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DESIGN OF JOINTS
FOR FILAMENT-
WOUND CYLINDERS

BDX-613-142

May, 1966

Prepared by:
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MASTER

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ABSTRACT

Analytical methods applied to design and evaluation of bonded scarf joints between a filament wound shell and a metal flange ring, indicate severe stress concentrations at joint ends unless thick adhesive layers are used. Joints must be very long to develop the strength potential of glass fiber composites. A joint utilizing thin metal shims between layers of glass fibers shows definite promise of developing high strengths.

Conventional analytical principles were employed and a computer was used to facilitate the required mathematical solutions. Destructive tests of prototype joints supported the design considerations.

FOREWORD

This report describes three types of metal joints in filament wound cylindrical shells and provides a method for calculating joint strength. The joint design method illustrated on p. 51 appears to be a practical method for which approximate joint strengths can be determined analytically. This report provides a method for computer solution of joints of this type; however, for approximations of the various parameters involved, the following manual method may be used. This manual method is for purposes of approximation; and until it is verified by additional testing, it should be used as a guide rather than for actual design.

Assumptions:

1. Neglect the strength of the hoop wrap layer.
2. Neglect the shear contribution at cup interfaces.

Calculations for strength per metal layer per inch of circumference follow.

Date Required:

- t_f = thickness in inches of longitudinal epoxy-fiber reinforced layer (on the test specimen this was .006).
- $\sigma_{m_{max}}$ = metal tensile strength in psi (the test specimen utilized aluminum with $\sigma_{m_{max}} = 75,000$).
- $\sigma_{f_{max}}$ = tensile strength in psi of longitudinal epoxy-fiber reinforced layer (for the test specimen, this value is 250,000 psi).

$\tau_{\max}$ = adhesive shear strength in psi (assumed $\tau_{\max} = 5,000$)

E_m = modulus of elasticity of the metal in psi ($E_m = 10.5 \times 10^6$)

E_f = modulus of elasticity of the epoxy-fiber matrix in psi
(assumed $E_f = 5 \times 10^6$)

Calculations:

$$t_m = \frac{\sigma_{f_{\max}}}{\sigma_{m_{\max}}} \quad t_f = \text{metal thickness}$$
$$= \frac{250,000}{75,000} (.006) = .020$$

$$T = \frac{t_m E_m}{t_f E_f} = \text{stress concentration factor}$$
$$= \frac{.020 (10.5 \times 10^6)}{.006 (5 \times 10^6)} = 7.00$$

ℓ = length of joint; the optimum length may be determined experimentally or by computer solution. For rough design, $\ell = \frac{FT}{\tau_{\max}}$. (ℓ for the test specimen was 1.71).

(If T is less than unity, use the inverse)

Strength of Joint:

$$F_{\text{ult}} = \frac{\tau_{\max}}{T} = \frac{5000 (1.71)}{7.00}$$

$F_{\text{ult}} = 1221$ pounds per metal layer per inch of circumference

Total Joint Strength:

$$P = F_{\text{ult}} (\text{number of metal layers}) (\text{mean circumference})$$
$$= 1221 (4) (\pi \times 5.9)$$
$$= 90,600 \text{ pounds}$$

Observed failure occurred at just over 90,000 pounds, and occurred as a shear failure of the aluminum rivets.

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Nomenclature

E	Modulus of Elasticity in Tension. PSI
F	Force Per Inch of Circumference, lbs
G	Modulus of Elasticity in Shear, PSI
ℓ	Joint Length, inches
m	t_m/ℓ
p	Pressure, lb/in. ²
r	x/ℓ
R	Radius, inches
S	Tensile Stress Concentration Factor, Dimensionless
T	Shear Stress Concentration Factor, Dimensionless
t	Material Thickness, inches
u	Displacement of Metal at Location X, inches
x	Location from end of Fiber Member, inches
ϵ	Tensile Strain, in/in.
γ	Shear Strain, Radians
δ	Displacement of Fiber Member at Location X, inches
σ	Tensile Stress, PSI
τ	Shear Stress, PSI
ϕ	"Function of"
θ	An Angle

Subscripts

f	Refers to Epoxy-Fiber Member
m	Refers to Metal Member
a	Refers to Adhesive Layer

I INTRODUCTION

The high tensile strength of small diameter glass fibers is being used to advantage in pressure vessels and rocket motor cases: The fibers are positioned as nearly as possible parallel to directions of principal stress. Modifications of shell membrane theory serve as a basis of design analysis.

Application of filament winding to structures other than pressure vessels may require joints or connections between component parts. Design of an efficient joint of reasonable weight and size is difficult, inasmuch as the resin of the filament-resin composite is very much weaker than the filaments or a metal member to which connection is required.

This study is concerned with the design evaluation of a bonded scarf-joint between a filament-wound cylindrical shell and a metal flange ring. Loading is limited to force parallel to the axis of the shell.

II THE DESIGN PROBLEM

A typical glass fiber-epoxy composite will exhibit an ultimate tensile strength (parallel to the fiber direction) of 250,000 PSI or more, whereas the resin itself may have a strength in shear in the order of 4000-6000 PSI, and the adhesive shear strength (along a metal surface) may be even lower. The modulus of elasticity of a fiber-epoxy composite will be in the order of one-sixth of the corresponding value for steel. A joint which is intended to transfer load from the high-strength low-modulus fiber-composite to a metal member must provide a large shear area. Differences in elastic constants and thermal expansion coefficients produce severe stress gradients in the bonded materials, and consequent high shear stress levels at the bond surface(s).

It has been recognized that an elastic adhesive layer which separates the two connected elements may be employed to "absorb" differences in thermoelastic properties, although the required adhesive layer thickness and fabrication control appear to limit the practical utilization of this concept.

III CONCLUSIONS

This study has developed an analytical means of evaluating the strength of adhesive-bonded joints. Numerical solutions indicate severe stress concentrations at the ends of scarf joints unless thick adhesive layers are employed. Scarf joints must be very long to develop the strength potential of glass fiber composites. Of the joints tested, the joint using thin metal shims between layers of glass fibers indicated the greatest promise in developing high strengths.

