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STRUCTURAL ANALYSIS OF ADHESIVELY BONDED JOINTS OF TUBULAR GEOMETRY

by

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ABSTRACT

Bonded tubular joints are being considered in structural assemblies as replacements for welded connections since welding generally reduces the overall strength of materials and induces severe stress concentrations. The performance of bonded joints depends on the adhesive properties and the stress-transfer mechanism between the adherend and the adhesive, in addition to bond parameters such as shear area and bondline thickness. We conducted both linear and non-linear finite element analysis, of the joints, made prototype joints and measured their stress distributions under experimental loadings, and analytically revised joint geometry to minimize normal and shear stresses. Prototype joints tested under compressive loads exhibited tubular buckling at loads considerably below the expected upper limits of joint bond strength, indicating substantial stress concentrations in the tubular joints. The stresses can be reduced by appropriate geometric modifications of the joints.

1.0 INTRODUCTION

Adhesively bonded lap joints, and such geometric variations as the tubular joint or the tube and socket, are increasingly being used in primary structural assemblies.(1) The

KEY WORDS: Structural Adhesives, Stress Analysis, Aluminum Joining, Joint Geometry, Shear Strength, Tubular Joints, Overlap Length, Bondline Thickness, Stress-Strain Curve, Finite Element Analysis, Stress Concentration

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performance of such joints depends on 1) adhesive shear behavior, 2) bond parameters such as bondline thickness and shear area, and 3) stress distribution and stress transfer mechanisms between the adhesive and the adherend. If its performance can be optimized, the tubular joint is an ideal candidate to replace the welded connection to aluminum end fittings of clevis hinged joints of foldable structures.

As part of a U.S. Army-sponsored program, we have studied the stress distribution in an adhesively bonded tubular joint and attempted to optimize its configuration by minimizing the normal and shear stresses on the joints. We also made prototype joints bonded with a toughened adhesive,*(2) and measured their stress distributions under experimental loadings. The adherend surfaces were phosphoric acid anodized(3) to induce failure either in the adhesive or in the aluminum tube away from the joint, rather than at the interface. Adhesive and adherend stress distributions in the joint overlap, were determined by linear, three-dimensional and non-linear axisymmetric finite element analysis (FEA), performed with the ANSYS Code. The joints were tested to validate the FEA predictions.

2.0 EXPERIMENTAL PROCEDURE

2.1 Linear 3-D Analysis of Bonded Joint   A detailed, three-dimensional, linear-elastic, static, finite element analysis of the bonded hinge joint was performed with the ANSYS Code, using standard FEM brick elements (Fig. 1a,b). Since the joint was structurally symmetric, only one-half of it was modeled; the bondline was assumed to be 0.19mm (0.0075 in.) thick. The adhesive was modeled as a linear-elastic, isotropic continuum with 2413 MPa (350 ksi) modulus and a Poisson's ratio of 0.35. The allowable axial stress in the adhesive was assumed to be 41.37 MPa (6 ksi) under rapid loading.

2.1.1 Parametric Optimization Studies   A linear-elastic, axisymmetric model was used for the parametric studies for computational efficiency. After the prototype model was completed (Sec. 2.3), the bondline thickness was increased to 5.1mm (0.02 in.) to more accurately reflect the actual joint configuration. Only axial loads were imposed on the joint, and the geometry was successively modified so as to minimize the stress concentrations in the adhesive. The geometric configurations considered were (Fig. 2):

Case 1: The unmodified joint geometry was used as a baseline for comparative purposes.

Case 2: An internal rigid plug within the outer tube was simulated by specification of appropriate boundary conditions (i.e., constraining the nodes on the inner wall

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of the tube in the radial direction). The aim was to investigate the effects of constraining the radial contraction due to Poisson's ratio effects.

Case 3: The ends of the clevis and the outer tube being joined were tapered uniformly over the length of the joint.

