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PRIMARY ADHESIVELY BONDED STRUCTURE TECHNOLOGY (PABST)

Design Handbook for Adhesive Bonding

DOUGLAS AIRCRAFT COMPANY
MCDONNELL DOUGLAS CORPORATION
LONG BEACH, CALIFORNIA 90846

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The Air Force Primary Adhesively Bonded Structure Technology (PABST) Program was started in February 1975 for the purpose of validating the concept of adhesive bonding primary fuselage structural members together in lieu of the use of rivets. The goal of the Program was to provide a cost saving of 20 percent of the acquisition and maintenance cost of ownership. It was further desired to show a 15 percent weight saving for the bonded assemblies. The Program is now complete and the damage tolerance and durability of bonded structure has been shown to exceed, by far, that of riveted structure. Also

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the cost and weight savings goals appear to be obtainable for a production article. The design guidelines for a bonded primary structure are presented. The complete design criteria are identified for successfully developing bonded primary structure. It includes the fatigue, damage tolerance, and fail-safe criteria for both the metallic structure and adhesive bond area. A section discusses the material selection requirements, and another one presents the manufacturing considerations required to properly bond the primary structure. A review of the inspection needs is presented followed by a discussion of the repairability of the bonded structure. The cost and weight analysis review the methods used to determine results obtained on the Program. A complete section covers the loads, load transfer, and failure modes of bonded joints and this is followed by an analysis section which describes numerical methods for analyzing all manner of bonded joints with and without adjacent metal flaws. The final section in the report discusses the types of tests and associated specimens needed to properly select an adhesive for use in a bonded primary structure.

FOREWORD

This Handbook presents design information generated during the Primary Adhesively Bonded Structure (PABST) program, Contract F33615-75-C-3016. The effort described herein was performed by the Douglas Aircraft Company, Long Beach, California, a division of the McDonnell Douglas Aircraft Corporation, with Mr. W. W. Thrall, Jr., as the Program Manager.

This work was sponsored by the Air Force Flight Dynamics Laboratory (AFFDL) under joint management and technical direction of AFFDL and the Air Force Materials Laboratory (AFML), Wright-Patterson Air Force Base, Ohio. This contract is administered as a part of the Advanced Metallic Structures, Advanced Development Programs (AMS ADP), Program Element Number 63211F, Project 486U. Mr. William R. Johnston is the Acting Program Manager and Mr. William L. Shelton is the Project Engineer (AFFDL/FBA) for the PABST program.

This work was performed during the period 15 March 1977 to January 1979.

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GLOSSARY

ADHEREND - An object bonded or to be bonded to another object by an adhesive.

ADHESION - The state in which two surfaces are held together by interfacial forces which may consist of valence forces, interlocking action, or both.

ADHESIVE - A substance capable of holding materials together by adhesion that is capable of transmitting significant structural loads.

AUTOCLAVE - A closed vessel for producing an environment of fluid pressure, with or without heat, to an enclosed object while undergoing a chemical reaction or other operation.

AUTOCLAVE BONDING - A process similar to the pressure bag technique. The layup is covered by a pressure bag, and the entire assembly is placed in an autoclave capable of providing heat and pressure for curing the part. The pressure bag is normally vented to the outside.

B-STAGE - An intermediate stage in the reaction of an adhesive in which the material softens when heated and swells on contact with certain solvents but does not entirely fuse or dissolve. Uncured film adhesives are generally in this stage.

BEADS - Spheres used immediately in contact with the layup during the curing process to transmit normal pressure on surface of parts being bonded.

BINDER - A component of an adhesive composition which is primarily responsible for the adhesion of the bond.

BLEEDER CLOTH - A nonstructural layer of material used in the manufacture of bonded parts to allow the escape of excess gas and resin during cure. The bleeder cloth is removed after the curing process and is not part of the final bonded assembly.

BLISTER - A local elevation of the surface of an adherend, somewhat resembling the shape of a blister on the human skin, that may even burst and become flattened.

BLOCKING - Undesirable adhesion between two adhesive-coated surfaces in contact with each other. This may occur under moderate heat or pressure during storage, handling or application.

BOND - An adhesive, cementing material, or fusible ingredient that combines, unites, or interfaces between adherends.

BOND STRENGTH - The unit load applied in tension, compression, flexure, peel, impact, or shear required to break an adhesive-bonded assembly with failure occurring either within the adhesive, or at the adhesive-adherend interface.

C-STAGE - The final stage in the reaction of an adhesive, in which the adhesive is relatively insoluble and infusible. Generally, an adhesive that is fully-cured is in this stage.

CAUL PLATES - Smooth metal plates, free of surface defects, approximately the same size and shape as a bonded layup, used immediately in contact with the layup during the curing process to transmit normal pressure on surface of parts being bonded.

CO-BONDING - The act of joining together, by the process of adhesive bonding, two or more parts or assemblies during the same bond cycle.

COHESION - The state in which the particles of an adhesive or in general a single substance is held together by chemical forces.

CRAZING - A network of fine cracks extending on or under the surface of, or through a layer of, adhesive.

CREEP - The dimensional change, with time, of a material under load, following the initial instantaneous elastic or rapid deformation.

CURE - To further change the physical properties of an adhesive irreversibly by chemical reaction.

CURING AGENT - That part of a two-part adhesive which combines with the resin (binder) to produce a cured adhesive film.

CURE STRESS - A residual internal stress produced during the curing cycle when different materials (e.g., aluminum and titanium) of a bonded layup hav different thermal coefficients of expansions.

DAMAGE TOLERANCE - The ability of the airframe to resist failure due to the presence of flaws, cracks, or other damage for a specified period of unrepaired usage. (Ref. MIL-STD-1530A)

DELAMINATION - The separation of layers in a laminate.

DISBOND - A lack of proper adhesion in a bonded Joint. This may be local or cover a majority of the bond area. It may occur at any time in the cure or subsequent life of the bond area and may arise from a wide-variety of causes.

DURABILITY - The ability of the airframe to resist cracking (including stress corrosion and hydrogen induced cracking), corrosion, thermal degradation, delamination, wear, and the effects of foreign object damage for a specified period of time. (Ref. MIL-STD-1530A)

ENVIRONMENTAL CONDITIONING - Exposure of the test specimen to conditions or influences that are representative of expected usage environments but are not normal to laboratory test environments.

FAYING SURFACE - That surface of an assembly that interfaces with the surface of another assembly.

FLASH - Adhesive extruded from the edges of a joint after curing.

INTERFACE - The surface forming a common boundary between two contacting parts.

JOINT, BONDED - That part of a structure at which two adherends are held together with a layer of adhesive.

JOINT, LAP BONDED - A joint made by placing one adherend partly over another and bonding together the overlapped portions.

LAMINATE - A product made by bonding together two or more layers of materials with an adhesive.

LAYUP - A process of fabrication involving the placement of successive layers of materials.

MATERIAL SPECIFICATION - Document listing the physical and mechanical properties and performance requirements of a material.

PEEL STRESS - Bond strength in pounds per inch width when two adherends are joined and then separated by peeling and recording the strength value.

PLASTICITY - A property of adhesives that permits permanent and continuous deformation without rupture, upon the application of a force that exceeds the yield value of the material.

POLYMERIC MATERIAL - Material consisting of large molecules of high molecular weight, formed by the reaction of simple molecules (monomers) having functional groups which permit their combination to proceed to high molecular weights under suitable conditions such as heat and pressure.

POROSITY - A condition of trapped pockets of air, gas, or void within a bond.

PRIMER - A coating applied to a surface before application of an adhesive to improve the performance of the bond.

SCRIM - A carrier cloth or reinforcing fabric woven or unwoven into an open mesh construction, used in the processing of B-stage adhesive films to facilitate handling.

SECONDARY BONDING - The joining together, by the process of adhesive bonding, of two or more parts or assemblies previously bonded.

SET - To convert an adhesive into a fixed or hardened state by chemical or physical action. (See also cure)

SHELF LIFE - The length of time a material, substance, or product can be stored under specified environmental conditions and continue to meet all applicable specification requirements and/or remain suitable for its intended function.

SLIPPAGE - Undesired movement of the adherends with respect to each other during the bonding process.

SPECIFICATION - A statement containing a minute description or enumeration of particulars.

TACK - The property of an adhesive that enables it to form a bond of measurable strength immediately after the adhesive and adherend are brought into contact under pressure.

VACUUM BAG MOLDING - A process in which the layup is cured under pressure generated by drawing a vacuum in the space between the layup and a flexible sheet placed over it and sealed at the edges.

VOID - The absence or lack of adhesive in a bonded area.

VISCOSITY - The property of an adhesive to frictionally resist internal flow that is directly proportional to the applied force.

WARP - A significant variation from the original, true or plane surface.

LIST OF SYMBOLS

A_{str} = stiffener cross-sectional area

a = half crack length

b = extent of disbond on each side of discontinuity

d = extent of adhesive plastic zone, adjacent to discontinuity

da/dN = crack propagation rate - inches/cycles

E_1, E_2, E = Young's modulus for adherends

e = 2.718281828, base of natural logarithms

F_y = yield stress

G = adhesive shear modulus (elastic)

K_c = fracture toughness coefficient for cracked sheet

l = effective extent of adhesive load zone adjacent to discontinuity

ℓ = total overlap of the bonded joints

ln = natural logarithm

p = stiffener pitch

s = panel width

T_1, T_2 = adherend loads per unit width

ΔT = temperature differential ($T_{operating} - T_{cure}$)

t_1, t_2, t = adherend effective thicknesses per unit bond width

t_{skn} = thickness of sheet

w = width of bond

x, y, ξ, χ = coordinates

α_1, α_2 = adherend (metal) coefficients of thermal expansion

γ = adhesive shear strain

γ_e, γ_p = adhesive elastic and plastic shear strains

Δ = displacement (sheet distortion)

δ = opening of crack in sheet

δ_1, δ_2 = adherend displacements across bond line

ν = Poissons ratio

η = adhesive layer thickness

λ = exponent of adhesive shear stress distribution

π = 3.141592654, Pi

σ_1, σ_2 = adherend stresses

τ = adhesive shear stress

τ_p = adhesive plastic (maximum) shear stress

Subscripts

b = pertaining to bond

c = pertaining to crack in sheet

e = elastic

i = center adherend

o = outer adherend

p = plastic

skn = pertaining to sheet (skin)

str = pertaining to stiffener (stringer)

Superscripts

o = identified with discontinuity

∞ = identified with conditions far away from discontinuity

tot = total

SECTION I
INTRODUCTION

1.1 Background

The use of adhesive bonding in components of aircraft structure has been confined primarily to secondary structure. Prior to extending the use of adhesive bonding to primary structure, problems with adhesive durability, inspection, and effects of defects had to be resolved.

The one classical problem on existing aircraft with adhesively bonded structure has been environmental degradation at the bond interface. Moisture entering the bond through edges and around fasteners has caused surface exfoliation and crevice corrosion. This is associated with inadequate surface preparation and protection and presence of clad in the bondline.

Extensive government and industry exploratory development programs have resulted in improved adhesives, primers, and surface preparation and treatments. In addition, non-destructive inspection and manufacturing techniques for adhesive bonds have been vastly improved.

These developments led to an Air Force initiated advanced development program called "PABST" (Primary Adhesively Bonded Structure Technology) where Douglas Aircraft Company, a Division of McDonnell Douglas, was prime contractor (Ref. Contract No. F33615-75-C-3016). The objective of this program was to demonstrate and provide final validation that adhesive bonding could result in substantial cost and weight savings when compared to conventional fabrication techniques, while providing significant improvements in structural integrity and durability. To date the program has shown that the improved adhesive bonding systems have an outstanding potential for reduced fatigue damage and corrosion, and consequently a greatly increased durability.

1.2 Purpose

The purpose of this Design Handbook is to provide to the designer guidance and a basic understanding of the principles of bonded design that are applicable to primary adhesively bonded structure for flight vehicles, both military and commercial. The handbook is intended for the designer who is experienced in conventional construction and in the basic principles of structural design utilizing mechanical fastening methods. The methods and procedures developed for bonded structure are based principally upon test results and analytical capabilities generated in the PABST program. To assure maximum structural integrity it is recommended that a test program be utilized to demonstrate and validate each design of a new adhesively bonded structural concept as is generally done for conventional structure.

1.3 Scope

The elements that a designer must consider to efficiently design a bonded structure are discussed in this handbook. Any data presented reflects the current status of adhesive systems. Specific data required for design, such as material properties, analytical methods, and test results are best obtained from the Reference material shown within this handbook.

Proper attention to the material herein presented can minimize cost and save weight while providing structural integrity and useful life for an adhesively bonded structure.

A schematic representation of the inter-disciplinary relationships for the proper design of bonded structure is shown in Figure 1.1.