IV ANALYSIS: No Adhesive Layer

Consider a joint (Figure 1a) connecting two members, in which the load is transferred by a shear stress at the member interface. Assume:

1. That the elements behave elastically.
2. That bending effects may be neglected.
3. That the tensile stress is constant across the cross section of an element at any location;

then it may be observed that the external load per inch of shell circumference is equal to the sum of the loads in the two members.

$$F = \sigma_f t_f + \sigma_m t_m \quad (4.1)$$

Since the strains in the two elements must be identical (there is no "slip" at the interface),

$$\frac{\sigma_m}{E_m} = \frac{\sigma_f}{E_f} \quad (4.2)$$

Figure 1a

Joint Between Two Members - Load Transferred
by Shear Stress at Member Interface

Figure 1b

Short Section of One Element Isolated in Equilibrium
Under Influence of Stress

Combination of 4.1 and 4.2 yields:

$$\sigma_f = \frac{F}{\frac{E_m}{E_f} t_m + t_f} \quad (4.3)$$

$$\sigma_m = \frac{F}{t_m + \frac{E_f}{E_m} t_f} \quad (4.4)$$

The above two relations would indicate that the stress (in either member) is less at any point interior to the joint, than would be the stress in the single element exterior to the joint if material thicknesses are constant.

Figure 1b shows a short section of one element isolated in equilibrium under the influence of stress. Force balance requires:

$$\sigma_m t_m = (\sigma_m + d\sigma_m) (t_m + dt_m) + \tau dx \quad (4.5)$$

Expanding and discarding a second order term gives:

$$-\tau = \frac{d(\sigma_m t_m)}{dx} \quad (4.6)$$

Equation 4.6 states that the shear stress at the interface is equal to the gradient of the force in the elements. If an abrupt change exists in the thickness of the element, the shear stress will be very high: Extreme stress concentration exists at the shear interface unless the material thickness is gradually tapered to zero. (An attempt to better evaluate the interface shear stress is presented in Section V, and depends on an assumed adhesive layer of finite thickness.)

In order to avoid high shear-stress concentration a tapered scarf joint as shown in Figure 2 may be proposed. If the taper angle θ is small,

Figure 2

Cross-section of Tapered Splice

equations 4.3, 4.4, and 4.6 may be applied. Stress distributions as found by these equations for this joint (with linear taper in thicknesses) are graphed in Figure 3. For this configuration,

$$t_m = t_1 \left(1 - \frac{x}{l}\right) \quad (4.7)$$

$$t_f = t_2 \left(\frac{x}{l}\right) \quad (4.8)$$

Combination of 4.3, 4.7 and 4.8 gives

$$\sigma_f = \frac{F}{\frac{E_m}{E_f} t_1 \left(1 - \frac{x}{l}\right) + t_2 \left(\frac{x}{l}\right)}$$

Which may be written as

$$S_f = \frac{1}{\frac{t_1 E_m}{t_2 E_f} \left(1 - \frac{x}{l}\right) + \left(\frac{x}{l}\right)} = \frac{\sigma_f}{F/t_2} \quad (4.9)$$

The ratio $\frac{\sigma_f}{F/t_2}$ is a dimensionless ratio, and may be considered to be a "stress concentration factor". Note that if $\frac{t_1 E_m}{t_2 E_f} = 1$, the factor = 1. Similarly,

$$S_m = \frac{\sigma_m}{F/t_1} = \frac{1}{\left(1 - \frac{x}{l}\right) + \frac{t_2 E_f}{t_1 E_m} \left(\frac{x}{l}\right)} \quad (4.10)$$

The shear stress found as follows: From 4.7 and 4.10

$$\sigma_m t_m = \left(\frac{F}{t_1}\right) \frac{t_1 \left(1 - \frac{x}{l}\right)}{\left(1 - \frac{x}{l}\right) + \frac{t_2 E_f}{t_1 E_m} \left(\frac{x}{l}\right)}$$

$$\text{or, } \sigma_m t_m = \frac{F \left(1 - \frac{x}{\ell}\right)}{\left(1 - \frac{x}{\ell}\right) + \frac{t_2 E_f}{t_1 E_m} \left(\frac{x}{\ell}\right)} \quad (4.11)$$

From 4.6 and 4.11

$$\tau = \frac{d(\sigma_m t_m)}{dx} = F \frac{d}{dx} \left[\frac{\left(1 - \frac{x}{\ell}\right)}{\left(1 - \frac{x}{\ell}\right) + \frac{t_2 E_f}{t_1 E_m} \left(\frac{x}{\ell}\right)} \right]$$

Which becomes

$$\tau = \frac{F}{\ell} \frac{\frac{t_2 E_f}{t_1 E_m}}{\left[\left(1 - \frac{x}{\ell}\right) + \frac{t_2 E_f}{t_1 E_m} \left(\frac{x}{\ell}\right) \right]^2}$$

Or, considering $\frac{\tau}{F/\ell}$ to be a "shear stress concentration factor",

$$\frac{\tau}{F/\ell} = \frac{\frac{t_2 E_f}{t_1 E_m}}{\left[\left(1 - \frac{x}{\ell}\right) + \frac{t_2 E_f}{t_1 E_m} \left(\frac{x}{\ell}\right) \right]^2} \quad (4.12)$$

Note that for $\frac{t_2 E_f}{t_1 E_m} = 1$, this factor = 1.

The plot is for $E_m/E_f = 6$ and $\frac{t_2}{t_1} = \frac{180,000}{250,000}$. The maximum shear stress concentration factor exists at $x/\ell = 1$, and is equal to $\frac{t_1 E_m}{t_2 E_f}$. The metal has an equal stress concentration. The strength of this joint approaches the strength of the material external to the joint as the design length is increased.

$$\frac{E_m}{E_f} = 6$$

$$\frac{t_1}{t_2} = \frac{250,000}{180,000}$$

$$\sigma_{f, \text{ult}} = 250,000$$

$$\tau_{\text{ult}} = 10,000$$

$$EFF_a = 1.0 \text{ for } \frac{t_1}{t_2} = 208$$

Figure 3

Stress Concentration Factors for
a Scarf Joint of Small Angle

Figure 4a

Stress Distributions in Tapered Joint

Figure 4b

In Metal Element

Figure 4c

In Adhesive Element

Figure 4d

In Fiber Element

V ANALYSIS: With Adhesive Layer

Figure 4a presents a joint in which an elastic adhesive layer transfers load from one member to the other through shear. Stress distribution within this joint is found in the following analysis. (The adhesive and fiber layers are of constant thickness).