Case 4: Chamfers were introduced at both ends of the adhesive joint to increase the amount of adhesive at the very ends, where large stress concentrations were observed. The aim was to distribute the loads at the two ends over a thicker bondline ,thereby reducing the stress magnitudes.

2.2 Nonlinear Axisymmetric Analysis A nonlinear analysis was performed to examine the stress redistribution resulting from adhesive yielding. The stress-strain curve used for this analysis was a bilinear constitutive (idealized) model (Fig. 3) and was assumed to be valid for describing relatively rapid loading of the joint.

Figure 4 shows the FEM discretization used for the nonlinear analysis. A capacity load of 0.063 MN (14 kilopounds) per joint was applied incrementally. The first increment was designed to produce a maximum stress in the adhesive that was just short of yield. Subsequent load increments were small enough to ensure that plastic strain increments were less than 5% of the initial elastic yield strain. This limitation on the load increment magnitude is necessary to ensure that the iterative solution procedure converges to the required accuracy within each increment cycle.

2.3 Prototype Joint Testing Joint models typical of the clevis hinged joints were fabricated and bonded. The model consisted of tubular aluminum 6061-T6 male/female connections (Fig. 5). The male component was a solid cylindrical piece with one end machined so as to form a lap joint with the female hollow tube. A part of the machined male piece (about 3.8cm long) covered by a PVC sleeve provided an unbonded region between the connection to allow for compression displacement. Strain gauges were affixed to the specimen at different locations (Fig. 6) to monitor the stresses in the joint while it was tested in compression.

3.0 RESULTS AND DISCUSSION

3.1 Linear 3-D Analysis of Bonded Joints This analysis of the unmodified joint design revealed a large stress concentration at the outer surface of the leading edge of the adhesive and also demonstrated that under the applied maximum allowable load of 0.063 MN the maximum stress would exceed the strength of the adhesive. We therefore modified the joint geometry in an attempt to minimize the stress concentration.

3.1.1 Parametric Optimization Studies Increasing the bondline thickness to 5.1mm (Case 1) had no effect on the stress distribution of the unmodified joints. Since the highest stresses were at the fillet between the adhesive layer and the outer tube, we confined our geometric modification of the joint to this area. Figure 7 shows Von Mises' stress for the outer and inner surface, as well as the middle layer of the bondline. We obtained the values at the outer and inner surfaces by averaging stress values in the adjacent elements of the outer tube and inner clevis stem, respectively. All stresses shown are a result of imposition of a 0.063 MN maximum allowable load.

Since the lap shear strength of the proposed adhesive was 45.3 MPa (6570), psi the data clearly indicated the need for modifications. Based on our previous work on joint fracture testing, [4] we expected that increasing the stiffness of the joints would lower the stress concentration. We therefore modeled stresses for the configuration described in Cases 2-4 (Fig. 2).

Figure 8 shows the Von Mises' stress in the bondline for Case 2, incorporating a rigid plug adjacent to the end of the inner tube. This modification yielded only a marginal reduction (20%) in the magnitude of the stress concentration at the lead edge of the joints. Similarly, the tapered joint geometry of Case 3 did not yield any significant advantage over the unmodified joints, a result contrary to conventional experience from planar adhesive lap joints; the difference may be attributable to the larger out-of-plane (radial) stiffness in the axisymmetric configuration. Introducing chamfers at the edges of both the outer tube and the inner clevis stem as in Case 4, reduced the stress concentration by about 20%, similar to the effect of the rigid plug.

Since these geometric modifications did not reduce the stress concentration sufficiently to prevent yielding in the adhesive, we performed a nonlinear analysis to determine the size of the plastic zone and the redistributed stresses in the remaining elastic region.

3.2 Nonlinear Axisymmetric Analysis We performed the nonlinear analysis on the joint model incorporating a rigid plug adjacent to the end of the inner tube (Case 2) because it produced the best results in the linear analysis. Figure 9 shows the stress distribution in the adhesive after the final increment, which corresponds to a load of 0.063 MN. The stresses plotted are those at the adhesive midplane.