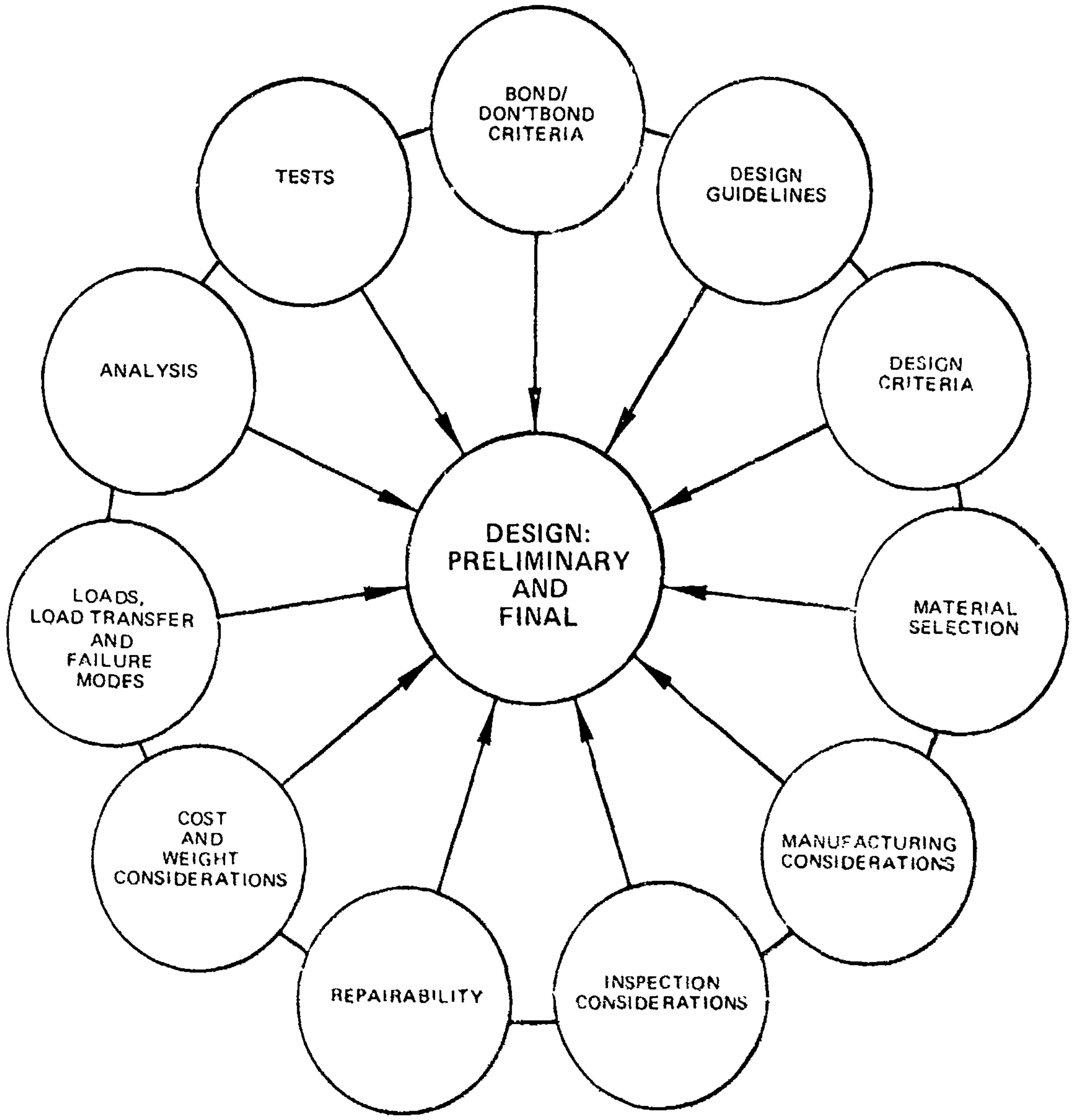


FIGURE 1 DESIGN PROCESS FOR BONDED STRUCTURE

SECTION II

BOND/DON'T BOND CRITERIA

Adhesive bonding provides an alternative to mechanical fastening in the joining of detail parts to make a structure. Compared with mechanical fastening, it has both advantages and limitations. To obtain the best from bonding, the designs must be tailored for that method of assembly and often will not look like an equivalent riveted design.

The justification for using structural adhesive bonding derives from one or more of the following advantages:

(1) reduced manufacturing cost, (cheaper structure)

(2) reduced maintenance cost, (more durable structure)

or (3) improved structural efficiency. (lighter structure)

These three basic advantages over purely mechanically-fastened structures derive from specific factors such as the following:

(a) fastener holes, a source of cracking, are eliminated.

(b) a bonded stiffener is more effective than a riveted stiffener in holding the crack tip shut in an attached cracking sheet.

(c) corrosion is reduced since the holes and metal-to-metal faying surfaces associated with fasteners are eliminated.

(d) sonic fatigue resistance is improved since the bond provides greater support around the boundaries than riveted construction and decreases the panel size by the amount of the bond width.

(e) the capability to resist repeated loads; e.g., buffeting in aircraft, is improved by bonding finger doublers onto the skin, and

(f) residual strength, including the ability to withstand foreign object damage, is improved by properly designing the bond width, stiffener spacing and area and selecting the bond material.

There are also some weaknesses with respect to mechanically-fastened structures and the disadvantages include:

(a) a bonded stiffener may, under some very special conditions, become completely unfastened, unlike a riveted one. The stiffeners in a test panel were disbonded under uniaxial load. No disbonds were induced in the curved panels or the full scale demonstration under conditions of biaxial stress and large metal flaws.

(b) there will be a higher local stress in a continuous member bonded to a discontinous member with no tapering than for riveted construction, thereby increasing the chance to initiate a fatigue crack. Local doublers will correct condition.

(c) adhesive bonding is stiffer than riveting and fails at a lower relative displacement between the members being joined. Consequently, bonded structure is less forgiving with respect to poor detailing, load redistribution, and damage tolerance than is conventional riveted structure. This demands a great level of understanding of and proficiency in the design of bonded structure than for riveted structure.

Since these various advantages do not apply in every case, the bond/don't bond decisions rest upon identifying the circumstances under which they do. The following list outlines some of the more crucial trade-offs in the choices.

° The structural advantages are most pronounced for thin structure, decreasing steadily for progressively thicker structure. The effective limit for structural bonding with typical ductile adhesives for subsonic aircraft is about 0.125 inch thick aluminum for double-lap joints and 0.063 inch thick for single-lap joints. This limit can be extended slightly by more elaborate joint details at a higher cost but at about a thickness of 0.25 inch the bonds are simply not as strong as the members being joined (See Figure 2.1)

° Bond and rivet should be considered as a viable option along with either pure bonding or entirely mechanical fastening. Despite the apparent (and often real) extra cost from two operations, judiciously located fasteners can serve as invaluable tooling aids for bonding and thereby reduce cost. Also, the bonding of reinforcing doublers around the perimeters of subassemblies to permit a reduced operating stress where those subassemblies are mechanically joined to each other can provide an enhanced fatigue life. Rivets can provide a fail-safe load path for a bonded joint.

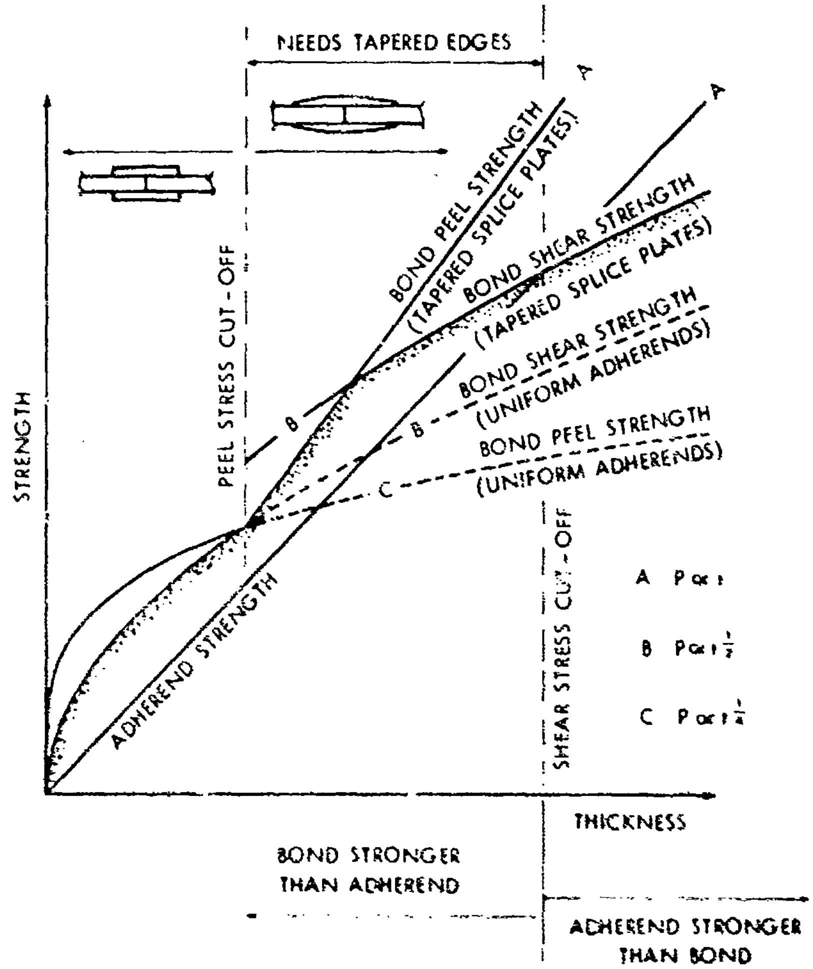


FIGURE 2 EFFECT OF ADHEREND THICKNESS ON BOND STRENGTH

° Lightning strikes could be a problem for an all bonded fuselage. A 100 Ka lightning strike across a bonded longitudinal joint on a small specimen gave small physical damage at the corners of the doublers. It is not certain that large bonded panels, which are mechanically fastened at the extremities, would be so affected. Another problem with lightning strikes is the inducement of electromagnetic fields between the bonded units whether they are doubler splice plates or wide area doublers. The extent of this problem and its final solution has not been determined.

° The economic production of bonded assemblies relies upon the flexibility of at least one of the two detail parts being joined. This permits the use of small deflection of the parts to improve the fit and uniformity of bond layers. Without such relief, rigid detail parts bonded together must be far more precise than equivalent parts that can be forced together by the much greater forces associated with large mechanical attachments. Therefore, as a general rule, adhesive bonding is not appropriate if the structural requirements prohibit any light flexible details to absorb tolerances.

° Adhesive bonding offers the possibility of reduced maintenance costs largely as the result of diminished, or eliminated, corrosion and fastener holes. When the surface preparation is good and the adhesive is environmentally resistant, the resulting adhesive bonding ensures the elimination of faying surface corrosion. The decrease in number of holes reduces the number of sites to initiate exfoliation corrosion. Bonded structures are inevitably less prone to fretting than riveted ones. Much of the past adverse experiences with bonding and corrosion are due to the combination of inadequate or improper surface preparation on clad alloys with adhesives which absorb water. The water then separates such adhesives from the metal and then induces crevice corrosion. This characteristic problem has not been experienced on the PABST program and has not been seen in many old adhesives which have given satisfactory aircraft service for decades. Good adhesive bonding not only transfers load between the various details but also resists corrosion. To obtain a comparable resistance to adverse environments from a riveted design requires a thorough application of faying surface sealant, which adds to the cost as well as the life. Indeed, the use of an adhesive as a sealant in conjunction with mechanical fasteners has been shown to be very effective in reducing leakage from the fuel tanks of both transport and fighter aircraft.

° In trying to resolve a choice between producing a bonded or a riveted structure, one should prepare appropriate layouts for each method of manufacture. The competitive designs will not necessarily look alike, so an assessment of the relative costs of riveting and bonding the same design may not be appropriate. Under current conventional manufacturing methods, the cost of the bonded structure may be sensitive to the size of available autoclaves and the desired production rate. It should be remembered that non-autoclave tools can be built to bond large panels. Also standard sheets widths can be bonded together to yield wider finished bond assemblies than are possible when using more costly premium width sheets. These factors influence the size and number of the subassemblies which are to be subsequently mechanically fastened to each other. Within each such panel, however, the use of hand or automatic riveting has a profound impact on cost. Therefore, the available manufacturing facilities (both direct and by sub-contracting) play a big part in influencing the choice of manufacturing method. Likewise, the relative expertise of the designers for bonding and riveting should also be considered.

° The repair method for maintaining the structure in service influences the operating stress levels of the intact structure. The need for fail safety of a partly damaged structure also restricts the operating stress levels of the intact structure. Furthermore, an adhesively bonded structure cannot withstand a sustained load above the yield strength of the metal, because the bond then creeps, eventually resulting in progressive failure. On the other hand, a corroded riveted structure in service must be proportionately weaker than an equivalent uncorroded bonded one. Therefore, one must employ the correct governing conditions in assessing any weight savings attributable to bonding instead of riveting. A comparison on the basis of a single application of ultimate load to the intact structure is not sufficient.

Bonding technology which reflects the 1979 state of the art can be reduced to a series of key words or catch phrases.

- Clad is bad
- Prep is paramount
- Control it
- Chromates are critical
- Demand durability
- If you're going to do it, DO IT RIGHT.

Specifics can be found in the body of this report.

SECTION III
DESIGN GUIDELINES

3.1 Basic Approach

The following guides are necessary for the design of successful bonded structure and should be implemented at the start of and throughout the design phase. Proper attention to these guidelines will minimize cost while providing structural integrity to a bonded structure.

- The manufacturing methods, bonding tool(s), and structural arrangement must be established simultaneously, not sequentially nor independently because some decisions in one discipline preclude decisions in other disciplines, refer to Section VI.

- The primary objective, when designing a bonded panel, is to ensure that the bond line never becomes the critical link under any load or failure condition, refer to Section 10.3.2.4.

- Recognize that the adhesive bonds are not uniformly loaded, refer to Section 10.3.2.1 and 10.3.2.

- Bonding is most suitable for thin detail parts and becomes progressively less suitable for thicker members, for which mechanical fastening is therefore needed, refer to Section 10.3.2.4.

- Bonds are best when loaded in shear, refer to Section 10.3.2.3.

- Peel loads on bonds should be avoided or minimized, refer to Section X.

- Non-linear analysis must be used for adhesive-bonded joints, refer to Section 10.3.2.4(c), and 11.5.3.

- The degree of inspection should be in proportion to the local intensity of bond load, refer to Section VII.