Figures 4b, c, and d present free-body elements of the metal, adhesive, and fiber members. For the fiber element, force equilibrium requires:

$$\sigma_f t_f + \tau(dx + du) = (\sigma_f + d\sigma_f) (t_f) \quad (5.1)$$

Expanding and dividing by dx gives:

$$\tau \left(1 + \frac{du}{dx}\right) = t_f \frac{d\sigma_f}{dx}$$

Since $\frac{du}{dx}$ is much less than one,

$$\tau = t_f \frac{d\sigma_f}{dx} \quad (5.2)$$

The shear stress τ at the surfaces of the adhesive layer may be related to the displacements u and ζ of a point at unstressed location x by considering the shear strain in the adhesive layer. Hooke's law for shear deformation is

$$\tau = \gamma G \quad (5.3)$$

and the shear stress gradient is

$$\frac{d\tau}{dx} = G \frac{d\gamma}{dx} \quad (5.4)$$

From Figure 4c, at location x ,

$$\gamma_x = \frac{u - \zeta}{t_a} \quad (5.5)$$

and at location $x + dx$

$$\gamma_{(x + dx)} = \frac{(u + du) - (\zeta + d\zeta)}{t_a} \quad (5.6)$$

Subtracting 5.5 from 5.6 and dividing by dx ,

$$\frac{\gamma_{(x + dx)} - \gamma_x}{dx} = \frac{d\gamma}{dx} = \frac{1}{t_a} \left(\frac{du}{dx} - \frac{d\zeta}{dx} \right) \quad (5.7)$$

Combining 5.7 and 5.4,

$$\frac{d\tau}{dx} = \frac{G}{t_a} \left(\frac{du}{dx} - \frac{d\zeta}{dx} \right) \quad (5.8)$$

The tensile strain in the metal member is

$$\epsilon_m = \frac{\sigma_m}{E_m} = \frac{d\zeta}{dx} \quad (5.9)$$

and the tensile strain in the fiber member is

$$\epsilon_f = \frac{\sigma_f}{E_f} = \frac{du}{dx} \quad (5.10)$$

Substituting 5.9 and 5.10 into 5.8,

$$\frac{d\tau}{dx} = \frac{G}{t_a} \left(\frac{\sigma_f}{E_f} - \frac{\sigma_m}{E_m} \right) \quad (5.11)$$

From 5.2,

$$\frac{d\tau}{dx} = t_f \frac{d^2 \sigma_f}{dx^2} \quad (5.12)$$

Combining 5.11 and 5.12

$$t_f \frac{d^2 \sigma_f}{dx^2} = \frac{G}{t_a} \left(\frac{\sigma_f}{E_f} - \frac{\sigma_m}{E_m} \right) \quad (5.13)$$

σ_m may be eliminated from the last equation by use of a gross equilibrium requirement:

$$F = \sigma_f t_f + \sigma_m t_m \quad (5.14)$$

or,

$$\sigma_m = \frac{F - \sigma_f t_f}{t_m} \quad (5.15)$$

It follows that:

$$\frac{d^2 \sigma_f}{dx^2} - \frac{G}{t_a t_f} \left(\frac{1}{E_f} + \frac{t_f}{t_m E_m} \right) \sigma_f = - \frac{GF}{t_a t_f t_m E_m} \quad (5.16)$$

Complete solution of the last equation includes incorporation of the boundary conditions:

$$\sigma_f = 0 \text{ at } x = 0 \quad (5.17)$$

$$\sigma_f = F/t_f \text{ at } x = \ell \quad (5.18)$$

A dimensionless form of 5.16 has advantage in that a solution for given proportions is independent of gross size. In dimensionless form, the stress variable may be considered to be a "stress concentration factor."

$$\text{let } \frac{\sigma_f}{F/t_f} = S_f$$

$$x/\ell = r$$

$$t_a/\ell = l_a$$

$$t_f/\ell = l_f$$

$$G/E_f = E_1$$

$$G/E_m = E_2$$

$$t_m/\ell = m, \text{ which is a function of } r$$

Equation 5.16 may be written as

$$\frac{d^2 S_f}{dr^2} - \frac{1}{l_a l_f} \left(E_1 + \frac{l_f}{m} E_2 \right) S_f = \frac{-E_2}{l_a m} \quad (5.19)$$

Which is of the form

$$\frac{d^2 S_f}{dr^2} - \phi_1(r) S_f = \phi_2(r) \quad (5.20)$$

Where

$$\phi_1(r) = \frac{E_1}{l_a l_f} + \frac{E_2}{l_a m} \quad (5.21)$$

and

$$\phi_2(r) = \frac{E_2}{l_a m} \quad (5.22)$$

Boundary conditions are:

$$\begin{aligned} S_f &= 0 \text{ at } r = 0 \\ S_f &= 1 \text{ at } r = 1 \end{aligned} \quad (5.23)$$

Equation 5.20 is a second order, ordinary differential equation with variable coefficients, (i. e., non-linear). No direct method of solution is known, but computer solution for any given numerical situation is not difficult. A corresponding finite difference equation, and Fortran II Program for its solution are given in Appendix A. An analytical solution for the case of constant metal thickness is given in Appendix B.