The result of the nonlinear FEM analysis at a maximum stress of 16.55 MPa (2400 psi), is about 3% of the stress found with the limit load analysis. The width of the yield zone is not very large and the greatest proportion of the load is carried by the leading edge of the joint. We can thus infer an enormous strength reserve in the joint configuration. For example, the limit load on this joint is 0.30 MN (68 kilopounds), if we assume full yielding across the bondline.

3.3 Prototype Joint Testing The strain gauge results of a model joint test are shown in Fig. 10. Note that these results match those of the elastic region of the nonlinear analysis (compare data point"A" in Figs. 9&10). Note also the severe stress concentrations at the leading edge of the bonded joints (i.e., the fillet between the male and female connections) at gauge locations 1L and 2L. In contrast, the stresses at gauge locations 3L and 3A are minimal. This region around the solid block of the male component serves essentially as a plug to lower the stress concentration. Incipient plastic deformation of the aluminum tubes occurred at loads of 0.16 MN (35 kilopounds) and above. The tube failed in compression buckling at about 0.23 MN (52 kilopounds), which is considerably lower than the limit load of 0.30 MN. This performance can be attained only with stress redistribution.

4.0 SUMMARY AND CONCLUSIONS

The stress concentrations in a bonded tubular joint were analyzed via linear and nonlinear finite element modelling and prototype joint testing. Prototype joints tested under compressive loads displayed severe stress concentrations at the leading edge of the joint and failed by tubular buckling at loads considerably lower than the expected limit load of the bonded joint.

A detailed three-dimensional linear elastic stress analysis of the joint revealed large stress concentrations at the outer leading edge of the adhesive. A linear elastic axisymmetric parametric study, conducted to minimize the stress concentration under axial tensile loading, showed that appropriate geometric modification of the joint (i.e., an internal rigid plug within the outer tube and chamfering of tube ends) lowered the stress concentration by about 20%. The tapered joint geometry did not yield any significant reductions in stress concentrations, contrary to the experience with conventional lap joints, this result may be attributable to a larger radial stiffness in the axisymmetric configuration.

A nonlinear axisymmetric analysis indicated an enormous strength reserve in the joint configuration because the vast majority of the load was carried by the leading edge of the joint.

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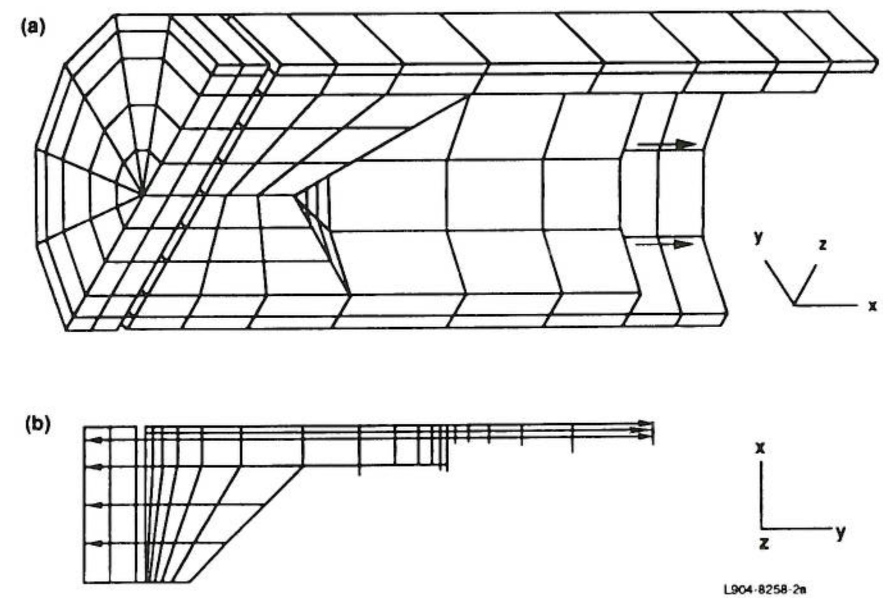