- Lightly loaded adhesive areas should not be eliminated completely by minimizing overlaps, refer to Section 10.3.2.1.

- Adhesive bonds cannot endure sustained loads intense enough to yield the metal - the bonds then fail progressively, refer to Section 10.5.

- Bonds and rivets should not be looked upon as interchangeable. A bonded stiffened panel probably should not look like an equivalent riveted design, refer to Section 6.1.

- Bond-line thickness should be controlled to the same level of uniformity as is demanded of mechanical fasteners, refer to Section 6.1, 10.3.2.4(c) and 10.6.1.

- Fasteners are better than bonding in some instances, so arbitrary goals of excluding all fasteners should be resisted, refer to Section 10.5.

- The use of hot-bonded edge doublers in combination with mechanical fasteners and sealant at manufacturing breaks can develop more than adequate fatigue lives, more economically, with a much smaller number of fasteners than a purely mechanical splice, refer to Section 10.3.3 and 10.4.3.

- In selecting the adhesives and the processing methods, it is more important that the adhesive remain attached for the entire service life than that the adhesive has the highest strength but falls off prematurely, refer to Section 10.3.2.4(c).

- Cold-set epoxy adhesives should not be used for anything other than temporary repairs - they will not last and simply absorb water and cause crevice corrosion, refer to Section VIII.

- Selected operating stress levels must permit repairs with rivets and sealants; or hot bonding in the field equivalent to the original autoclave bond will be required, refer to Section VIII.

° Multi-stage bonding should be avoided wherever possible, refer to Section 6.1.

° The aluminum alloy can affect bondability. It has been determined that the clad 7000 Series aluminum alloys have very short life in a moist atmosphere. Therefore, you should not attempt any bonding of the clad 7000 Series alloys. Non-clad 7000 and 2000 Series alloys are very durable after surface preparation. Metal-to-metal bonding with clad 2000 Series alloys has been satisfactory but could be a problem when bonding to honeycomb core.

° Use of improved processing for surface preparation and a moisture resistant adhesive and primer is critical. Refer to Section V.

3.2 Interfaces With Other Disciplines

The design and manufacture of adhesively bonded structure cannot proceed in isolation of other disciplines. Some known constraints are outlined below.

While some simple structures can be designed to be economical as highly stressed parts to be thrown away and replaced if damaged, most structure must be designed more conservatively to permit repairing. This topic is discussed in Section VIII.

Conventional mechanical joining of metal details provides a continuous electrical load path for the entire structure. This is then used as a common ground (return) line for almost all electrical circuits. It also renders the structure relatively immune to lightning strikes and does not develop differential potentials on the outer surface which would interfere with navigation and/or communication systems. With an all-bonded structure, however, the metal elements would be electrically isolated. The only conductive adhesives known today have poor mechanical properties, so the present state of the art requires at least some rivets to provide continuity. One should coordinate the selection of the sites of those rivets with the tooling function (to aid in manufacture) and the stress analysis (to exclude fastener holes from highly stressed areas.)

Environmental considerations dictate special treatment for the high temperatures caused by the wake from jet engines. If the temperature of the bonded part exceeds about 200°F, the strong ductile adhesives used at lower temperatures must be replaced by a brittle adhesive to survive that environment.. The other commonly occuring environmental effect is that of acoustic fatigue which affects some structure. The tremendous number of load cycles so generated require special attention (with fingered doublers and/or honeycomb stabilization) to develop an adequate structural life.

A less obvious constraint on design of bonded structures comes from acoustic considerations. The use of honeycomb and bonded finger doublers to enhance structural fatigue life is well known. However, particularly for the fuselages of passenger aircraft, the transmission of engine and/or aerodynamic noise must be restricted so as not to cause discomfort inside. Some solutions to this aspect of the problem are found by tuning the natural frequency of the panels or structure. But others require mass to absorb the noise. In this latter case, it makes no sense to save structural weight by expensive sculpturing of skins and the like only to have more lead lining added to absorb the noise. Consideration of the noise problems early enough would save such unnecessary costs.

The constraints of manufacturing facilities and techniques are discussed in Section VI. It suffices here to note that the availability of facilities and equipment for processing, fabricating, and assembling details affects the breakdown of the structure into subassemblies. The production rate, in terms of those same facilities, can also affect that breakdown if the rate is sufficient to exceed some capability. Redesign may be necessary to alleviate such potential problems, which should be anticipated or uncovered during the design rather than after.

3.3 <u>Joint Configuration Definitions</u>

Various bonded joint geometries and the nomenclature used throughout this report are identified in Figure 3 . As a general rule, the word "double" in the joint title implies a symmetric joint, in which the load is shared between two or more adhesive bonds, with no out-of-plane deflections. The

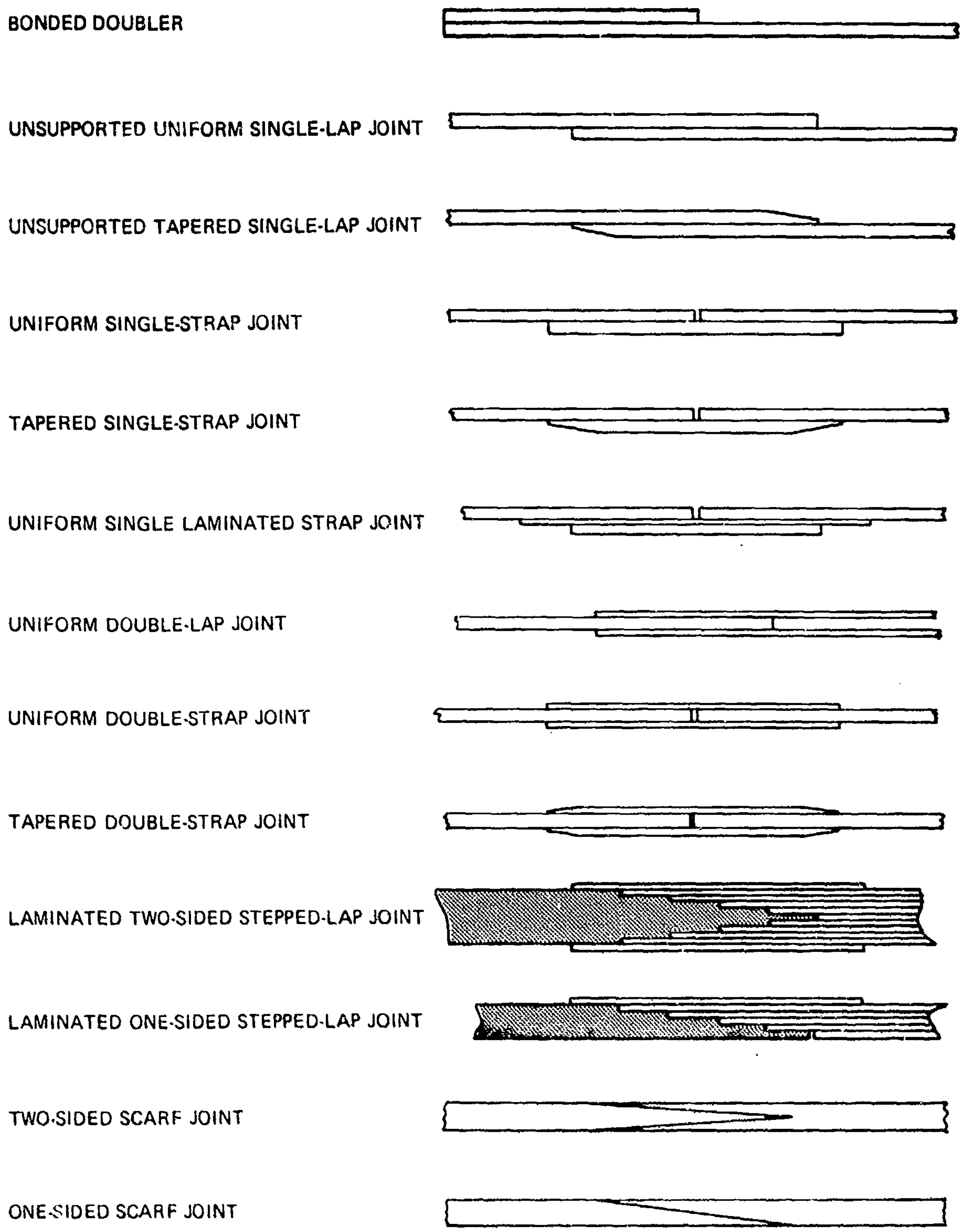


FIGURE 3 IDENTIFICATION OF BONDED JOINT CONFIGURATIONS

word "single" is associated with joints having all the load passing through one or more adhesive bonds all offset to one side of the primary load path. This implies as asymmetric configuration (with respect to the plane of the sheets being bonded) and, unless lateral support is supplied externally, such joints will deflect out-of-plane because of the eccentricity in load path. This induces adherend bending moments and adhesive peel stresses.

SECTION IV
DESIGN CRITERIA

The criteria for an adhesively bonded component of an airframe must contain the requirements of applicable military aircraft specifications with appropriate modifications that are peculiar to the adhesively bonded structure. These specifications include the MIL-A-008860 Series, MIL-STD-1530 (USAF) and MIL-A-83444 (USAF) documents. The intent is that the implementation of this criteria in the bonded structure will result in a structural integrity equivalent to that required for airworthiness. The implementation must be demonstrated by test and analysis as necessary.

The design criteria for conventional, mechanically fastened structures, is well known and will not be considered any further here. Attention will be focused on the requirements of bonded construction with particular emphasis on fatigue and damage tolerance. The PABST program was one of the first structural airframes designed to the requirements of the Air Force's damage tolerance specification, MIL-A-83444. Slow crack growth, uninspectable structure was the category selected for the PABST design from the MIL-A-83444 specification. Additional civil aviation residual strength requirements were also imposed.

4.1 Applicable Documents

The following documents apply to the design of the PABST Full Scale Demonstration Components. Certain deviations were taken to these specifications and are detailed in Reference 9. In any design, deviations to applicable documents must be spelled out in detail and approved by the contracting agency.

MIL-STD-1530 (USAF) "Aircraft Structural Integrity Program, Airplane Requirements" (1 September 1972).
MIL-A-8344 "Airplane Damage Tolerance Requirements."
MIL-A-008866A "Airplane Strength and Rigidity, Ground Tests."
MIL-A-008867A "Airplane Strength and Rigidity, Reliability Requirements, Repeated Loads and Fatigue."

4.2 Fatigue Criteria

The fatigue criteria shall incorporate an utilization model considering all pertinent loadings arising from preflight taxi, post-flight taxi including effects of reverse thrust, landing impact, vertical and horizontal gusts, flight maneuvers, pressurization, thermal loads, ground handling loads and the influence of the environment on the strengths of the various materials.

4.2.1 Service Life

The design service life and design usage of the airplane under consideration, exclusive of scatter factor, must be established. A typical utilization, basic to the design of the PABST Full Scale Demonstration Component, is shown below.

Flight Service Life	30,000 Hrs., 12,507 Flights & 46,194 Landings
Pressurizations	19,014
Landings, Full Stop	29,977
Touch and Go's	16,127

The projected equivalent utilization for fatigue analysis of the PABST FSDC structure is given in Table 1 .

4.2.2 Design Fatigue Life

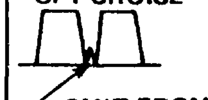
The design fatigue life is the service life defined above multiplied by a scatter factor of 4.0.

4.2.3 Service Loads and Environment Spectra

The basic inputs to define the cyclic loads spectra are defined in MIL-A-008861A and MIL-A-008866A modified to incorporate the higher sink rates associated with STOL type aircraft. Such modifications may be necessary in view of special applications necessary for specific aircraft under consideration. The procuring agency approval is necessary for such modifications. For the metal FSDC structure, the environment used was room temperature and laboratory air. For actual aircraft, representative environments must be used or otherwise accounted for.

TABLE 1

PABST UTILIZATION

FLIGHT NUMBER	DESCRIPTION	SERVICE LIFE: FLIGHT HOURS	PERCENT TOTAL FLIGHT HOURS	STOL	CTOL	LANDINGS: TOUCH AND GO	LANDINGS: TOTAL	LANDINGS: % TOTAL	NUMBER OF MISSIONS	PAYLOAD LB	
1-1	BASIC	20,007	66.7	7,236	7,236	7,236	21,708	47.1	7,236	20,250	OPT CRUISE  OMIT FROM FLIGHT 1-2
1-2		1,894	6.3		947	947	1,894	4.0	947	54,250	
		21,901	73.0	7,236	8,183	8,183	23,602	51.1	8,183		
2-1	BASIC TRAINING	1,974	6.6	1,234	1,234	7,404	9,872	21.4	1,234	20,250	15,000 FT 10,000 FT 
2-2		126	0.4	90	90	540	720	1.5	90	54,250	
		2,100	7.0	1,324	1,324	7,944	10,592	22.9	1,324		
3-1	LOW ALTITUDE RESUPPLY	3,000	10.0	4,500	1,500		6,000	13.0	1,500	27,000	500 FT  2 HOURS
3-2		3,000	10.0		6,000		6,000	13.0	1,500	62,000	
		6,000	20.0	4,500	7,500		12,000	26.0	3,000		
		30,001	100.0	13,060	17,007	16,127	46,194	100.0	12,507		

2(8183) = 16,366 FULL PRESSURE CYCLES
2(1324) = 2,648 PARTIAL PRESSURE CYCLES
19,014 PRESSURE CYCLES (ACTUAL UTILIZATION HAS 17,150 PRESSURE CYCLES).