Note that once the tensile stress distribution in the fiber member has been found by solution of 5.20, the stress distribution in the metal member can be easily obtained from 5.15. In dimensionless form, if

$$\frac{\sigma_m}{F/t_m \text{ at } x=0} = S_m \quad (5.24)$$

From 5.15,

$$S_m = \left(\frac{F - \sigma_f t_f}{t_m} \right) \frac{1}{F/t_m \text{ at } x = 0} \quad (5.25)$$

or

$$S_m = (1 - S_f) \frac{t_m \text{ at } x = 0}{t_m}$$

The shear stress in the adhesive layer is given by 5.2. A dimensionless "shear stress concentration factor", T , is defined as:

$$T = \frac{\tau}{F/\ell} \quad (5.26)$$

From 5.2,

$$T = \frac{t_f \frac{d\sigma_f}{dx}}{\frac{F}{\ell}}$$

or

$$T = \frac{d \left(\frac{\sigma_f}{F/t_f} \right)}{d \left(x/\ell \right)} = \frac{dS_f}{dr} \quad (5.27)$$

Thus, the shear stress concentration factor is simply the dimensionless gradient of the tensile stress concentration factor for the fiber member.

The analysis here presented has been used to evaluate joints of constant metal thickness as well as linearly tapered metal thickness. Section VI presents this application of analysis.

VI APPLICATION OF THE ANALYSIS

Data obtained from computer solutions for different configurations and material constants may be compared through plots of stress concentration factors vs. r (position along the connection). The following plots are here included: (joint length = 1 in.)

Figure 5a: Constant Metal Thickness (Alum.)
.010" Adhesive Layer.

Figure 5b: Tapered Steel
.0015" Adhesive Layer.

Figure 5c: Tapered Steel
.0030" Adhesive Layer.

Figure 5d: Tapered Steel $t_a = t_f$.

Figure 5e: Tapered Titanium $t_a = t_f$.

VII JOINT "EFFICIENCY"

A ratio named "joint efficiency" may be used as a figure of merit for comparative evaluation of various joint designs. Define the joint efficiency as

$$\text{Eff.} = \frac{\text{Joint Failure Load}}{\text{Failure Load for the Fiber Element}} \quad (7.1)$$

The failure load for the fiber element is taken in the "undisturbed" fiber element external to the joint. The joint failure could initiate in the adhesive interface, in the fiber member, or in the metal member. The joint efficiency will depend on the lowest failure load, and is here taken as the load at initial fracture or yield rather than ultimate load.

For failure of the adhesive interface,

$$\text{Eff}_a = \frac{\tau_{\text{ult}} \ell / T}{\sigma_f t_f} \quad (7.2)$$

For failure of the fiber member,

$$Eff_f = \frac{1}{S_{f, \text{ maximum value}}} \quad (7.3)$$

The last ratio will not control in joints of constant fiber member thickness, since $S_f \leq 1$.

For failure (yielding) of the metal member,

$$Eff_m = \frac{\text{Min } (\sigma_{\text{yield}} + t_m) + \sigma_{ff} t_f \text{ at location of Min } (\sigma_{\text{yield}} t_m)}{\sigma_{ff} t_f} \quad (7.4)$$

The location at which metal yielding first occurs is at the position of maximum S_m .

Limiting values of stress must be known to predict the efficiency of any joint. It is emphasized that the maximum point values of shear stress as calculated from tests should be used, not "average" values as based on gross shear area. Values of the joint efficiencies for configurations shown in figures 3, 5a, 5b, and 5c are indicated on the figures, together with the limiting stress values used in determination of the efficiencies.

A "thin shim" joint has been proposed, in which a large number of very thin, high-strength stainless-steel shims are bonded between layers of fiber plastic, as shown in Figure 6. Ends of the metal members are displaced in such a manner that a severe change in total metal thickness is avoided. Bolts or pins complete the total transfer.

$$\frac{E_m}{E_f} = \frac{10 \times 10^6}{5 \times 10^6} = 2$$

$$\sigma_{f, \text{ ult}} = 250,000 \text{ PSI}$$

$$EFF_a = 0.26$$

$$\frac{G}{E_m} = \frac{1 \times 10^5}{10 \times 10^6} = .01$$

$$\tau, \text{ ult} = 10,000 \text{ PSI}$$

$$EFF_m = 0.36$$

$$\sigma_{m, \text{ yield}} = 60,000 \text{ PSI}$$

Figure 5 a Stress Profiles for Lap Joint

$$\frac{E_m}{E_f} = \frac{30 \times 10^6}{5 \times 10^6} = 6.0 \quad \sigma_{f, \text{ ult}} = 250,000 \text{ PSI} \quad \text{EFF}_a = .123$$

$$\frac{G}{E_m} = \frac{1 \times 10^5}{30 \times 10^6} = .00333 \quad \tau, \text{ ult} = 10,000 \text{ PSI} \quad \text{EFF}_m = .858$$

$$\sigma_{m, \text{ yield}} = 180,000 \text{ PSI}$$

Figure 5 b Stress Profiles (Inner Cup of Test Joint)

$$\frac{E_m}{E_f} = \frac{30 \times 10^6}{5 \times 10^6} = 6.0$$

$$\sigma_{f, \text{ult}} = 250,000 \text{ PSI}$$

$$EFF_a = .163$$

$$\tau_{\text{ult}} = 10,000 \text{ PSI}$$

$$EFF_m = .926$$

$$\frac{G}{E_m} = \frac{1 \times 10^5}{30 \times 10^6} = .00333$$

$$\sigma_{m, \text{yield}} = 180,000 \text{ PSI}$$

Figure 5c Stress Profiles (Inner Cup of Test Joint)

$$\frac{E_m}{E_f} = \frac{30 \times 10^6}{5 \times 10^6} = 6$$

$$\sigma_f, \text{ ult} = 250,000 \text{ PSI}$$

$$EFF_a = .327$$

$$\tau, \text{ ult} = 10,000 \text{ PSI}$$

$$EFF_m = 1.0$$

$$\frac{G}{E_m} = \frac{1 \times 10^5}{30 \times 10^6} = .00333 \quad \sigma_m, \text{ yield} = 180,000 \text{ PSI}$$

Figure 5d Stress Profiles (Inner Cup of Test Joint)

$$\frac{E_m}{E_f} = \frac{17 \times 10^6}{5 \times 10^6} = 3.4$$

$$\sigma_f, \text{ ult} = 250,000 \text{ PSI}$$

$$EFF_a = .345$$

$$\frac{G}{E_m} = \frac{1 \times 10^5}{17 \times 10^6} = .005882$$

$$\tau, \text{ ult} = 10,000 \text{ PSI}$$

$$EFF_m = 1.0$$

$$\sigma_m, \text{ yield} = 175,000 \text{ PSI}$$

Figure 5 e Stress Profiles (Inner Cup of Test Joint)
(Identical to configuration of Figure 5 d
except titanium metal)

This joint may be evaluated by assuming that the tensile force in each fiber-plastic layer is transferred by two shear interface surfaces. This situation is represented in Figure 7. Further, it is assumed that the entire fiber-plastic layer deforms in shear in a manner as assumed for an adhesive layer in the analysis derived from Figure 4a. Thus, $t_a = t_f$ is assumed. Stress profiles for particular dimensions and material properties are shown in Figure 8. The shim thickness is here taken as 0.008", the fiber-plastic thickness as 0.006": The ratio corresponds to the ratio of ultimate strengths of 250,000 ÷ 187,000, and half-thickness dimensions were used to account for the double shear area.