Figure 1. Discretized finite element models used for linear analysis

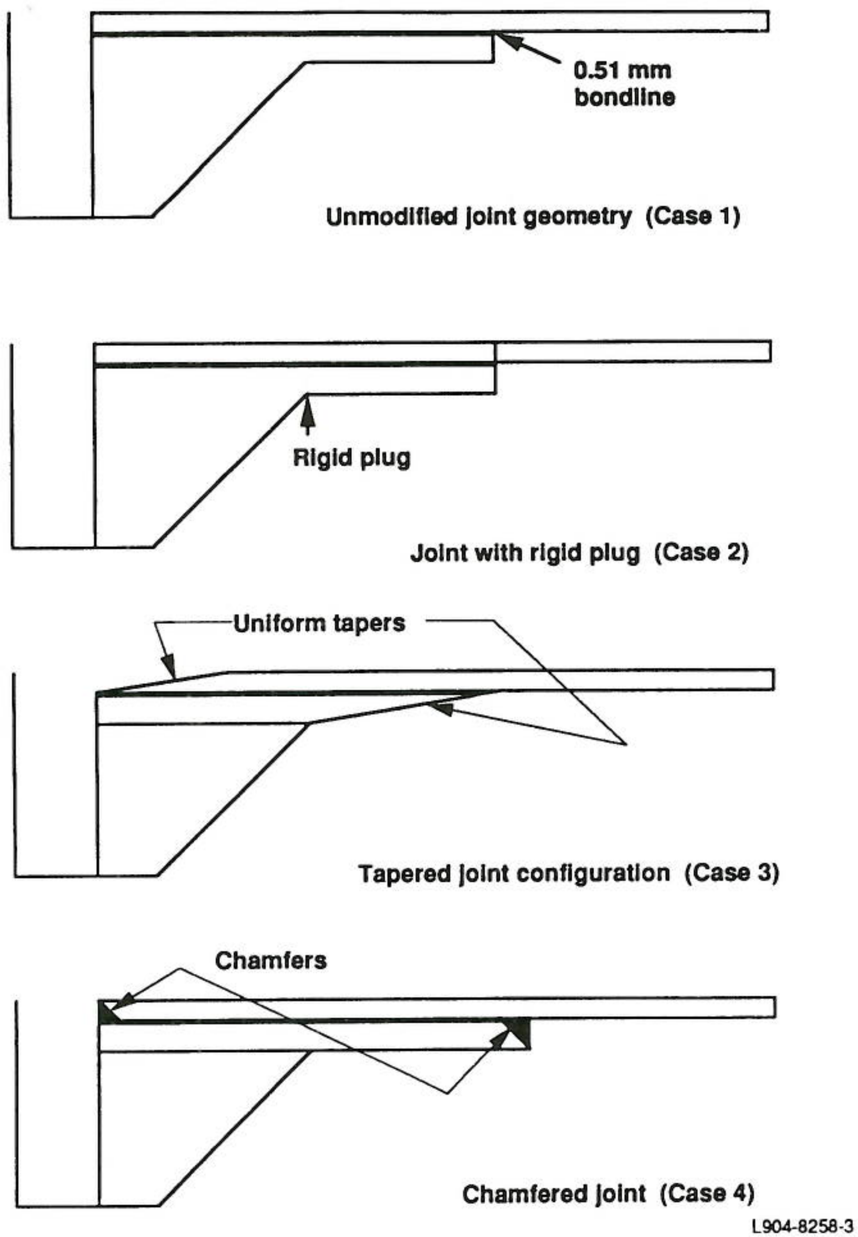

Figure 2. Sketch of geometric configurations of model joint.