4.3 Slow Crack Growth Damage Tolerance Criteria - Metallic Structure

PABST safety of flight structure was qualified as slow crack growth under the appropriate sections of MIL-A-83444 and designed so that the possibility of catastrophic failure was extremely remote. Compliance with these criteria involved residual strength and crack growth analysis and tests. In addition, the structural design and analysis accounted for the fail safe criteria in the following paragraph. In any future designs, it may be advantageous to qualify the structure as one of the following:

1) Slow crack growth structure
2) Fail-safe multiple load path structure
3) Fail-safe crack arrest structure

4.4 Fail Safe Criteria - Metallic Structure

The PABST FSDC structure was designed to fail safe capability comparable to that of commercial airplane fuselages, as defined in Federal Aviation Regulation 25. The fail safe requirements of MIL-A-83444 were not applicable since the structure was qualified to slow crack growth criteria.

The structure was to be capable of withstanding (1) limit load with a two bay crack and (2) the maximum average internal member load occurring in 20 lifetimes, or limit load whichever is less, for foreign object damage as specified in the following subsections.

4.4.1 Longitudinal Cracks

The structure with a longitudinal crack shall be able to withstand (1) a two-bay skin crack or a skin-to-longeron disbond and the center frame (or splice) intact, and (2) a 15 inch long foreign object damage skin crack with both the center frame (or splice) and crack arrest member (if present) failed. For the first requirement, at least the skin crack adjacent to a frame (or splice), where high stresses are induced from frame bending and pressure, shall be considered. All cracks considered shall be assumed to propagate in both directions.

4.4.2 Circumferential Cracks

The structure with a circumferential crack shall be able to withstand (1) a two-bay crack with the center longeron (or splice) intact, and (2) a 15 inch long foreign object damage crack with the longeron or splice and crack arrest member (if present) failed. All flaws shall propagate in both directions.

4.5 Damage Tolerance Criteria - Adhesive Bond Areas

4.5.1 General Requirements

The requirements of MIL-A-83444, for metal and mechanically joined elements shall be supplemented with the following requirements for the design of adhesive bonds joining two or more elements of the structure. Compliance with these criteria shall be demonstrated by analysis and/or test. The analytical damage tolerance assessment shall be confined to residual strength estimates. The analyses shall assume the presence of flaws in the bond placed in the most unfavorable location and orientation with respect to applied stress and material properties. The experimental investigation shall be limited to distinguishing between flaws which grow and those which do not. Thermal and humidity effects shall be accounted for.

Entire panels or parts which are improperly processed; i.e., parts with global damage, was to be rejected. Parts with local contamination or flaws would be reworked to a quality in which the flaws shall not grow to unacceptable sizes within two airframe lifetimes.

4.5.2 Initial Flaw Sizes

An initial flaw shall be assumed to exist in each and every bond in its most critical location including those highly stressed areas resulting from variable bondline thickness. The size of the flaw shall be the greater of (1) the minimum detectable size for the NDI technique used on the bond, or (2) the smallest flaw remaining after a larger flaw has been repaired. Each flaw shall be analyzed for residual strength independently of all other flaws, either in the bond or metal. Initial flaws shall be located so there is no interaction between them.

4.5.3 Bond Inspectability

The detail design shall minimize the use of uninspectable bonds and, wherever practical, shall be such as to force the first evidence of failure into a visible or easily inspectable area. Techniques, such as staggering the ends of the overlaps, shall be used to facilitate inspection of the bonds. Each uninspectable bond shall be limited in extent to a subcritical size.

4.5.4 Flaw Growth in Bonds

Flaws in bonds induced in service shall not grow from initial sizes defined above to critical size within two airframe lifetimes. All flaws large enough to grow in service shall be repaired prior to delivery of an aircraft to preclude corrosion. In addition, bonds which contain subcritical flaws in areas subject to corrosion shall be sealed to provide environmental resistance.

4.5.5 Fail Safe Capability

The fail safe capability of the bonded structure shall be demonstrated by test and/or analysis. The structure shall be capable of withstanding (1) limit load with each of the following two-bay disbond configurations:

1) A two bay disbond in only one side of a double strap butt splice,
2) A two bay disbond in a single strap butt splice, or single lap splice,
3) A two bay longeron-to-skin disbond, and
4) A two bay shear-clip-to-skin or crack-arrest-member-to-skin disbond;

and (2) the maximum average internal member load occurring in 20 lifetimes, but less than limit load, for impact or the foreign object damage specified as:

1) A 15" disbond on both sides of a splice, and
2) A 15" long foreign object damage skin crack with both the center frame (or splice) and the crack arrest member failed or with both the longeron (or splice) and crack arrest member failed as applicable.

SECTION V
MATERIAL SELECTION

In order to properly select the materials applicable to a bonded structure, the designer must consider the total bonding system. The structural configuration, design criteria, and the mission requirements such as environment, life and load spectra may well decide the process system as well as the metal selection. Tests at the coupon level and with more complex structural arrangements under simulated environmental and loading conditions expected in the structure during its service life should be made to validate the final bonding system. (See Section XII)

5.1 Adherends

Early in the PABST Program, tests to determine the durability and environmental resistance of the different alloys, cladding and of heat treatments were performed. The most significant item was the difficulty in producing a durable surface treatment on alclad 7075. The phosphoric acid anodize, which produced exceptionally durable surfaces on the 2000 series alloys and on non-clad 7000 series alloys, did not perform well on the alclad 7075. This lack of durability was also more significant when the adhesive system was a 250°F cure modified epoxy than when the system was a 350°F cure modified epoxy.

Durability of the finished product and processability are major considerations along with fatigue resistance, damage tolerance, weight and other criteria that influence the adherend material selection.

Thermal mismatch of adherends should be avoided. Bonding two materials that have different thermal expansion coefficients causes residual stresses after bonding and additional stresses (induced) as the temperature exposure varies regardless of cure temperature. High modulus of elasticity adhesives are more susceptible to these stresses. Further explanations of this problem may be found in Section X.

Prior to bonding assemblies, the details were assembled and run through a cure cycle with verification film used in place of the adhesive. This verification film formed to the assembly and reflected the variations in the glue line thickness that could be expected with any given set of details. Using this information, those details that could be reworked were reformed to give a closer tolerance to the proposed glue line. Those details, where rework was not feasible or economically practical, had the discrepancy noted on the bonding fabrication paper and in those areas, an additional layer of adhesive was used during the layup of the assembly. This procedure was used more often on complicated assemblies where prefitting was difficult and expensive. Some mismatch can be absorbed during the cure cycle because of the pressure used in the bonding process, but not all. If more than one additional layer is required, the designer should have supporting data on the acceptability of more adhesive since mechanical properties of any given bond joint will vary with the addition of extra adhesive due to variations in the glue line thickness.

Verifilming is a considerable "extra cost" operation in any bonding process. Its use should be restricted to original tool prove-out, first article assemblies, complicated assemblies where detail fit is not readily determinable by other means, and as a periodic check of possible tool wear or tolerance change during production. In the PABST Program the verifilm operation was used on all assemblies since all the assemblies could be considered first article. The verifilm was most valuable in showing where details slipped during the cure cycle, which necessitated minor modification of the tools, and indicating low or no pressure areas around the shear tee and longeron intersections. The verifilm also indicated the potential problems to be expected with large area doubler bonds.

5.2 Surface Treatment

The adherend surface may be prepared for bonding by employing a surface treatment process such as acid etching or anodizing. Anodizing is an electrolytic process and typically produces a surface as shown in Figure 4 . The anodic layer is delicate and subject to damage and for this reason must be adhesive primed soon after anodizing. The primer should rigidize this surface and protect it from contamination and damage.

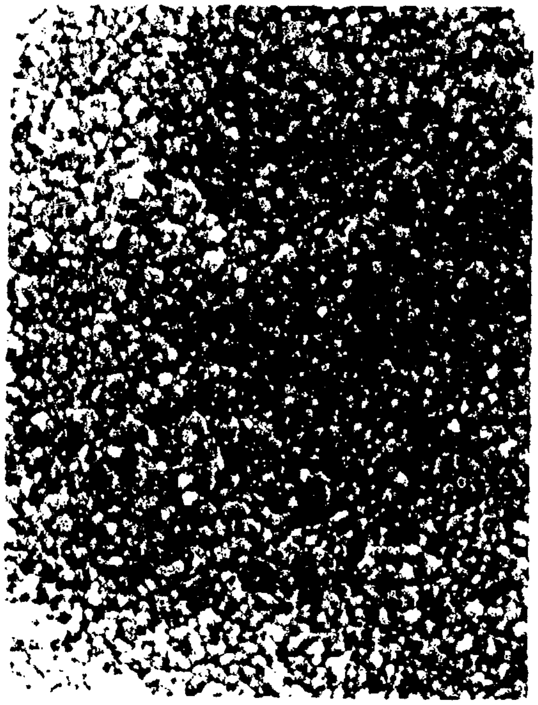


FIGURE 4 TYPICAL ANODIZED SURFACE 16000X

The surface treatment of the adherends is as important as part of the bonding system as the adhesive, the primer or the adherends themselves. An improper surface treatment can cause inadequate adhesion of the adhesive to the adherend. Even good adhesion at nominal temperatures and environment may not be adequate in a warm, humid exposure. Testing in these environments is the only method for assurance of a proper surface treatment. The alloy of the adherend may also influence the ability of a given surface treatment process to produce an environmentally resistant surface.

5.3 Adhesive Primers

Primer systems are used with adhesives for various reasons. Some primers activate and cleanse the adherend surface so that it is more receptive to the adhesive. Other primers contain corrosion preventers or inhibitors which are intended to resist corrosion of the metallic adherends. Primers also protect the cleaned surface of the adherends, which can be very reactive if anodizing or special etch processes have been used. A contaminated or disturbed surface can cause an inferior bond.

The primers can be special adhesive formulations themselves or just some of the parent adhesives dissolved in a suitable solvent. Many primers offer a good base for the application of protective coatings to the final article. Some primers may be applied and the adhesives located in place after only an air dry on the primer. Other primers are reliant on the hardener in the adhesive for curing and are sometimes called co-cure systems. Still other primer systems must be fully cured at elevated temperatures before the adhesives are applied. These cure times and procedures are important. Primers that are not dried or cured properly may trap solvents which could cause porous bond lines in the final article. They are also more susceptible to absorbing atmospheric moisture which will also degrade the bond line. The period betwen primer application and the final assembly should be controlled to reduce the contamination possibilities of the surface.

Over-cures of primers will make them brittle, change their mechanical properties, and make them less receptive to good bonding of the adhesive. The adhesive and primer manufacturers' instructions are usually backed by extensive tests. The parameters and tolerances should be tested by the using facility under production conditions. These tests should assist the user in making adjustments, as necessary, to the tolerances to optimize and reduce the variables in the operation.

5.4 Adhesives

Adhesives are available in several forms. They are generally a mixture of materials that give the desired design properties in the final bonded joint. The adhesives materials may be in several forms prior to usage; such as films, pastes, liquids, powders, etc. These materials are transformed to their desired bond joint form through catalization, drying, thermo induced molecular change, melting or other means including combinations of these types of processes. The characteristics of the desired design dictate the type and form of the adhesive to be used. The facilities and capabilities of the production area also have an effect on the form of the adhesive to be used. Film adhesive, the form used on the project discussed in this handbook, are within themselves furnished in several forms and resin types. Chemically, the film may consist of one or several materials such as epoxies, phenolics, polyamids, polyimids, silicone, etc. The generic type of resin usually dictates the type of properties the final bond will have such as flexibility, temperature resistance, fluid resistance, shear strength, etc. Modifications to any resin film system can be made to enhance handling properties, thermal expansion control, thickness control and other functions. These modifications include carriers of cloth or mat, and fillers of powdered metal or other inorganic materials. The carrier materials aid in handling the adhesive and in some cases, impart desirable properties such as peel strength and bond line thickness control. Fillers are used for oxidation control, flow control, stabilization of thermal expansion and bond line thickness control.

It is apparent from the above, that adhesives are available for almost any desired design.

The adhesive system used for testing in the PABST Program was a modified epoxy 250°F cure film with a mat carrier. The adhesive is nominally available in film weights of .03, .045, .06 and .08 pounds per square foot. The original testing in the Program utilized .045 pound per square foot material. After verifilming some of the larger test panels, it was decided to use the .06 pound per square foot material on all future testing. This change would help reduce the number of voids and additions of extra layers of adhesive in areas where the details may deviate from the design tolerances.

Adhesives that are supplied in films are usually fully compounded by the manufacturer and have the hardener already mixed with the base resin. The temperature of the cure completes the polymerization. The adhesive may still be susceptible to long term exposure at ambient temperatures and, therefore, to get the longest usable life of the material in the work area, low temperature storage is recommended (0°F or less).

Many controls are required on the adhesives and primers to guarantee the optimum properties in the final bonded assembly. After the original testing for material and process selection, the material requirements must be documented with adequate testing required to provide the desired material to production. The processing methods that have been developed must also be documented and should contain adequate control requirements for handling, storing, applying and curing of the adhesive. Continual monitoring of these materials, conditions and operations is necessary to consistently produce the desired end product.

Adhesives are perishable items and must be controlled in the production area. Some of the more serious problems in production are caused by contamination, moisture absorption and over ageing of the adhesive.

The selected controls for the system used in these tests were a storage temperature for the adhesive and primer of 0°F or below and an allowed time at ambient temperature of five days. Maintaining these controls gave assurance that the mechanical properties obtained on incoming receiving tests could be expected to be obtained in the final assembly.

The adhesive mechanical properties may be affected by variations in the time to rise to cure temperature and cure length. Testing will determine the temperature profile of the cure required for each adhesive system.