If the analytical solution of Appendix B is applied to this joint, it may be seen that the shear stress concentration factor, T , increases linearly with an increase in joint length. This joint is not increased in strength by increasing the joint length, and a joint efficiency of 30% is indicated.

Figure 9 presents a design concept in which an attempt is made to "tension" the fiber elements by passing the fibers around a corner. Here the axial tension in the fiber member is transferred to a metal ring by compressive stress across the interface (in addition to a shear bond). The concept appears attractive, but is limited by compressive strength of the resin as is shown in the following.

Figure 10 pictures a differential arc of the fiber member under force equilibrium as it exists in contact with the radius of the metal ring.

The radial pressure force must balance the radial components of tension forces in the fiber element; the shear stress at the interface has no radial component. Thus,

$$pRd\theta = \sigma_f t_f \sin \frac{d\theta}{2} + (\sigma_f + d\sigma_f) t_f \sin \frac{d\theta}{2} \quad (7.5)$$

on simplification, the simple expression

$$p = \frac{\sigma_f t_f}{F} \quad (7.6)$$

is obtained.

Figure 6

"Thin Shim" Multiple Interface Joint

Figure 7

Multiple Interface Joint Considered as Two
Interface Surfaces for Mathematical Analysis

$$\frac{E_m}{E_f} = \frac{30 \times 10^6}{5 \times 10^6} = 6$$

$$\sigma_{f, \text{ult}} = 250,000 \text{ PSI}$$

$$\text{EFF}_a = .30$$

$$\frac{G}{E_m} = \frac{1 \times 10^5}{30 \times 10^6} = .00333$$

$$\tau_{\text{ult}} = 10,000 \text{ PSI}$$

$$\text{EFF}_m = 1.0$$

$$\sigma_{m, \text{yield}} = 187,000 \text{ PSI}$$

Figure 8 Stress Profiles - Thin Shim Joint

Figure 9

"Corner Tension" Joint

Figure 10

Differential Arc of Fiber Member
In Contact with Radius of Metal Ring

The last relation may be used to determine the radius required to prevent compression failure of the resin at the fiber-metal interface. At $\theta = 0$, the maximum value of $\sigma_f t_f$ exists (designated "F" in previous analyses), and correspondingly the highest compressive pressure, p , exists if R is constant.

For $\sigma_f = 250,000 \text{ PSI}$
 $t_f = 0.06 \text{ in}$
 $p = 10,000 \text{ PSI}$

The required radius R is

$$R = \frac{\sigma_f t_f}{p} = \frac{250,000 \times 0.06}{10,000} = 1.50''$$

It may be concluded the radial thickness of a joint designed around this concept may be excessive. If R is reduced as θ increases, an improvement in size may be achieved, with load transfer initially by shear only but with compression transfer increasing as θ increases.

APPENDIX A

Numerical Solution

A solution of equation 5.20 is desired. The values of $\phi_1(r)$ and $\phi_2(r)$ are known for the closed region $0 \leq r \leq 1$.

$$\frac{d^2 S_f}{dr^2} - \phi_1(r) S_f = -\phi_2(r) \quad (5.20)$$

Making use of finite difference approximations, for equal increments r :

$$\frac{dS_f}{dr} \cong \frac{(\Delta S_f)_{n+1} + (\Delta S_f)_n}{2(\Delta r)} \quad (A.1)$$

Where $n = 1$ to n incremental steps. (The computer program attached used 100 increment steps to obtain reasonable accuracy.)

$$\frac{d^2 S_f}{dr^2} \cong \frac{(\Delta S_f)_{n+1} - (\Delta S_f)_n}{(\Delta r)^2} \quad (A.2)$$

5.2 is approximated by:

$$\frac{(\Delta S_f)_{n+1} - (\Delta S_f)_n}{(\Delta r)^2} - \phi_1(r) S_f = -\phi_2(r) \quad (A.3)$$

Digital solution of equation A.3 is accomplished by assuming an initial slope, i.e., a value of $(\Delta S_f)_1$ is assumed. ΔS_{f2} is calculated from Equation A.3, and the process is repeated to find the value of S_f at the end of each increment Δr . If $S_f \neq 1$ at $r = 1$, the entire calculation is repeated for a corrected initial value of $(\Delta S_f)_1$. The computer program provides for an iterative, converging, means of finding S_f as a function of r .

KEY TO FORTRAN REPRESENTATION

$$SF = S_f = \frac{\sigma_f}{F/t_f}$$

$$R = r$$

$$DELR = \Delta r$$

$$AL = L_a = t_a/\ell$$

$$FL = L_f = t_f/\ell$$

$$AM = m = t_m/\ell$$

$$PH1 = \phi_1(r) = \frac{E_1}{L_a L_f} + \frac{E_2}{L_a m}$$

$$PH2 = \phi_2(r) = \frac{E_2}{L_a m}$$

$$E = E_1 = G/E_f$$

$$F = E_2 = G/E_m$$

Example Of Computer Solution

The following is recorded as an example of the application of the computer program in calculating the stress distribution in a particular joint. For this example, the joint shown on figure 5d is used.