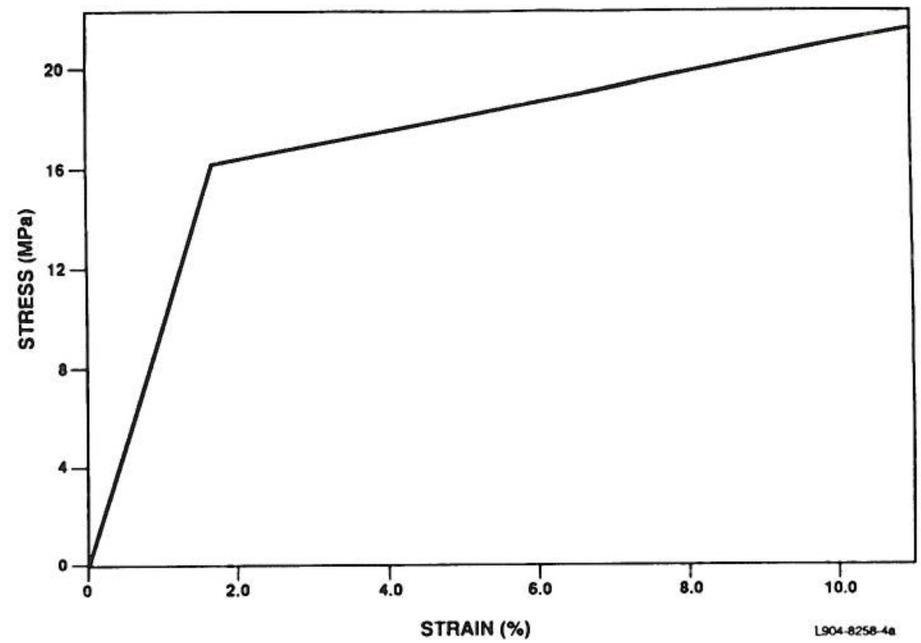

Figure 3. Idealized stress-strain curve selected for nonlinear analysis.

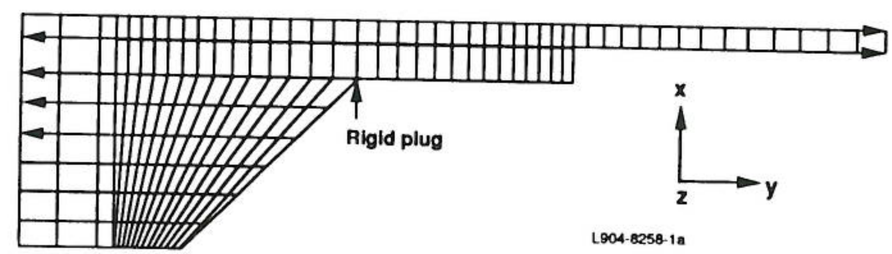

Figure 4. Discretized finite element model used for nonlinear analysis.