5.5 Sealants

Sealants are used for several purposes. The major use is to provide a hermetical seal between details. This seal can also perform the function of preventing corrosion in the joint by including corrosion inhibitors in the sealant. Sealant is also used to over coat exposed edges where additional resistance to impacts is needed over the painted surface. It also adds protection to the cut edge where the bond joint is exposed to the atmosphere. Another use is where attachments pass through bonded laminates. The protection is two-fold here since the sealant produces the desired hermetical seal and also prevents dissimilar metal contact.

Sealants are produced in many forms to allow several different methods of application such as spraying and troweling. They are also available with varying pot lives which can range from fast cures of a few hours to longer cure times which allow time for large assemblies to be mated before the sealant cures.

Some sealants have significant mechanical properties which lends them for use as adhesives. This is not to be construed as a recommendation to use these sealants as adhesives.used in conjunction with mechanical fasteners, these sealants can sifnificantly improve fatigue life.

5.6 Coatings

Coatings or paints are usually corrosion control materials. They also perform the function of decorating the assembly. Depending on the level and type of exposure, one or more coatings may be required for the protection desired. The mechanical and physical properties of the coatings can vary allowing the selection of flexible materials where flexing and movement can be expected in the assembly and fractures in the coating were undesirable. Very rigid coatings are available that have very good resistance to very hostile environments. The types of organic compounds used in the coatings helps determine the environments that they are most resistant to.

The adhesive primer is considered part of the coating system on the subject of this discussion. It forms a very good paint base and is resistant to all of the environment exposures expected by the typical cargo aircraft.

SECTION VI

MANUFACTURING CONSIDERATIONS

A typical manufacturing sequence for a bonded structure using a female tool is shown in Figure 5. The designer must be familiar with the manufacturing techniques available in order to optimize the design with respect to cost and structural integrity. In bonded structures, the selections of the structural arrangement, design philosophy of whether to use few or many details, the choice of panel and frame segment size, and the judicious location of holes are best made in conjunction with the selection of bonding method and sequence of assembly and none of these factors should be looked upon as the prerogative of any one discipline alone. This section discusses some of the manufacturing considerations that can impact the design.

6.1 Bonding

The designer should be aware of any special requirements of the adhesive system being used. For instance, some adhesives require a fast heat up rate. In this case, the structure and the tooling being used must be compatible with that heat up rate. Some adhesives require higher cure temperatures and/or longer cure times than others, which may reduce some physical properties of the metal parts.

For some complex structure two stage bonding may be desirable. In a two stage bonding process, two or more previously bonded assemblies are bonded together in a second bond cycle. During the initial bond cycle the adhesive has flowed and cured. During any subsequent cycle the adhesive will not flow again, but the bond strength is significantly reduced at the elevated bonding cure temperatures. If high preloads exist between the bonded details, a second bond cycle could allow relative movement of the details. Therefore, pressure should be applied to all of the details during the second cure cycle so that the chance of movement is eliminated. Adhesive properties are not affected by repeated cure cycles.

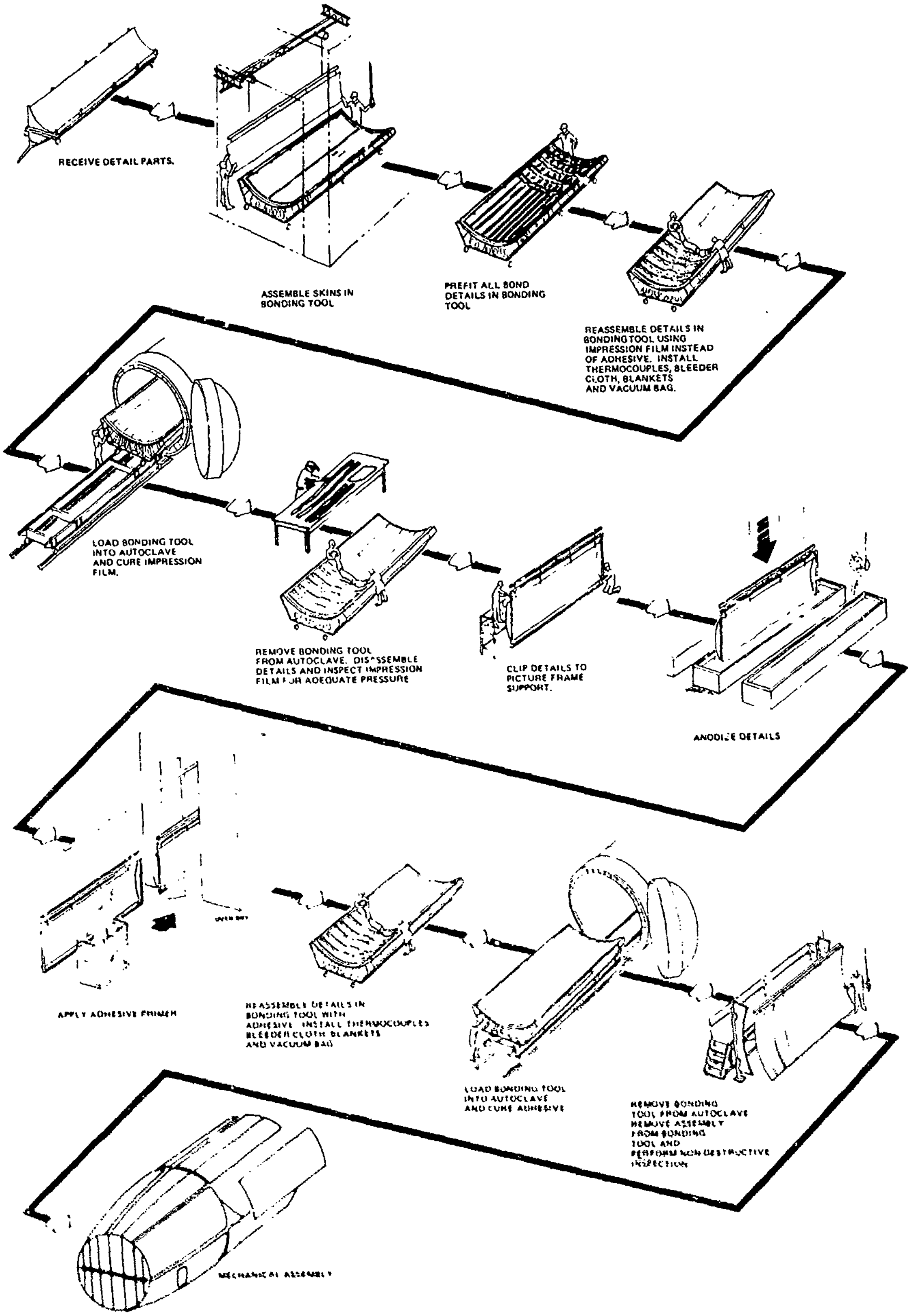


FIGURE 5 TYPICAL MANUFACTURING SEQUENCE

The adhesive bond is the primary load transfer path and by design must be stronger than the metal parts being joined. In order to realize this high strength in the bonded joint, an important factor is the glue line thickness and the uniformity in thickness along the length of the bonded joint. Thin metal bonded parts can be pushed together with autoclave pressure and a uniform bond line obtained. Depending on the cross section of the parts, there is a tendency to get pinch-off or thinning of the glue line at the extremities of the joint. Any thinning relative to the total joint is not desirable because high load transfer will take place at these places. The bonding tool plays an important part in the final bond line thickness. For example, the female tool is an excellent fixture for bonding doublers to thin skins and longerons to the skin/doubler combinations. It is difficult to bond stiff members, such as frame shear tees, to the above skin assembly which is backed by the stiff female tool unless the contour of the shear tee exactly matches the shape of the bonding tool and allowance made for the buildup of material between the shear tee and bonding tool. The larger the circumferential dimension, the more difficult the matchup is. The male tool is an excellent fixture for supporting stiff members such as the frame shear tee. In this instance, the flexible skin assembly is then pushed down against the frame shear tee and will easily conform to the exact shape of the shear tee and will give a uniform bond line. See Figure 6.2 for a general comparison of the tools.

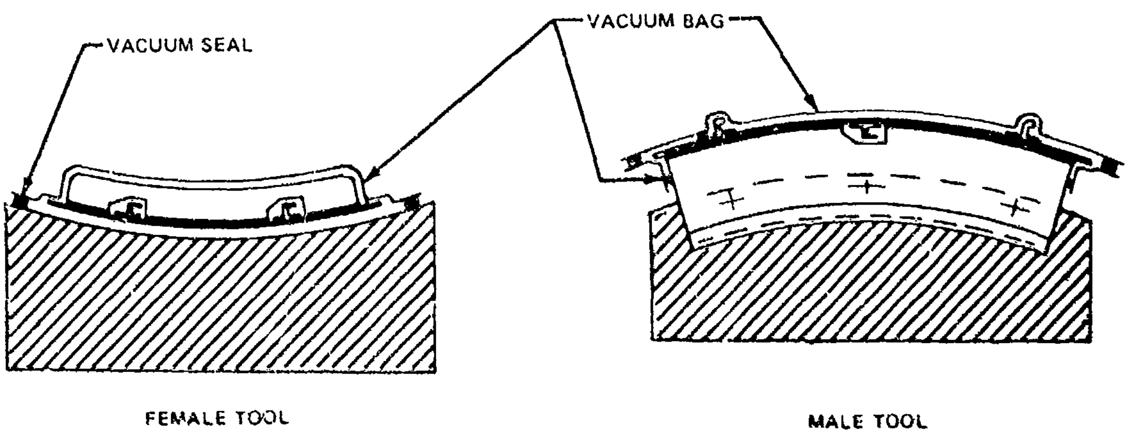


FIGURE 6 BOND TOOL CONCEPTS

A major problem associated with the use of any bonding tool is the application of the vacuum bag. The purpose of this cover is to exclude the autoclave pressure from the faying surfaces of the bond. The cover, or bagging material, is usually transparent and is drawn tightly across the part to be bonded by a vacuum. And hence its name, vacuum bag. As the complexity and number of the parts being bonded increase, so do the contortions that the bag takes as it is drawn down tightly around all of the details. All sharp corners must be protected so the bag will not be easily torn. If the bag tears during the cure process, it is possible to lose the positive pressure on the parts being bonded and thereby get no bonding, and the subsequent loss of all the details. Figure 7 is an example of how the sharp edges of the parts being bonded are buried under a mass of quarter inch hollow aluminum spheres. In addition, the figure shows the other materials used within the bonding tool. In this instance, the bagging material is draped over the top of both the spheres and the bleeder fabric then sealed to the bonding tool around the edges. The aluminum spheres will distribute the autoclave pressure to the parts being bonded but have been known to deform at the points of contact and become locked together so that they may not transmit the pressure uniformly. Hence the use of a silicone rubber on top of parts to be bonded for a more uniform pressure distribution.

The male tool supported the parts on contour boards secured to a picture frame tool. These parts were envelope bagged. This means that a vacuum bag was applied to each surface of the parts as they were secured to the bonding tool and the two bags were sealed together around the periphery. In this case, the great depth of the supporting tool caused difficulties in getting a good installation of the vacuum bag. Also, the part had to be turned over after one side was started so that the bag could be installed on the opposite side. Again, a time-consuming and difficult operation. For flat parts, it is possible to envelope bag with no tool involved and suspend the bagged details in the autoclave for the pressure/temperature cure cycle. Figure 8 shows a panel being fitted into the male tool. Figure 9 shows this part with the bagging material on one side with the part turned over awaiting the second vacuum bag.