• Numerical data required by the program are:

SF(1), the initial value of the tensile stress function: this value must be "guessed". Unless a reasonably close value is assumed, the program may not converge to provide correct answers. A few trials will provide the background needed to provide a satisfactory number.

$$AL = t_a/\ell = .0214/1 = .0214$$

$$FL = t_f/\ell = .0214/1 = .0214$$

$$E = G/E_f = 1 \times 10^5 / 5 \times 10^6 = .020 \text{ (The value of the shear modulus } G \text{ is not precisely known, and further refinement in its value is needed. The value, of course, depends on the material used at the shear interface.)}$$

$$F = G/E_m = 1 \times 10^5 / 30 \times 10^6 = .003333$$

$AM = t_m / \ell$. For this joint, of linear taper, the metal thickness is $t_m = .05886 - (.05886 - .01428) x$; for $\ell = 1$, $R(1) = x$, hence $AM = .05886 - .04458 * R(1)$.

The computer program prints the values of r , S_f , ΔS_f , $\frac{dS_f}{dr}$, and $\frac{d^2S}{dr^2}$.

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```
H NAME DIP
H EQUIP=CARDRE,PRINTER
  DIMENSION DELS(120),SF(120),DIF(120),DIFF(120),R(120)
  L=0
  AL=.0214
  FL=.0214
85 K=1
  G=0
  SF(1)=.4
  DELS(1)=SF(1)
  E=.020
  F=.003333
  DELR=.01
  1 R(1)=.01
  PRINT 20,SF(1)
20 FORMAT (1F15.8)
  DO 2 I=1,100
  AM=.05886-.04458*R(I)
  PH1=1./((AL*FL)*(E+(FL*F)/AM))
  PH2=F/(AL*AM)
  DELS(I+1)=DELR*DEL R*(PH1*SF(I)-PH2)+DELS(I)
  DIF(I)=(DELS(I+1)+DELS(I))/(2.*DELR)
  DIFF(I)=(DELS(I+1)-DELS(I))/(DELR*DELR)
  SF(I+1)=SF(I)+DELS(I+1)
  R(I+1)=R(I)+DELR
  PRINT 5,R(I),DELS(I+1),SF(I+1),DIF(I),DIFF(I)
  5 FORMAT (1X,5F15.8)
  2 CONTINUE
  IF(K-12)62,7,7
62 IF(SF(100)-1.)8,7,6
  6 GO TO (39,40,41,42,43,44,45,46,47,48,49),K
39 G=.01
  GO TO 50
40 G=.001
  GO TO 50
41 G=.0001
  GO TO 50
42 G=.00001
```

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```
GO TO 50
43 G=.000001
  GO TO 50
44 G=.0000001
  GO TO 50
45 G=.00000001
  GO TO 50
46 G=.000000001
  GO TO 50
47 G=.0000000001
  GO TO 50
48 G=.00000000001
  GO TO 50
49 G=.000000000001
50 SF(1)=SF(1)-G
  DELS(1)=SF(1)
  PRINT 21
21 FORMAT (40X,11HEND OF PASS)
  GO TO 1
  8 K=K+1
  IF(K-12)60,61,61
61 SF(1)=SF(1)+G
  GO TO 1
60 SF(1)=SF(1)+G
  GO TO 6
  7 L=L+1
  PRINT 90
90 FORMAT (/,14HEND OF SECTION)
82 END
```


	r_1	$\Delta S_f(itl)$	$S_f(itl)$	$\frac{dS_f}{dr}$	$\frac{d^2 S_f}{dr^2}$
$\Delta S_{f1} = 0.00441227$					
0.01000000		0.00416609	0.00857837	0.42891836	-2.46180691
0.02000000		0.00393718	0.01251555	0.40516389	-2.28908638
0.03000000		0.00372447	0.01624002	0.38308283	-2.12712545
0.04000000		0.00352695	0.01976698	0.36257125	-1.97549047
0.05000000		0.00334369	0.02311067	0.34353233	-1.83259353
0.06000000		0.00317382	0.02628450	0.32587592	-1.69868930
0.07000000		0.00301654	0.02930103	0.30951811	-1.57287211
0.08000000		0.00287108	0.03217211	0.29438088	-1.45457323
0.09000000		0.00273675	0.03490887	0.28039172	-1.34325827
0.10000000		0.00261291	0.03752178	0.26748331	-1.23842474
0.11000000		0.00249895	0.04002073	0.25559319	-1.13959981
0.12000000		0.00239432	0.04241505	0.24466350	-1.04633812
0.13000000		0.00229850	0.04471354	0.23464071	-0.95821978
0.14000000		0.00221101	0.04692456	0.22547537	-0.87484848
0.15000000		0.00213143	0.04905598	0.21712188	-0.79584964
0.16000000		0.00205934	0.05111532	0.20953828	-0.72088872
0.17000000		0.00199438	0.05310970	0.20268609	-0.64956966
0.18000000		0.00193622	0.05504592	0.19653008	-0.58163327
0.19000000		0.00188454	0.05693047	0.19103813	-0.51675581
0.20000000		0.00183908	0.05876955	0.18618112	-0.45464762
0.21000000		0.00179958	0.06056912	0.18193272	-0.39503176
0.22000000		0.00176581	0.06233493	0.17826935	-0.33764276
0.23000000		0.00173759	0.06407252	0.17517000	-0.28222541
0.24000000		0.00171474	0.06578726	0.17261621	-0.22853359
0.25000000		0.00169710	0.06748436	0.17059190	-0.17632908
0.26000000		0.00168456	0.06916892	0.16908335	-0.12538056
0.27000000		0.00167702	0.07084594	0.16807913	-0.07546245
0.28000000		0.00167438	0.07252032	0.16757005	-0.02635391
0.29000000		0.00167660	0.07419692	0.16754909	0.02216220
0.30000000		0.00168363	0.07588055	0.16801141	0.07030034
0.31000000		0.00169546	0.07757601	0.16895427	0.11827329
0.32000000		0.00171209	0.07928809	0.17037711	0.16629315
0.33000000		0.00173354	0.08102164	0.17228143	0.21457234
0.34000000		0.00175988	0.08278151	0.17467092	0.26332462
0.35000000		0.00179115	0.08457267	0.17755137	0.31276612
0.36000000		0.00182746	0.08640013	0.18093078	0.36311633
0.37000000		0.00186892	0.08826905	0.18481936	0.41459927
0.38000000		0.00191567	0.09018472	0.18922958	0.46744439
0.39000000		0.00196786	0.09215258	0.19417624	0.52188788
0.40000000		0.00202567	0.09417825	0.19967655	0.57817374
0.41000000		0.00208933	0.09626758	0.20575019	0.63655498
0.42000000		0.00215906	0.09842664	0.21241944	0.69729489
0.43000000		0.00223513	0.10066177	0.21970926	0.76066839
0.44000000		0.00231782	0.10297959	0.22764742	0.82696336
0.45000000		0.00240747	0.10538706	0.23626465	0.89648213
0.46000000		0.00250442	0.10789148	0.24559477	0.96954301
0.47000000		0.00260907	0.11050056	0.25567490	1.04648193
0.48000000		0.00272184	0.11322240	0.26654558	1.12765417
0.49000000		0.00284318	0.11606558	0.27825103	1.21343622
0.50000000		0.00297360	0.11903918	0.29083935	1.30422771
0.51000000		0.00311365	0.12215283	0.30436275	1.40045356
0.52000000		0.00326391	0.12541674	0.31887785	1.50256621
0.53000000		0.00342501	0.12884175	0.33444592	1.61104805
0.54000000		0.00359765	0.13243940	0.35113323	1.72641399
0.55000000		0.00378257	0.13622198	0.36901138	1.84921428
0.56000000		0.00398058	0.14020256	0.38815763	1.98003749
0.57000000		0.00419253	0.14439509	0.40865539	2.11951373
0.58000000		0.00441936	0.14881445	0.43059455	2.26831819