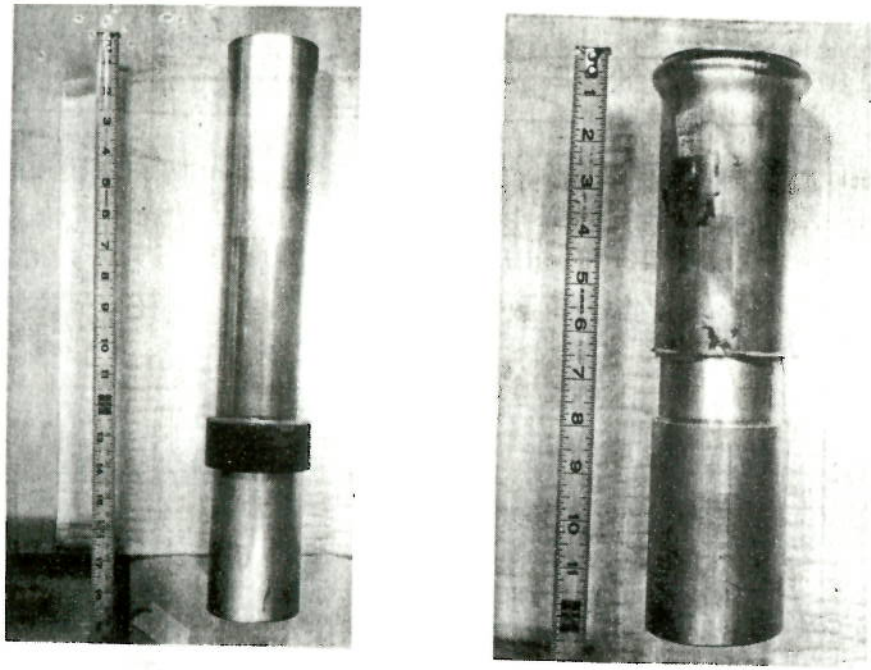

Figure 5. Prototype joint (a) before test and with teflon sleeve and (b) after test showing tubular buckling.

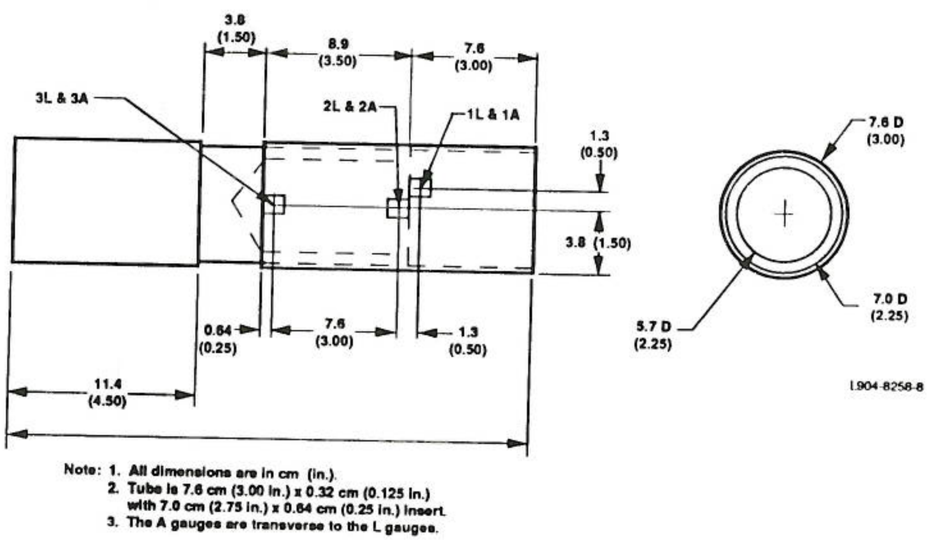

Figure 6. Strain gague locations on model joint.

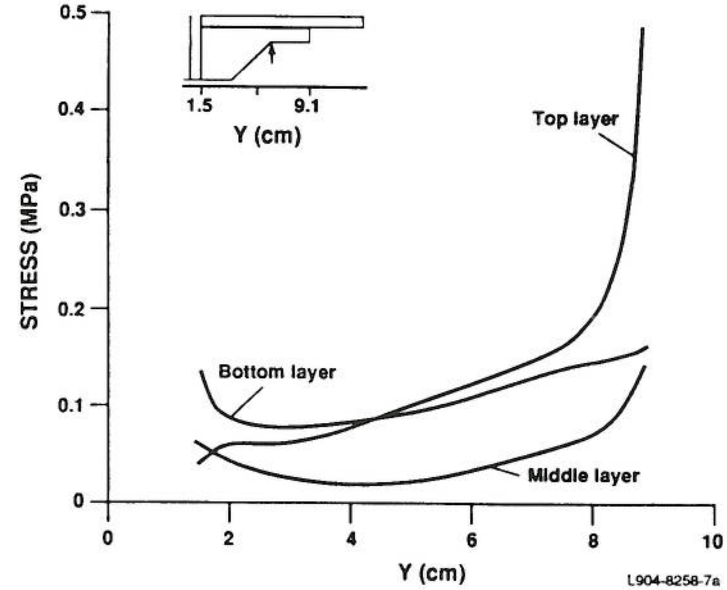

Figure 7. Von Mises's stress in the bondline of the unmodified joint geometry, Case 1.

