanating from the hole near the bearing-contact zones, or from surface damage originating at the edges of fixture bushings). However, 0% load transfer specimens containing 6.426 mm (0.2530 in.) holes failed in a net section compression mode (characterized by through-section fractures emanating from the hole at or near the
location of peak bypass stress concentration). Thus, the small variance in initial clearance was responsible for a change in failure mode.

FIG. 5 -- Bearing-bypass loading apparatus.

Failure data for specimens containing 6.350 and 6.426 mm (0.2500 and 0.2530 in.) holes are compared in Table 3. The greatest variance in performance was observed in the pure bypass case (0% load transfer), in which the mean failure strain of specimens with 6.426 mm holes was 93% of that obtained for specimens with 6.350 mm holes. It is notable that this was the one load case in which different failure modes were observed. Strength variances due to initial clearance were typically less than 3% at other bearing-
bypass load ratios. An additional finding of note is that slight inaccuracies and variances in load transfer calculations do not significantly affect the total load-carrying capability of joints under compression bypass-dominated loading.

TABLE 3 -- Hole clearance effects upon mean bolted joint compression strength.

<table>
<thead>
<tr>
<th>Percentage Load Transfer at Fastener</th>
<th>Strength with 6.426 mm Hole/Strength with 6.350 mm Hole</th>
</tr>
</thead>
<tbody>
<tr>
<td>0%</td>
<td>0.93</td>
</tr>
<tr>
<td>10%</td>
<td>1.00</td>
</tr>
<tr>
<td>15%</td>
<td>1.01</td>
</tr>
<tr>
<td>25%</td>
<td>0.94</td>
</tr>
<tr>
<td>50%</td>
<td>0.98</td>
</tr>
<tr>
<td>100%</td>
<td>0.98</td>
</tr>
</tbody>
</table>

Fatigue Strength

The third phase of the program (1999–2000) investigated the influence of hole filling and clearance upon the fatigue behavior of composite laminates [5]. One laminate configuration (D, shown in Table 1) was used in the experiments, which were conducted in room temperature, ambient humidity conditions. Specimen configurations and hole diameters were similar to those used in the second phase, with the fasteners installed in a “finger tight” condition (0.3 to 0.7 N-m, or 3 to 6 in.-lb, torque).

Specimens were tested using constant amplitude control (sine wave) with a minimum/maximum load ratio of -1.0. Test frequency remained constant for each specimen, and ranged between 5 and 10 Hz, depending upon the variable load amplitude. A thermocouple was attached to each specimen to ensure its temperature did not exceed 49°C (120°F) during the test.

Two “modes” of failure were observed in the experiments: net section failure (all open hole specimens and half of the 6.426 mm (0.2530 in.) diameter filled hole specimens) and offset net section failure (all 6.350 mm (0.2500 in.) diameter filled hole specimens, and the remaining 6.426 mm filled hole specimens). Several specimens exhibited extensive transverse tension and shear fractures.

Filled hole fatigue strain capability is normalized relative to open hole capability for a given number of cycles in Figure 7. Run-out data points are indicated by arrows. A nearly constant 11% enhancement in compressive strain capability was observed for the 6.426 mm (0.2530 in.) diameter filled hole specimens between 5000–300000 cycles. Notably, this value is equivalent to that obtained in the static tests. Conversely, strain enhancements for the 6.350 mm (0.2500 in.) diameter filled hole specimens ranged between 29% at 1000 cycles to 13% at 3000000 cycles. Based upon this comparison, it was concluded that no significant degradation in fatigue performance was caused by fastener-hole contact and bearing stresses in the filled hole specimens; rather, fatigue performance was enhanced by fastener presence.
Based upon the significant progress and findings generated under the multi-phase NRTC/RITA project, the author began work on a new standard for filled hole specimen testing. As standards existed for open hole tension (ASTM D 5766/D 5766M, Test Method for Open-Hole Tensile Strength of Polymer Matrix Composite Laminates) and open hole compression (D 6484) testing, it was decided to limit the new standard to those additional measurements, procedures and interferences critical to filled hole tension and compression testing. The standard was eventually approved as ASTM Practice D 6742 in October 2000, providing procedures to modify D 5766 and D 6484 for filled hole specimen testing. D 6742 was first published in the spring of 2001, and was subsequently modified in October 2002 to address changes identified during the re-approval of D 5766.