0.50000000	0.00466208	0.15347653	0.45407202	2.42717486
0.60000000	0.00492176	0.15839829	0.47919219	2.59686062
0.70000000	0.00519959	0.16359788	0.50606754	2.77820968
0.80000000	0.00549680	0.16909468	0.53481918	2.97211828
0.90000000	0.00581475	0.17490943	0.56557753	3.17954993
0.00000000	0.00615491	0.18106434	0.59848298	3.40154102
0.10000000	0.00651883	0.18758316	0.63368672	3.63920686
0.20000000	0.00690820	0.19449137	0.67135150	3.89374837
0.30000000	0.00732485	0.20181621	0.71165253	4.16645926
0.40000000	0.00777072	0.20958694	0.75477850	4.45873386
0.50000000	0.00824793	0.21783486	0.80093255	4.77207570
0.60000000	0.00875874	0.22659360	0.85033346	5.10810682
0.70000000	0.00930560	0.23589920	0.90321688	5.46857795
0.80000000	0.00989114	0.24579034	0.95983667	5.85537968
0.90000000	0.01051819	0.25630853	1.02046634	6.27055456
0.00000000	0.01118982	0.26749835	1.08540067	6.71631053
0.10000000	0.01190933	0.27940768	1.15495740	7.19503548
0.20000000	0.01268026	0.29208793	1.22947914	7.70931333
0.30000000	0.01350645	0.30559439	1.30933542	8.26104172
0.40000000	0.01439205	0.31998643	1.39492488	8.85505137
0.50000000	0.01534151	0.33532794	1.48667778	9.49462757
0.60000000	0.01635966	0.35168760	1.58505858	10.18153380
0.70000000	0.01745172	0.36913932	1.69056894	10.92053785
0.80000000	0.01862330	0.38776262	1.80375084	11.71584082
0.90000000	0.01988050	0.40764312	1.92519009	12.57200923
0.00000000	0.02122990	0.42887302	2.05552019	13.49401073
0.10000000	0.02267863	0.45155165	2.19542651	14.48725384
0.20000000	0.02423439	0.47578604	2.34545094	15.55763240
0.30000000	0.02590555	0.50169159	2.50609698	16.71157515
0.40000000	0.02770116	0.52939275	2.68033536	17.95610145
0.50000000	0.02963105	0.55902380	2.86661029	19.29888384
0.60000000	0.03170588	0.59072968	3.06684630	20.74831853
0.70000000	0.03393724	0.62466692	3.28215592	22.31360516
0.80000000	0.03633772	0.66100464	3.51374813	24.00483716
0.90000000	0.03892103	0.69992567	3.76203784	25.83310455
0.00000000	0.04170209	0.74162777	4.03115642	27.81061126
0.10000000	0.04469718	0.78632494	4.31906352	29.95080952
0.20000000	0.04792403	0.83424897	4.63106034	32.26855448
0.30000000	0.05140206	0.88565103	4.96630453	34.78028267
0.40000000	0.05515248	0.94080351	5.32772704	37.50421912
0.50000000	0.05919854	1.00000206	5.71755122	40.46061866
1.00000000	0.06356575	1.06356781	6.13821456	43.67204871

END OF PASS

APPENDIX B

Analytical Solution For Constant Metal Thickness

If the metal thickness is constant, Equation 5.20 is linear, and of the form

$$\frac{d^2 S_f}{dr^2} - A S_f = -B \quad (B.1)$$

With solution

$$S_f = C_1 \sinh \sqrt{A} r + C_2 \cosh \sqrt{A} r + \frac{B}{A} \quad (B.2)$$

Incorporation of boundary conditions stated in 5.23 leads to

$$S_f = \frac{B}{A} + \left(1 - \frac{B}{A}\right) \frac{\sinh \sqrt{A} r}{\sinh \sqrt{A}} - \frac{B}{A} \frac{\sinh \sqrt{A} (1-r)}{\sinh \sqrt{A}} \quad (B.3)$$

From the last equation the shear stress concentration factor is found, by 5.27, and is

$$T = \left(1 - \frac{B}{A}\right) \sqrt{A} \frac{\cosh \sqrt{A} r}{\sinh \sqrt{A}} + \frac{B}{A} \sqrt{A} \frac{\cosh \sqrt{A} (1-r)}{\sinh \sqrt{A}} \quad (B.4)$$

Equation B.3 serves as a check on the accuracy of the computer solution per Equation A.3.