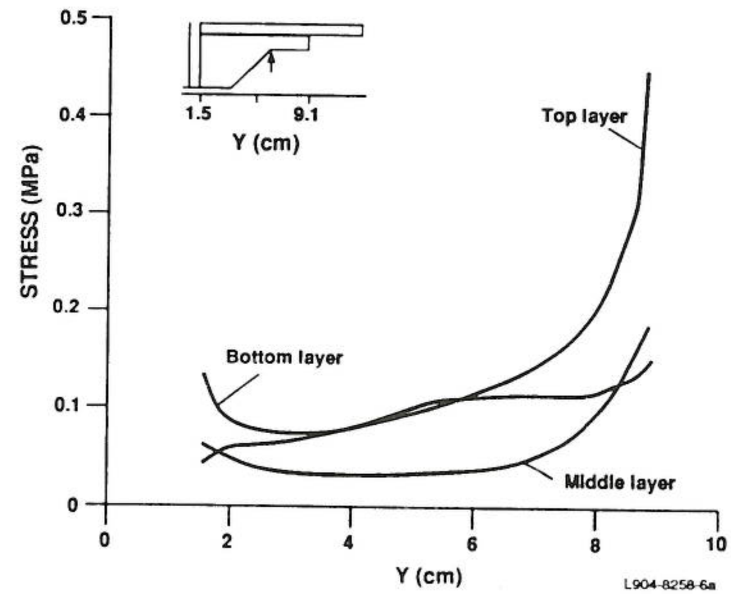

Figure 8. Von Mises's stress in the modified joint geometry incorporating a rigid plug within the outer tube, Case 2.

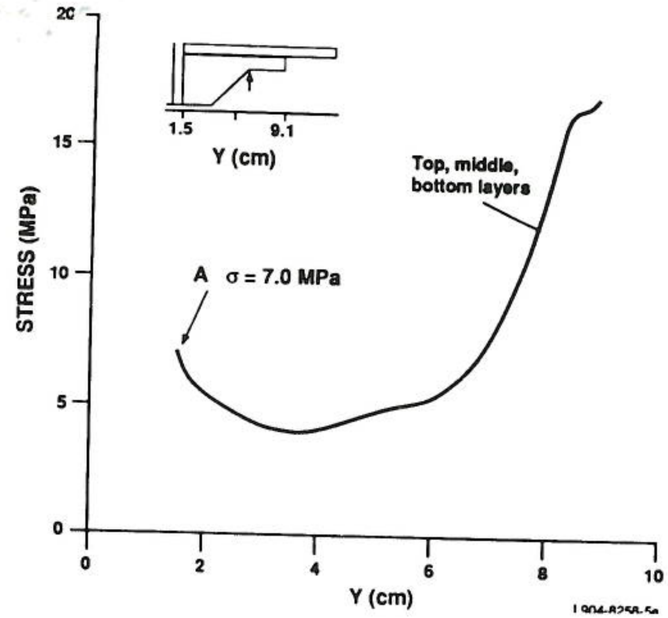

Figure 9. Von Mises's stress for nonlinear analysis; final load increment.