Interferences

A key attribute of D 6742 is the description of critical interferences (sources of variability) observed during filled hole testing. Based upon the NRTC/RITA findings, it was decided that an extensive discussion of fastener-hole clearance effects was of primary importance. The standard quantifies the significant variability in compression failure mode and strength that can be caused by a 25-μm [0.001-in.] change in clearance, necessitating that both the hole and fastener diameters be accurately measured and recorded. A typical aerospace tolerance on fastener-hole clearance (+75/-0 μm, or +0.003/-0.000 in.) is provided. Notes on clearance effects under tensile loading and fastener interference are also provided, and reference is made to the aforementioned NRTC/RITA research (Reference 3) as a source of additional information.
The practice next addresses differences in failure load and mode caused by changes in fastener preload, and that the critical preload conditions can vary depending upon the type of loading, the material system, laminate stacking sequence, and test environment. Historically critical hole and preload conditions for both tension and compression loading are provided, along with historical references and sources of data. Additionally, the interferences address fastener type, hole preparation procedures, environment (providing historically critical conditions for both tension and compression loading), and geometric effects related to the use of countersunk fasteners.

Filled Hole-Specific Specimen Details, Measurements and Procedures

Test apparatus and specimen geometry are generally in accordance with ASTM D 5766 for tension tests and D 6484 for compression tests. In regard to specimen geometry, the practice provides a nominal fastener diameter (6 mm, or 0.25 in.), and requires that the fastener type, installation torque, and washer details (type, number and locations) be specified as initial parameters and reported. For torqued fasteners, the torque wrench used to tighten the fastener was specified as being capable of determining the applied torque to within ±10% of the desired value, for consistency with ASTM Test Method for Bearing Response of Polymer Matrix Composite Laminates (D 5961/D 5961M). The reuse of fasteners is discouraged because of potential differences in through-thickness clamp-up for a given torque level, caused by wear of the threads.

Based upon the sensitivity of compression failure mode and strength to fastener-hole clearance, the practice requires that the micrometer or gage used shall be capable of determining the hole and fastener diameters to ±8 μm (±0.0003 in.). This is equivalent to the tolerance range successfully used to ensure that consistent hole diameters were tested during the NRTC/RITA project. Although filled hole tension strength is not as sensitive to clearance as is compression strength, the same tolerance range was specified for tension testing for consistency.

Test procedures are generally in accordance with ASTM D 5766 for tension tests and D 6484 for compression tests, with additional measurements for fastener diameter, countersink depth, and countersink flushness required. Cleaning, lubrication, fastener installation and torquing are to be conducted after specimen preparation and preconditioning.

Data Interpretation and Reporting

As shown in Table 4, the practice expands upon the failure mode codes listed in D 5766 and D 6484, adding those modes identified during the NRTC/RITA project. The new codes include failures offset from the center of the hole (LGO, MGO) and failures induced at the fastener, nut or washer edge (LGF, MGF), as shown in Figure 8.

In addition to the strength and width/diameter ratio calculations specified in ASTM D 5766 and D 6484, the practice requires calculation and reporting of the specimen's diameter-to-thickness ratio and countersink depth-to-thickness ratio. The report requires information on the location of the fastener head (bag side versus tool side surface), washer type and material, washer location, number of washers, cleaning process,
lubricant, measured hole and fastener diameters, hole preparation and fastener installation procedures, countersink angle, countersink depth, and countersink flushness.

TABLE 4 -- ASTM D 6742/D 6742M failure mode codes.