Since both $\sinh \sqrt{A}$ and $\cosh \sqrt{A}$ are both very large numbers, it may be observed that the maximum value of T occurs at $r = 1$, and is

$$T_{\max} = \left(1 - \frac{B}{A}\right) \sqrt{A} \quad (B.5)$$

In terms of the physical constants, this is

$$T_{\max} = \ell \sqrt{\frac{E_m t_m G}{E_f t_f t_a (E_m t_m + E_f t_f)}} \quad (B.6)$$

Note that if ℓ is increased, $T_{\max}$ increases in direct proportion. The strength is not increased by increasing the joint length. Figure B.1 presents a plot of joint strength as a function of length, and correlates test data from one inch strip test made by ACF for SCLL.

$$F = \frac{2\tau_{ult}}{\sqrt{\frac{2G_a}{Ett_a}}} \left[\frac{\sinh \sqrt{\frac{2G_a}{Ett_a}}}{1 + \cosh \sqrt{\frac{2G_a}{Ett_a}}} \right]$$

Figure B.1 Correlation of Joint Test Data (ACF)

Ref: ACF Test Specimens .001 through .015

$$\tau_{ult} = 5,090 \text{ psi}$$

$$\sqrt{\frac{2G_a}{Ett_a}} = 4.045$$

APPENDIX C

Experimental Tests

Figure C. 1a shows a multiple-shear surface scarf joint which was fabricated for test in the summer of 1965. This design employs three cup shaped metal elements which provide four shear surfaces; filament winding is carried out in two separate operations. A salt mandrel is used, and two hemispherical domes are used as end surfaces during wrap. The completed assembly and tensile loading fixture is shown in Figure C. 1b. One only joint of this design was tested, and only the inner layer of filament winding (two inner cups only) was employed. A good bond to 17-4PH stainless cups was not obtained. Failure load was 5200 lbs, as shown on Figures C. 2, C. 3, and C. 4. Photograph No. P41842 records the test setup, and Photograph No. P-41843 shows strain gages mounted on the specimen.

Figure C.1a Multiple-shear Surface Scarf Joint

Figure C.1b Inner Layer of Filament Winding Employed
(Two Inner Cups Only)

REF: Sketch PS-40586-98

P-41842

Test Setup Used for Testing Scarf
Joints Shown in Figure C.1

P-41843

Strain Gages Attached to Specimen
in Preparation for Test

Figure C. 2 Strain in Filament Wound Cylinder

Figure C.3 Strain in Filament Wound Cylinder

Figure C. 4
Strain in Filament Wound Cylinder

Figure C. 5a shows the mandrel configuration as used to fabricate a "Compression Corner" type joint. Sequential to the winding of two complete circuits with a single roving (20 end, 204 filaments/strand), the fibers were cut near the ends of the spherical domes, and circumferential wraps then wound so as to pull the longitudinal (7° wrap angle) fibers into the bottom of the grooves. The grooves were filled with circumferential wrap until the surface was tangent to the O. D. Following cure, the frayed ends were machined off, the salt dissolved with water, and loading plugs installed as shown in Figure C. 5b.

Figure C. 6 presents the test load-elongation curve for this joint. First failure at 17,000 pounds load was believed to be a result of shear failure at the shoulder; ultimate load was 30,000 pounds.

Photograph P-41946 shows that the failure was a shear failure in the resin; the longitudinal fibers were not fractured.

Figure C. 5a Mandrel Configuration Used To Fabricate a "Compression Corner" Type Joint

Figure C. 5b Fully Assembled Compression Corner Joint

P-41946

Shear Failure in Resin - Longitudinal Fibers Undamaged

Figure C. 6

Filament Wound Cylinder Joint Test

Figure C. 7a shows the mandrel as assembled to wind a "double groove" shoulder joint. This joint was fabricated following the same procedure used for the "single" groove joint. Figure C. 7b shows the loading plugs in place.

First failure (probably shear failure at the inner shoulder) appeared at 20,000 pounds (see Figure C. 8), which was not significantly higher than first failure of the single groove joint. Photograph P-40245 shows that ultimate fracture (at 60,000 pounds) pulled the fibers in tension (at the bottom of the outer groove). Although the ultimate load was doubled, useful load (at first permanent damage) was not enhanced by addition of a second groove.

Figure C.7a Mandrel Used To Wind a
"Double Groove" Type
Shoulder Joint

Figure C.7b Fully Assembled Double Groove Shoulder Joint

Figure C. 8

Filament Wound Cylinder Joint Test

P-42045

Fracture Under Test at 60,000 PSI

Figure C.9 a shows the mandrel as used to fabricate a laminated tapered shim joint. Four 6061T6 aluminum shims 0.020" thick were positioned between five longitudinal wrap layers, forming a composite approximately 0.140" thick. The shims were tapered from 0.020" thickness to a sharp edge (.002" max), and rolled to a cylindrical form. The taper portion is exterior to inner and outer aluminum cups, and a triple row of 3/16" 24ST "ice box" rivets carry the load in double shear. The bundle was "B-staged" prior to riveting, with final cure after driving the rivets. The aluminum shims were etched and primed with FM-47 prior to filament winding.

The tapered thickness of the shims was primarily provided to reduce severe stress concentration at the adhesive interface. Circumferential wraps were placed over the tapered length, to build total shell thickness to a uniform diameter and to hold a shim in place for the subsequent longitudinal wrap. Figure C.9 b shows the assembly with loading plugs in place.

Figure C.10 records the test load-elongation plot for the tapered shim joint. Fracture occurred at 90,000 pounds with no significant inelastic action prior to failure.

Photograph No. 42301 shows that failure occurred in shear of the rivets on the inner surface of the laminated shell and in tension of the outer cup. No damage to the laminated shell itself was evident.

Of the four joints tested, the laminated shim design is the most promising. No fundamental consideration exists which would limit the design strength of a joint developed from this concept.

Figure C.9 a

Mandrel Used to Fabricate
a "Laminated Tapered Shim"
Type Joint

Figure C.9 b Fully Assembled Laminated Tapered Shim Joint

Figure C.10 Filament Wound Cylinder Joint Test

P-42301

Failure of Rivets on Inner Surface of Laminated Shell
and In Tension of Outer Cup - No Damage to Shell