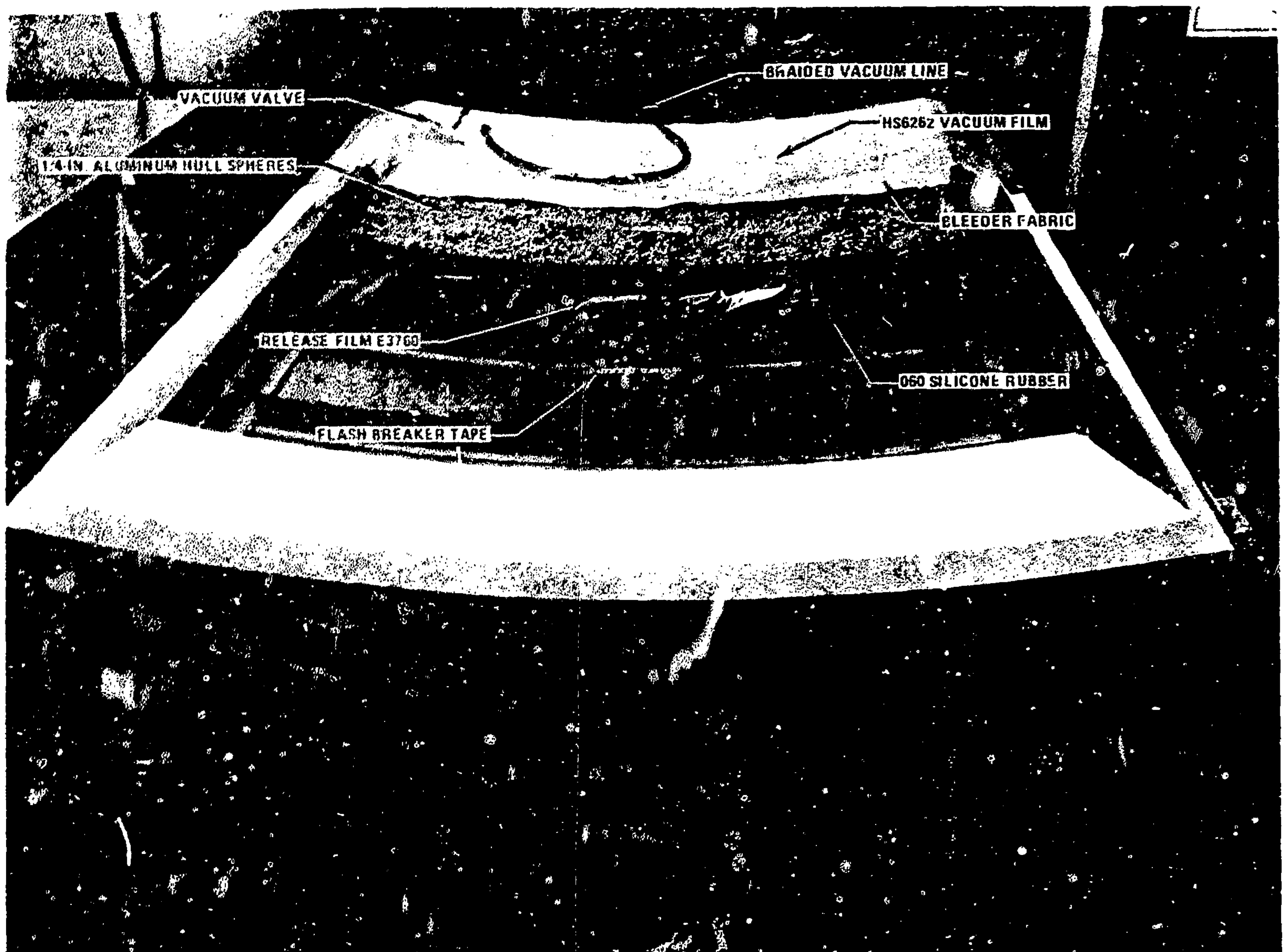


FIGURE 7 - FEMALE BONDING TOOL WITH ALUMINUM SPHERES USED TO DISTRIBUTE PRESSURE

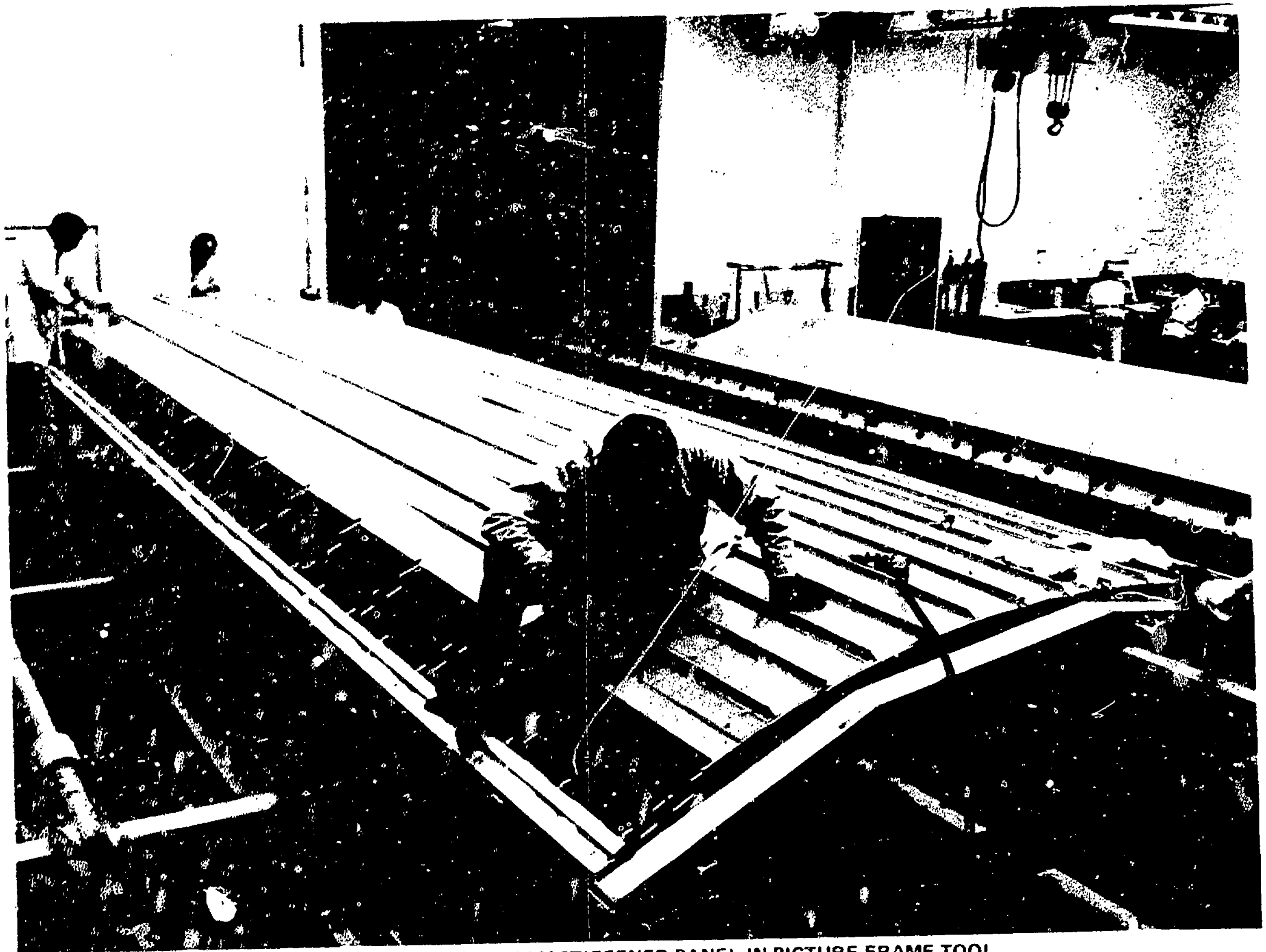


FIGURE 8 . EXTERNALLY STIFFENED PANEL IN PICTURE-FRAME TOOL

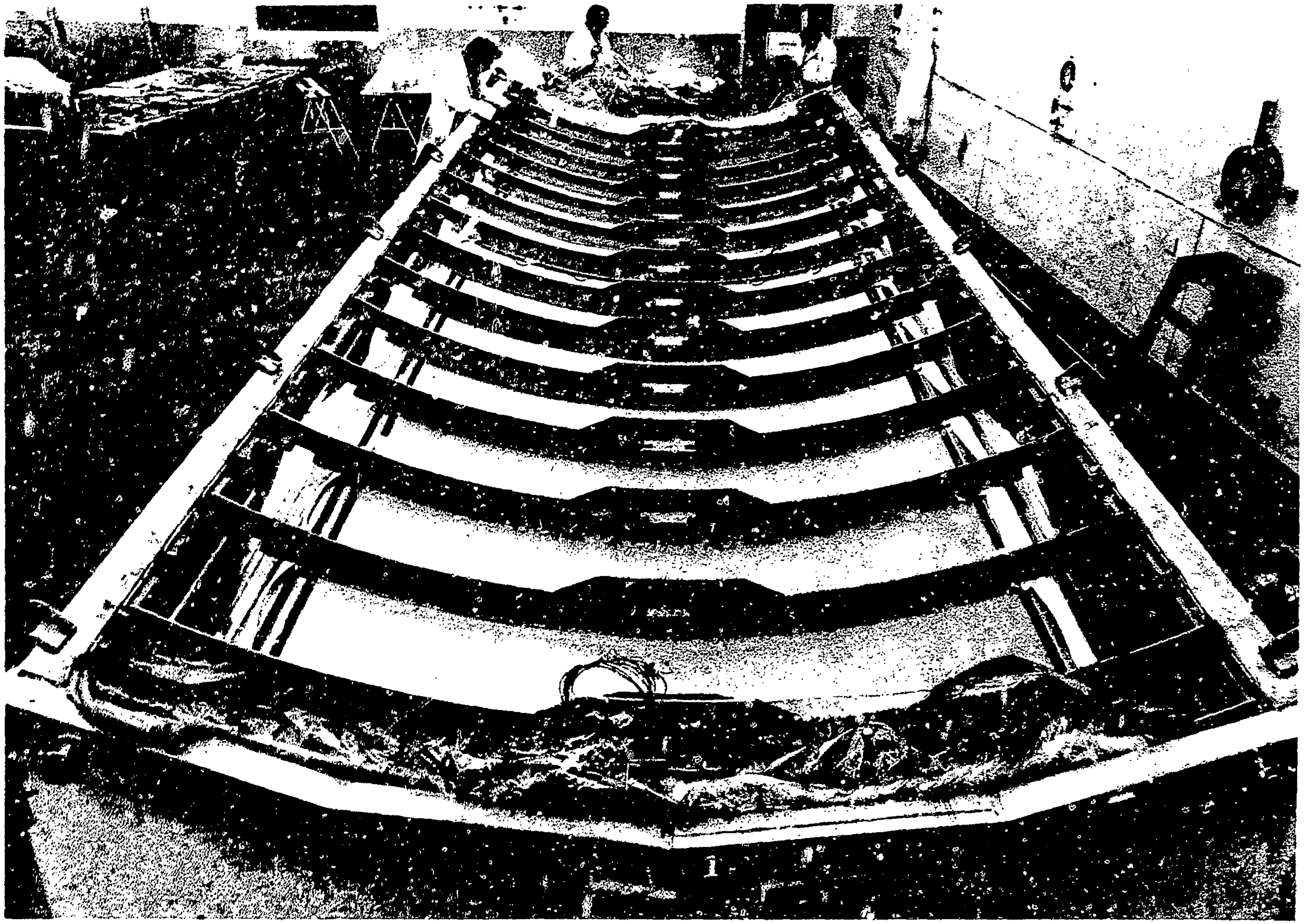


FIGURE 6.5. MALE TOOL WITH EXTERNAL BAGGING INSTALLED

During the manufacture of the PABST FSDC panels, lots of bleeder cloth was placed between the aluminum parts and the vacuum bag. It was felt that this cloth was essential for allowing the air to be withdrawn from under the bag. Subsequent development work has shown that this cloth can be eliminated, and with controlled vacuum application a smooth bag can be pulled down against the parts being bonded. It was possible to use too much bleeder cloth so that bridging occurred in the corners of the details giving an unequal pressure to the skin and subsequent inward deformation of the skin between the stiff frame shear tees. By bagging against the details without the bleeder cloth, this problem can be eliminated.

6.2 Panel Size

Manufacturing constraints have a strong impact on the size of bonded panels or assemblies. The autoclave size, diameter and length, establishes the size of the assemblies that can be bonded. Furthermore, the location of the stiffeners with respect to the edges of each panel can permit considerable manufacturing savings. If each bonded panel is sufficiently close to the same size as one or more, and the stiffeners are located on a common grid or spacing, even their cross-sectional areas are not identical. One tool can be used to bond several panels, and cost savings realized.

A considerable savings in cost can be accomplished by increasing utilization of an autoclave by sizing the panels to maximize the total skin area bonded per cycle. Instead of filling the autoclave with a single panel of the biggest possible size, far more of the structure can be bonded at the same time in nesting several slightly smaller panels and supporting them in a common supporting tool. Each panel is individually envelope bagged so that in the event of a bag failure, only one panel would be lost.

6.3 Tolerances

It was determined, during the course of the PABST program, that the drawing tolerances on sheet metal and extruded details for bonded parts can be the same as that currently required for making parts to be used on a mechanical fastened assembly. Figure 10 summarizes comparative tolerances of bonded details versus mechanical fastened parts.

SHEET MATERIAL	DIMENSION AND/OR CONFORMITY TO CONTOUR	BOND DETAIL	MECHANICALLY FASTENED	COMPARISON
	108 R CONSTANT SECTION	±3.00 R WITHIN 0.12 IN. OF THE LOFT LINE	WITHIN 0.032 IN. OF THE LOFT LINE	EQUIVALENT
	649.96 R NON-CONSTANT SECTION	0.047 IN. OF THE LOFT LINE	WITHIN 0.032 IN. OF THE LOFT LINE	EQUIVALENT
EXTRUDED DETAILS	108.00 R SHEAR TEE	WITHIN 0.12 IN. OF LOFT LINE	WITHIN 0.032 IN. OF THE LOFT LINE	LESS STRINGENT FOR BOND DETAILS
	CONCAVITY 1.750	0.009 IN./IN FLATNESS TRANSVERSE	0.004 IN./IN FLATNESS TRANSVERSE	LESS STRINGENT FOR BOND DETAILS
	CONVEXITY 1.750	0.006 IN./IN FLATNESS TRANSVERSE	0.004 IN./IN FLATNESS TRANSVERSE	LESS STRINGENT FOR BOND DETAILS
	DEGREES TWIST	1/2 DEG MAXIMUM TWIST	APPROXIMATELY 1/2 DEG TWIST	EQUIVALENT
	BEVEL 0.050	±1 DEG	±1 DEG	EQUIVALENT
	TOTAL LENGTH STRAIGHTNESS	APPROXIMATELY 0.0125 IN./FT	0.0125 IN./FT	EQUIVALENT
	WAVINESS	APPROXIMATELY 1/32 IN. AND SHALL FAIR IN NOT LESS THAN 10 IN. OR 0.003 IN./IN. OF LENGTH	1/32 IN. AND SHALL FAIR IN NOT LESS THAN 10 IN. OR 0.003 IN./IN. OF LENGTH	EQUIVALENT

FIGURE 10 COMPARATIVE TOLERANCES

6.4 Metal Fit Check

Because each of the PABST bonded assemblies were different, it was necessary to make a metal fit check on all detail parts to provide initial insurance that the bonding operation would be successful. All the detail parts were assembled and then laid up in the bonding tool to make certain that the autoclave could press the parts together. If finger pressure (5 lbs) could push the details together at the location on the tool, the parts would make a good bond assembly. For a production run, special metal fit check tools are built in order that the parts may be easily inspected after they are all assembled in the fit check tool. After all the parts are assembled in their proper position, a limited number of holes are drilled in the details. Later these holes will be used to hold the details together with undersized bolts prior to the application of the autoclave pressure. The fasteners are removed after parts are bonded together.