<table>
<thead>
<tr>
<th>First Character</th>
<th>Code</th>
<th>Second Character</th>
<th>Code</th>
<th>Third Character</th>
<th>Code</th>
</tr>
</thead>
<tbody>
<tr>
<td>Angled</td>
<td>A</td>
<td>inside grip/tab</td>
<td>I</td>
<td>bottom</td>
<td>B</td>
</tr>
<tr>
<td>Edge delamination</td>
<td>D</td>
<td>at grip/tab</td>
<td>A</td>
<td>top</td>
<td>T</td>
</tr>
<tr>
<td>Grip/tab</td>
<td>G</td>
<td>&lt;1 w from grip/tab</td>
<td>W</td>
<td>left</td>
<td>L</td>
</tr>
<tr>
<td>Lateral</td>
<td>L</td>
<td>gage</td>
<td>G</td>
<td>right</td>
<td>R</td>
</tr>
<tr>
<td>Multimode</td>
<td>M</td>
<td>multiple areas</td>
<td>M</td>
<td>middle, center of hole</td>
<td>M</td>
</tr>
<tr>
<td>Long, splitting</td>
<td>S</td>
<td>various</td>
<td>V</td>
<td>offset from center of hole</td>
<td>O</td>
</tr>
<tr>
<td>Explosive</td>
<td>X</td>
<td>unknown</td>
<td>U</td>
<td>offset of fastener edge</td>
<td>F</td>
</tr>
<tr>
<td>Other</td>
<td>O</td>
<td>unknown</td>
<td>V</td>
<td>various</td>
<td>V</td>
</tr>
<tr>
<td></td>
<td></td>
<td>unknown</td>
<td>U</td>
<td>unknown</td>
<td>U</td>
</tr>
</tbody>
</table>

LGO
Laminate compressive failure laterally across the specimen at the fastener hole, but offset from the center of the hole (bearing or surface failure local to hole, followed by 0-degree ply dominated kinking/buckling). Splits and delaminations may be present.

MGO
Laminate fails in compression offset from the center of the hole and exhibits multiple modes of failure in various sublaminates. Extensive splitting and delamination present.

LGF
Laminate compressive failure laterally across the specimen offset from the hole, at the fastener, nut or washer edge (surface failure, followed by 0-degree ply dominated kinking/buckling). Splits and delaminations may be present.

MGF
Laminate fails in compression at the fastener, nut or washer edge and exhibits multiple modes of failure in various sublaminates. Extensive splitting and delamination present.

FIG. 8 -- Acceptable filled-hole compressive failure modes offset from center of hole.
Application of ASTM D 6742 on the RAH-66 Allowables Program

Criteria and Specimen Design

The first implementation of ASTM D 6742 at Boeing-Philadelphia occurred in 2001-02, during the Engineering and Manufacturing Development (EMD) phase of the RAH-66 Comanche program. It became necessary to impose high temperature design environments of 232°C (450°F) with ambient humidity, and 191°C (375°F) for “wet” (moisture conditioned) laminates, for material systems used in the airframe’s tailcone. Subsequently, design allowables programs for several bismaleimide (BMI) composite systems were initiated, and D 6742 was used in the generation of design data for three laminate types. These included IM7/F655 tape-plain weave (PW) fabric hybrid laminates, IM7/5250-4 PW fabric laminates, and IM7/5250-4 RTM tape/PW fabric hybrid laminates.

NRTC/RITA research had demonstrated the sensitivity of filled hole compression failure strain to small variations in fastener-hole clearance, and that data generated for holes with 75 µm (0.003 in.) clearance were consistently lower than those for holes with less clearance. Accordingly, design data generated using specimens with holes of 75 µm clearance will be conservative for holes with less clearance. It was decided to assume that all Class I filled holes on RAH-66 structure are at maximum permissible bolt-hole clearance (75 µm per engineering specifications), and to generate design allowables using specimens which reflect that condition. This decision was supported by the use of Boeing statistics-based inspection procedures during the assembly of composite structures. Thus, RAH-66 filled hole compression specimens utilized the ASTM D 6742 configuration, using holes drilled to 6.426 ± 0.008 mm (0.2530 ± 0.0003 in.) diameter and with a nominal 75 µm clearance.