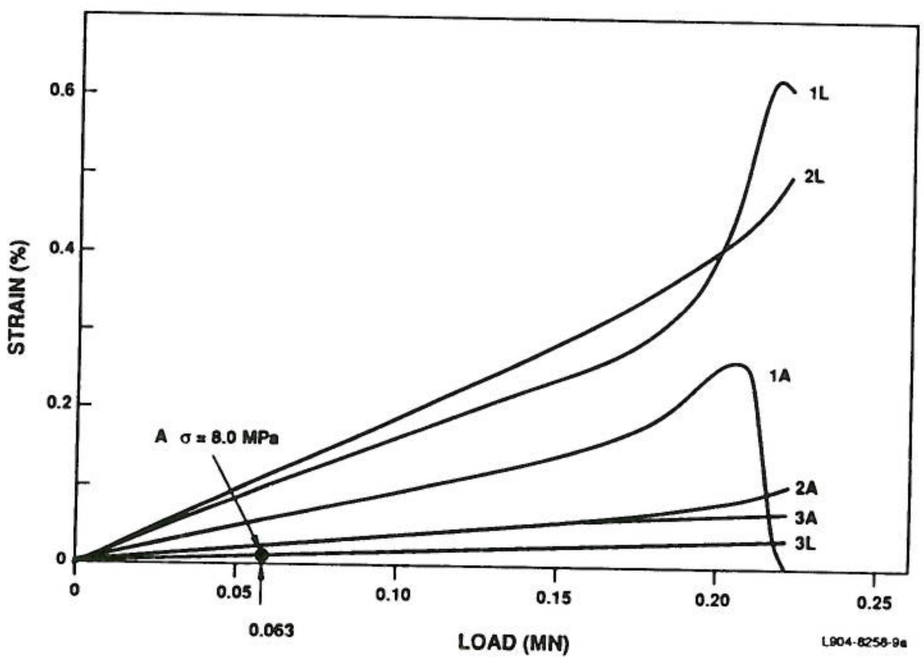

Figure 10. Strain in the model joint at different gague locations.

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HIGH-TEMPERATURE ADHESION PERFORMANCE OF SILICONE ADHESIVES

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ABSTRACT

As systems are designed for ever more demanding applications, the temperature constraints on the system components can become a limiting service consideration. For structures using adhesives to bond components together, these adhesives can become the temperature-limiting materials. Bonding performance for organic adhesives usually degrades above some critical temperature, after which small increases in service temperature can significantly increase the sensitivity of the bonds to delamination from even small stresses. Bonds formed with silicone adhesives also experience loss of bond strength with increasing temperature, but at significantly lower rates than those formed with nonsilicone adhesives.

This resistance of bond strength to increasing temperature, coupled with inherently superior thermal stability, allows silicone adhesive performance to extend to much higher temperatures than other adhesives. This improves the reliability of bonded components against sudden and possibly catastrophic bond failure. This paper compares adhesion performance of silicone and organic adhesives subjected to temperature extremes (-50 to +250°C) and high moisture conditions. Further improvements to the high-temperature bonding performance of some silicone adhesives with the use of a novel surface-activating agent are also presented.

KEYWORDS: Adhesives; Applications; Bonding; Delamination; High Temperature; Reliability; Silicones; Surface Treatment; Thermal Stability

1. INTRODUCTION

Choosing an adhesive for a given application requires intimate knowledge of both the application conditions and the performance of various adhesives under those conditions. As service temperatures rise, the performance characteristics of polymeric adhesives can rapidly change. When temperatures are near or below the glass transition of an adhesive, the material will be stiff, offering little elasticity to absorb bond stresses. Although materials at these temperatures may be quite strong, even small stresses to the bonded construction can cause delamination, and often catastrophic bond failure, as the stiff adhesive "pops" off the substrate.

Best bond performance is normally obtained at temperatures ranging between the glass-transition and melting temperatures for thermoplastic adhesives and between the glass-transition and thermal-degradation temperatures for thermoset adhesives. Once maximum bond strength is reached, adhesive performance usually begins to degrade with increasing temperature. Upper service-temperature limits are normally defined as those temperatures where adhesive perfor-