6.5 Verifilm

This operation allows the parts to be sent through the normal autoclave pressure and temperature cycle with a non-sticking simulated adhesive placed in each faying surface. After the cure cycle the verifilm is removed and the resulting simulated glue line thickness measured. All bonded panels on the PABST program went through the verifilm measurement. Where the thickness of the verifilm exceeded the acceptable value determined to represent the final glue line, an extra layer of adhesive was added to the joint. All the part details were identified by number so that in the subsequent bonding cycle all parts can be reassembled in the same order.

6.6 Wedge Crack Verification

Adhesive layup cannot proceed beyond the primer cure step until wedge-crack specimens are accepted by Process Engineering.

Two adhesive-bonded wedge-crack plates (6 by 6 by 0.125 inch, 7075-T6 bare aluminum alloy) are bonded and then cut into 1 by 6 inch specimens, resulting in a total of five specimens after sawcutting and polishing one edge.

A one inch wide, 0.125 inch thick wedge is driven into one end of the specimen to a depth of one inch, partially separating the specimens along the bond line. The crack length is scribed on the polished edge. The specimens containing the wedge are exposed to 140°F, and 95 to 100 percent relative humidity for one hour. Any crack length growth is indicated by scribing. The wedge is then driven deeper into the bond line, completely separating the bonded specimen, and the adhesive surface is visually inspected to determine the failure mode. The failure mode must be completely cohesive for the anodized and primed details to proceed with the adhesive layup step. Crack length growth for a cohesive failure does not normally exceed 0.060 inches.

6.7 Racking

An adjustable erector-set rack is used to accommodate various sizes and shapes of assemblies to be processed. At the beginning of the PABST program, aluminum wire was used to hold the parts to the rack. Later in the program, titanium wire clips and springs were used to hold the parts, and to ensure a proper electrical continuity during the phosphoric acid anodizing process, and reduce the amount of time it takes for racking. Titanium clips are provided in a variety of C-shapes, with V-shaped bends at each end, and the coil springs are provided in three sizes. The titanium clips and springs show a time saving of approximately 65 percent in racking, and 75 percent in unracking after anodizing relative to the use of aluminum wire. With this system, proper tension can be applied to the details and maintained throughout the processing. All details on the rack are checked before processing to ensure proper orientation, as the shape of some details could cause puddling. (for example: If a shear tee is positioned horizontally with the standing leg up, a puddle will form along the radius of the standing leg at the flange.) During the final check of details on the processing rack, any looseness detected is corrected by merely adding another clip or spring as required.

A bookkeeping system was devised for keeping track of wedge crack plate numbers, detail part numbers, and phosphoric acid anodize load number. One wedge crack plate was attached to each string of detail parts going through an anodize operation. This assured that any anomoly occurring in a string due to electric current flow would be reflected in the wedge crack piece in that string and subsequently show up on the wedge crack test. A number was etched on each wedge crack piece and that number was documented along with the detail part numbers as to its string number, exact position on the string and anodize load number. This data was entered on the fabrication outline (F.O.) and in the event the wedge crack specimen from a given string and anodize load failed in test, the F.O. was used to trace the detail parts numbers on that string and anodize load for corrective action.

6.8 Phosphoric Acid Anodizing

Phosphoric acid anodizing was introduced into Douglas' processing system in 1975. One 3 by 40 by 12 foot deep tank was converted to a phosphoric anodized unit by lining it with a six percent antimony lead alloy. The deionized water system was modified to allow a spray (instead of submersion) system for rinsing after phosphoric acid anodize.

In early usage, the phosphoric solution became contaminated. The problem was found to be an aerobic mold similar to the "mother" found in vinegar. A filtration system was installed to filter aerobic mold and other contaminates (such as lead, airborne objects, and insects). The new filtering system was designed to handle 10,000 gph to ensure maximum filtration.

During normal production use of the phosphoric acid anodizing system, a non-uniform appearance occasionally appeared (called a "halo" effect) on some details. It was requested by Process Engineering that one such detail, a large doubler, be sent to the laboratory for evaluation. A section was cut from the doubler, and wedge-crack coupons were cut from this section and bonded. The rest of the details were reprocessed and showed no discoloration.

The wedge crack specimens made from the discolorated doubler met the cohesive failure mode requirement of the specification. Process Engineering assessed the condition to be the result of a monor heat-treat problem, with alloy variation in the base material and not cause for rejection.

A primary consideration in the phosphoric acid anodize system is that detail parts cannot be handled after processing. Other systems (such as chromic acid anodize surface treatment, sulfuric acid, sodium dichromate etch [FPL], etc.) present much the same kind of problem. It was quickly discovered during wedge-crack testing that handling the details after processing damaged the surface treatment and caused adhesive failure in wedge-crack tests. The surface treatment is easily contaminated because of its porous surface.

Process Engineering's special order was issued with a warning in the process specification, reading:

Warning:

From the beginning of the anodize process cycle to the completion of the primer oven dry cycle, the processed details shall not be contacted or touched in any manner; i.e., by a gloved hand, bare hand, metal or other detail or tool, kraft paper, or other contacting device, material or method. Details that have changed position during the anodize processing cycle due to forces exerted by the processing solutions or sprays, and that cause subsequent problems in draining or the adequate application of the adhesive primer, may be repositioned by touching and moving the contact wire or by light contact with clean, white cotton-gloved hands, on the cut edge, non-bonding surface of the detail only. DO NOT TOUCH BONDING SURFACES UNDER ANY CIRCUMSTANCES.

6.9 Quality Assurance

After phosphoric acid anodizing is completed, the rack of details is moved on the monorail from the processing area to the adhesive primer area. From the time the details leave the post anodize drying oven, the adhesive primer application must be accomplished within two hours. Within that two-hour time span the anodic surface treatment is inspected by Quality Assurance, using a mercury vapor fluorescent lamp to illuminate the anodized surface, and a photographic polarizing filter lens as shown in Figure 11. The lens is held at an angle between 0 and 10 degrees to the detail part surface. The phosphoric acid anodize will display interference colors. During inspection of the surface the lens is rotated 90 degrees. An acceptable anodic coating is indicated by observing a change to the complementary color. (For example: from purple to a yellow green.) The reason for rotating the lens is that some pale shades of yellow or green are so close to white that without a color change they could be considered "no color" and would falsely indicate no anodic coating.

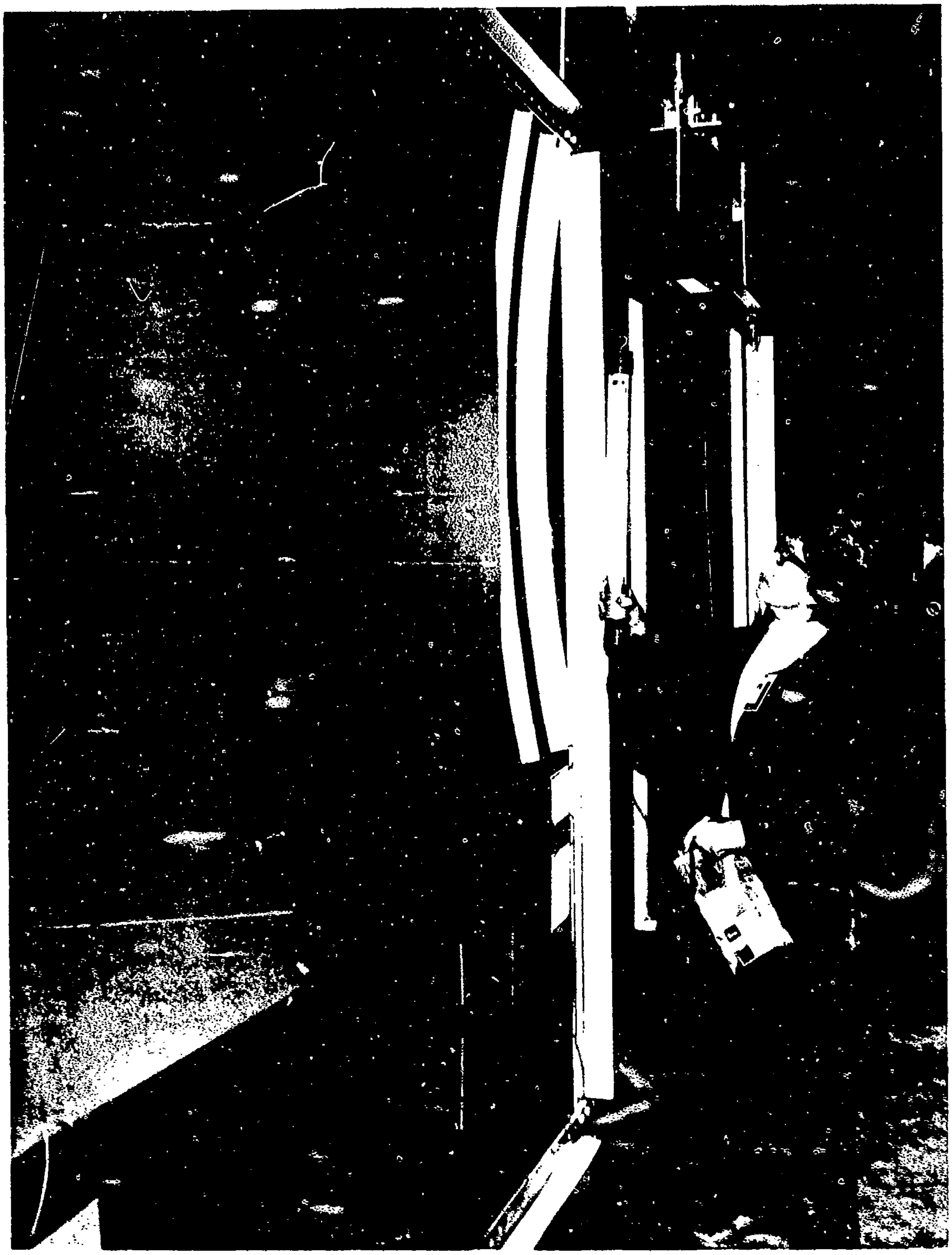


FIGURE 11. BOND DETAILS RACKED FOR PROCESSING

Aluminum details anodized under the same conditions may show different interference colors because of variations in alloy composition and metallurgical condition. Purple, yellow, blue and green hues are most frequently observed. All colors are acceptable, but a color change is mandatory. Anodized details will display a color change on all their surfaces. Any abrupt color differences in local areas (with the exception of electrical contact points) form the background color, such as those caused by finger prints or abrasions, are unacceptable. Details not acceptable to Quality Assurance are rejected and may be recycled with approval of Processing Engineering.

6.10 Adhesive Primer Application

After the anodic surface treatment is accepted by Quality Assurance, the racked details are moved into the adhesive primer booth. The adhesive primer application is a very important step. The adhesive primer plays a very important role in protecting the anodized surface against handling requirement during layup and in establishing a compatible surface for the FM-73 adhesive.

During early phases of the program, minor problems were encountered in applying BR-127 primer. Sometimes, after being applied and dried in the oven, the primer could be wiped off with methly ethly ketone (MEK) solvent. To correct this problem, personnel were provided several hours of training by instructors from Process Engineering and the Manufacturing Training departments. As the problem still arises occasionally, it is apparent that the ability to apply adhesive primer is skill that must be developed on an individual basis.

During the first two phases of the program, a DeVilbiss spray gun (JGA, fluid tip and needle, No. 36 air-cap) was used with 45 $\pm$ 5 psi nozzle pressure. The primer was thoroughly agitated to ensure that all solids were in suspension and that the primer was continuously agitated during spraying.

Primer is applied to all details in a wet uniform cross coat (1 $\overset{3}{\underset{4}{\longleftrightarrow}}$ 2) which, after the cure cycle, is 0.0001 to 0.0003 inch thick.

An alternate primer method investigated employed the recirculating pump system. This system, later adapted for use in Phase III, uses a DeVilbiss QBV-604 spray gun (DeVilbiss JGA-502 with a JGA 402-6 needle, AV-601-G fluid tip and an AV-1239-758 air cap.)

Adhesive primer is sometimes applied too thick. Consequently, several non-destructive techniques have been evaluated for measuring primer film thickness. Twelve test specimens were phosphoric-acid anodized and primer film thicknesses ranging from 0.06 mils to 0.4 mils were applied. Film thickness was calculated from the change in weight before and after adhesive primer application. The results were compared using the isometer, an eddy current measuring instrument, the C-Gage, a capacitance measuring instrument, and the betascope, making use of beta-ray back scattering. Using calculated values as the basis for comparison, it was concluded that for the film thickness range considered, the isometer and the C-Gage are somewhat more accurate for thin films than the betascope. The isometer and the C-Gage are more or less comparable in accuracy, but personnel from the Process Engineering coatings laboratory feel that neither is acceptable enough for film thicknesses less than 0.2 mils (for production inspection for PABST panels).

All primer thicknesses are verified by Process Engineering using the wedge crack pieces which were primed details in the same load as the detail parts. Specimens that do not meet the required thickness are rejected. Rejected details are processed through a stripping operation using hot chromic acid, then reprocessed through the phosphoric acid anodize system.

6.10.1 Curing of Adhesive Primer

The primed details were air dryed in the primer booth for a minimum of 30 minutes prior to oven drying at 235°F to 265°F for 50 to 70 minutes. After oven drying, the rack of details is moved to a staging area where each details was inspected visually by Quality Assurance. After inspection, the wedge-crack coupons removed from the anodic rack (handled with white cotton gloves) and wrapped in brown kraft paper. Wedge-crack coupons were then hand-carried to the Process Engineering laboratory for bonding. The wedge cracks were bonded, using a layer of adhesive applied to the faying surface and cured within 96 hours after adhesive primer application.