The filled hole compression design data are intended for use in margin of safety calculations under ultimate load conditions. For additional conservatism, open hole-based compression data were generated to demonstrate acceptable strength capability at limit load conditions, to satisfy fail-safety and ballistic requirements. It should be noted that the +75/-0 µm (+0.003/-0 in.) tolerance on clearance is retained for all standard fastener diameters, so that the “relative” clearance (compared to the hole diameter) permitted for fasteners larger than 6.35 mm (0.250 in.) decreases. Subsequently, the degree of straining required to overcome this clearance also decreases, making the data conservative for larger fastener diameters when adjusted using standard “hole size” factors. The use of 6.35 mm data is also conservative for fasteners of smaller diameter due to the “hole size” effect [6].

Filled hole compression specimens were tested using ASTM D 6742 specifications with one exception; flat platens were used to end-load the specimen and stabilization fixture rather than hydraulic grip loading. The end-loading procedure has recently been evaluated under an ASTM D30 round-robin testing exercise, has been shown to produce equivalent notched compression strength results, and will be added to D 6484 and D6742 in the near future. All filled hole specimens were fastened using BACB30VT8K pins (Fairchild VL10-8) and BACC30CC8 collars (Hi-Shear HST1571YN-8) torqued to nominally 4.0 N-m (35 in.-lb).
Experimental Results

Open and filled hole compression strain data for IM7/F655 tape/PW fabric hybrid laminates are shown in Figure 9; the 191°C (375°F) wet condition was found critical for this system. All data are shown normalized to the mean open hole regression strain for each value of AML (Angle Minus Loaded plies, equal to the percentage of ±45° fibers minus the percentage of 0° fibers). This permits a better understanding of the increase in failure strain caused by fastener hole-filling effects. Individual test data are shown along with mean and B-basis statistical regression lines.

For IM7/F655 tape/fabric hybrid laminates, the filled hole specimens demonstrated a 20–25% increase in mean failure strain compared to the open hole specimens. Due to the greater variability exhibited by the filled hole specimens, the resulting improvement in B-basis design strains was lower, at approximately 15%. The increased variation was caused by a greater variety of failure modes (offset at hole edge, offset at fastener edge, etc.) exhibited by the filled hole specimens.

Similar data for IM7/5250-4 PW fabric and IM7/5250-4 RTM tape/PW fabric hybrid laminates are shown in Figures 10 and 11; the 232°C (450°F) ambient condition was found critical for these systems. Filled hole specimens exhibited 19–33% increases in mean failure strain for the PW fabric laminates, and 16–22% increases for the hybrid laminates, compared to the open hole specimens. This resulted in a 22–37% improvement in B-basis design strains for the PW fabric laminates, and an 18–25% improvement for the hybrid laminates.

![FIG. 9 -- Comparison of open and filled hole compression failure strain behavior, IM7/F655 hybrid laminates, 191°C (375°F) wet conditions.](image-url)
FIG. 10 -- *Comparison of open and filled hole compression failure strain behavior, IM7/5250-4 plain weave fabric laminates, 232°C (450°F) ambient conditions.*

![Graph 1](image1)

FIG. 11 -- *Comparison of open and filled hole compression failure strain behavior, IM7/5250-4 hybrid laminates, 232°C (450°F) ambient conditions.*

![Graph 2](image2)
It is clear that significant (15–33%) increases in compression design strains were achieved using filled hole specimens for all three material systems. Design strain enhancements were exhibited throughout the range of laminate configurations (AML values) tested for all three systems. Also, as fail-safety requirements limit the filled hole strain enhancements only above 40%, the observed design strain improvements can be fully realized in the tailcone design.

Use of Semi-Empirical Analysis

Analysis methods and failure prediction methodologies developed under the NRTA/RITA project were used on the RAH-66 allowables program as a confirmation of failure strain versus AML behavior. The usefulness of similar finite element models and progressive damage analysis of composite joints has been demonstrated by Crews and Naik [2] and Chang [7]. An extensive description of the methods used is provided in References 3, 4 and 5; only key assumptions and methods are provided herein.

The finite element code used in this investigation was Samtech’s SAMCEF-BOLT Version 8.1 package [8], which consists of an automated finite element mesher, a non-linear material model, a failure model, and material property degradation rules for progressive failure analysis. In Version 8.1, the finite element model consists of a plate containing a hole, and an isotropic, elastic, frictionless pin in the hole. The plate mesh is composed of 2-D isoparametric quadrilateral membrane elements, while the pin mesh is composed of both triangular and quadrilateral elements. The analysis assumes that no through-thickness clamping pressure is applied, and does not account for stacking sequence effects. A key feature of the code is the ability to define the initial pin clearance by inputting separate hole and pin diameters. Contact is modeled using an iterative process, which prevents penetration of plate nodes within the pin boundary, and releases restrained nodes found to have positive contact reactions.

In the NRTC/RITA studies, the best compromise between predictive accuracy and computational efficiency was obtained when a 0.179 mm (0.0076 in.) element length local to the hole was used in the analysis. Models using larger elements (lower mesh densities) were found to be less accurate in predicting notched compression strength trends as a function of laminate configuration. Denser meshes required longer computation time, yet provided no significant improvement in predictive accuracy. The 0.179 mm element length was retained in the RAH-66 program.

A representative comparison of open and filled hole failure data with BOLT predictions for fiber failure are shown in Figure 12 for the IM7/F655 hybrid laminates. The predictions were based upon calibrations using the open and filled hole mean failure data at AML = -20, lamina strength data and a 0.179 mm (0.0076 in.) element length. BOLT predictions were found accurate to within ±3% of the experimentally obtained mean regression strains across the range of AML values tested.
This demonstration ultimately resulted in a reduction in allows-related testing costs of approximately 40% for the three material systems. As confidence in the ability to predict strain versus AML response increased, the number of laminate configurations tested was reduced to three for each material system; typically 5–8 configurations were required on previous allows test programs [1]. Subsequently, the number of specimens required to characterize a strength property was reduced to 20–25, for failure modes demonstrated consistent throughout the design space.

Conclusions

An NRTC/RITA project conducted at Boeing-Philadelphia resulted in an improved understanding of failure modes and strength properties observed in composite filled hole specimens and bolted joints loaded in compression. Fastener-hole clearance was identified as the key parameter affecting strength; variability up to 25% was observed for changes in clearance as small as 25 μm (0.001 in.). Fastener through-thickness restraint and clamp-up torque were also identified as parameters affecting filled hole compression strength.

Based upon this new information, ASTM practice D 6742/D 6742M was developed to provide consensus methods for determining filled hole tension and compression strengths for composite laminates. The new standard provides extensive interferences, measurements, procedures, and data analysis details, which can be used to modify ASTM
D 5766 (open hole tension) and D 6484 (open hole compression) to permit filled hole testing.

The new practice was successfully used to generate filled hole-based compression allowables on the RAH-66 program. A conservative filled hole specimen design was selected, such that the data generated are applicable to all fasteners installed per Boeing manufacturing procedures. Filled hole compression strength enhancements of 15–33% were demonstrated for high-temperature materials such as IM7/F655 and IM7/5250-4 bismaleimide laminates. Comparisons of the data with semi-empirical predictions demonstrated that strain allowables could generated across the required design space using fewer laminate configurations than in previous allowables programs, resulting in a significant (~40%) reduction in test-related costs.

Acknowledgments

Research tasks described in this document include tasks supported with shared funding by the U. S. Rotorcraft industry and government under the RITA/NASA Cooperative Agreement No. NCCW-0076, Advanced Rotorcraft Technology, Aug. 15, 1995, under WBS Nos. 97-7.1.6(2), 98-7.1.6(2), 99-03-7.1.6.1 and 00-B-03-7.1.6.1. Composite allowables development for the RAH-66 Comanche program was sponsored by the United States Army. The contributions of Integrated Technologies Corporation in specimen testing are greatly appreciated. The author also acknowledges Samtech S. A. for the development of the SAMCEF-BOLT code.
References