6.11 Fasteners in Bonded Structure

In cases where it is impractical to fabricate a complete bonded assembly, two or more bonded sub-assemblies may be permanently joined together by multi-stage adhesive bonding, mechanical fastening, or a combination of both types. The use of mechanical fasteners in primary adhesively bonded structure is generally limited to longitudinal and circumferential joints of major components and panels. Fasteners are also used when adding secondary details to a previously bonded structure.

Special consideration by Manufacturing in drilling and reaming for attachment hole preparation is a critical operation in the fabrication of bonded structure. It is essential that proper controls and procedures be utilized by Manufacturing to maintain hole tolerance and prevent damage to or delamination of the bondline. Excessive heating and chattering due to dulled drill bits must be avoided to ensure bond joint integrity.

The designer should remember that hole preparation and attachment installation costs constitute a major portion of Manufacturing cost for conventional construction as well as for bonded structure. Screws, bolts, and rivets (squeeze, pull or driven types) are generally used in conventional construction and may effectively be installed in bonded structure with proper hole preparation, and the following recommendations are listed below for assuring a high quality attachment installation.

6.11.1 Wet Installation

Fasteners in conventional construction are installed wet per MIL-F-7179 to prevent pressure leaks, fuel leaks, and metal corrosion problems. The same sealants are used in bonded structure. These sealants will remain flexible.

Since moisture is generally always present in the atmosphere, all fasteners, countersunk or not, must be installed wet in bonded structure to prevent moisture from entering and attacking the bondline causing corrosion or delamination. Wet installation of attachments provides this protection and eliminates the problem of environmental degradation at the bond surface.

6.11.2 Countersinking

In bonded structure, unlike conventional construction, there is no minimum sheet metal thickness recommended for countersinking. The countersink cavity may extend through the top sheet (leaving a knife edge) and bondline into the doubler, as shown in Figure 12 . This is acceptable since shear is carried in the bondline rather than in the fasteners. The only limitation is that the fastener head must cover the bondline which would otherwise be unprotected from environmental degradation.

6.12 Faying Surface Sealing of Bonded Assemblies

A faying surface sealant is required when mechanically joining bonded assemblies. Not only does the sealant eliminate pressure leaks, its initial intent, but provides protection against moisture that might otherwise enter the faying surface area through the edges of the assemblies being joined,preventing corrosion.

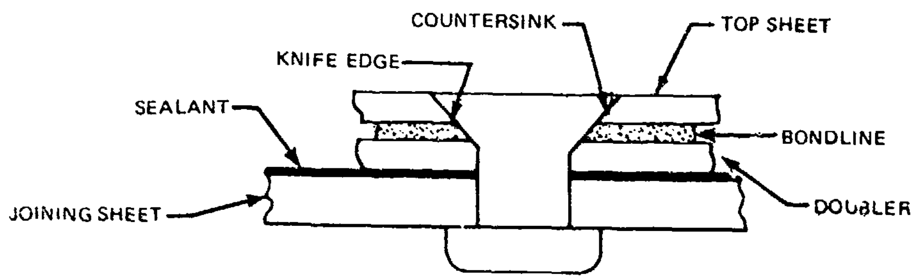


FIGURE 12. COUNTERSUNK FASTENER IN BONDED STRUCTURE

SECTION VII
INSPECTION CONSIDERATIONS

After a bonded panel has been fabricated it may require inspection to ensure that the bondlines are of sufficient quality to resist the expected loads. The first responsibility of the designer in this area is to design the structure so that the bondlines are inspectable by the type of nondestructive testing (NDT) method that will be used. Then, using the information on the real location of load transfer in adhesive bonded joints, see Section 10.3.2.1, it is possible to prepare a rational plan for minimum but adequate inspection and acceptance/rejection criteria for bonded structures and the detail parts.

There is a wide variety of NDT methods available for detecting voids, porosity, and debonds. Methods most applicable to bonded laminates are:

Ultrasonic Inspection

This method uses pulsed ultrasound at 1 to 10 MHz. Inspection may be performed by either the contact or the immersion method. The method may be automated to produce plan view recordings (C-scans) employing the pulse-echo, through tranmission, or reflector plate techniques (Figure 13). These techniques are useful for producing C-scan recordings of small test specimens and flat laminates, but special equipment is required for large panels, and contour followers for contoured parts. The ultrasonic method suffers from destructive wave interference at certain adhesive and metal thicknesses. By inspecting with several frequencies it will be possible to examine all of the part.

Ultrasonic Resonance Impedance

There are several instruments available that operate on the principle of resonance impedance. An ultrasonic transducer or probe (see Figure 14) is manually coupled to the part using a liquid couplant. The instruments are calibrated to respond to a shift in frequency and signal amplitude between a good and "no-bond" standard. For bonds less than the diameter of the transducer, the frequency shift and signal amplitude will vary between the bond/no-bond response. These instruments operate in the kilohertz range.

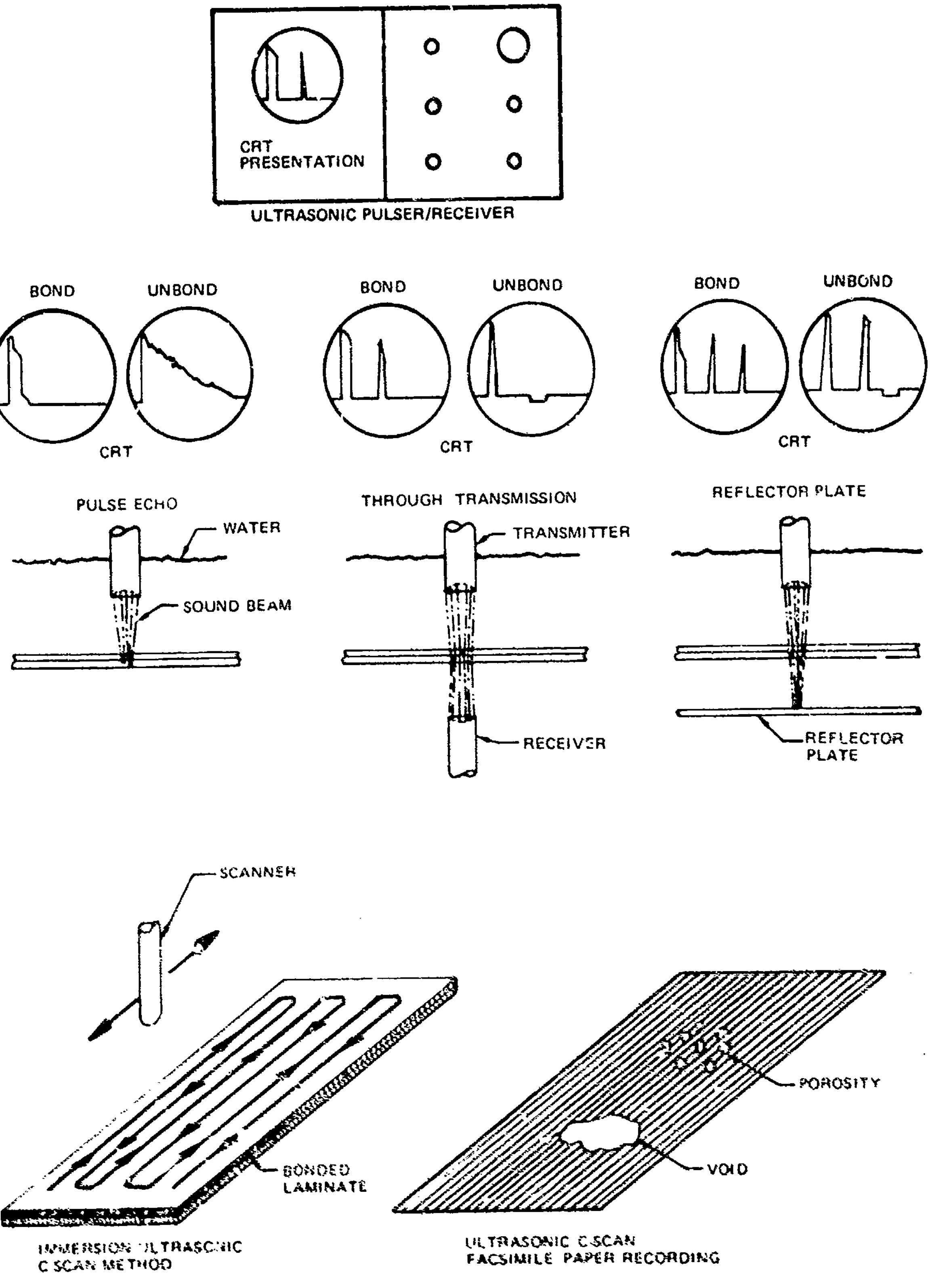


FIGURE 13 IMMERSION ULTRASONIC TESTING TECHNIQUES

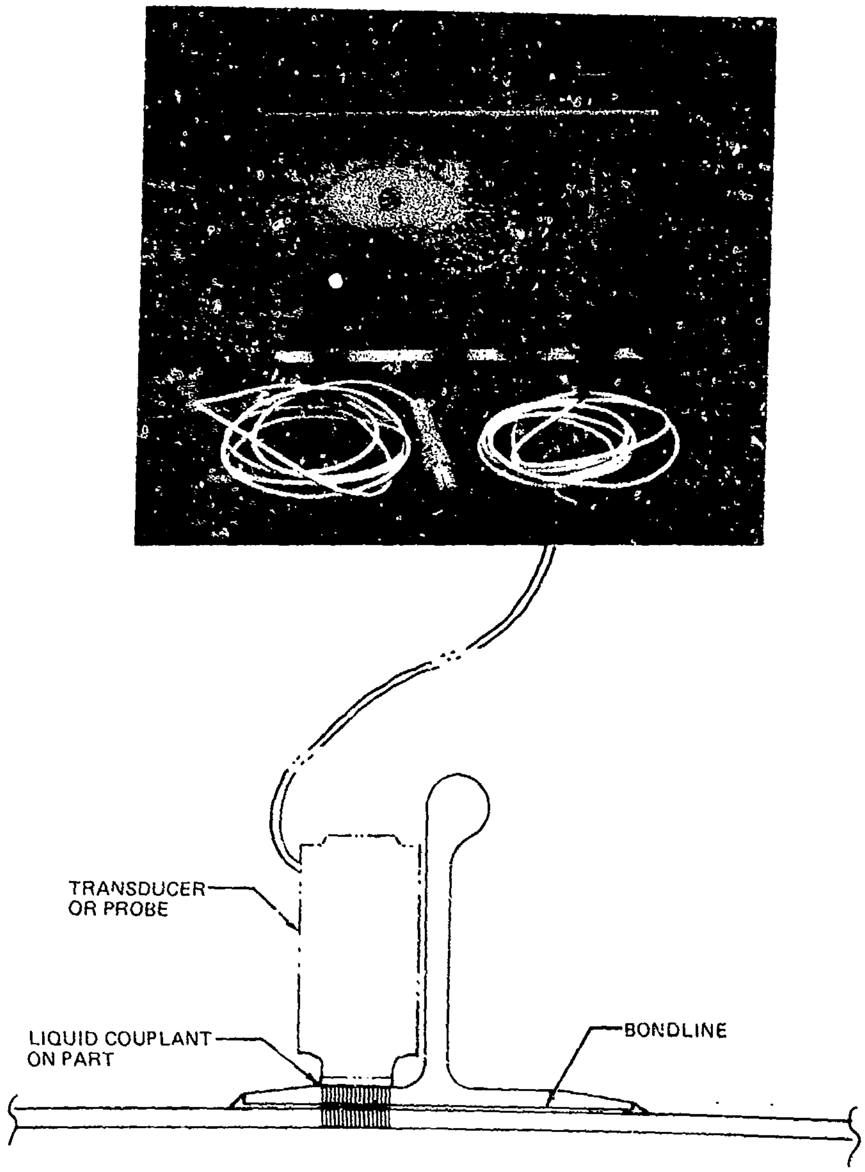


FIGURE 14 ULTRASONIC RESONANCE IMPEDANCE BOND TESTER

The parts must be manually scanned and flaw areas marked on the surface of the part.

Radiography

Some adhesives are x-ray opaque, enabling voids and porosity to be detected in metal-to-metal areas. This is extremely useful, especially for complex geometry joints which are difficult to inspect ultrasonically. If the adhesive being used is not x-ray opaque, neutron radiograph may be used with comparable success. The hydrogen atoms in the adhesive absorb neutrons making the adhesive opaque. Radiography, however, will not detect lack of bond areas where the adhesive is present but not bonded to one or both adherends.

Special Methods

NDT methods, which are not production state-of-the-art, include holographic interferometry, acoustical holography, infrared, and acoustic emission.

The Ultrasonic Resonance Impedance method is the most versatile state-of-the-art method but it does require some special considerations when designing the structure. The critical load carrying areas of a splice or stiffener is near the edge as explained previously. For this reason, it is important to be able to inspect this area thoroughly. For single bondlines, radiography is adequate but for two or more bondlines ultrasonic methods may be required to determine which bondline contains the void or porous adhesive. For three or more bondlines, the second layer bondline is difficult to inspect using ultrasonics, although it is fairly reliable when detecting voids or unbonds greater than 1/4 inch wide. Chamfers at the edge of the splice or stiffener must be at least 3/4 of the probe diameter in width and vertical clearances must be greater than the probe height (see Figure 15). In addition, adhesive flash must be kept off the surface to be inspected either by protecting the surface during bonding or by removing the flash after bonding.

Based on the expected loads, the designer can determine which areas will require inspection and what the acceptance and rejection criteria will be for each joint to be inspected. For instance, the bondline between a large

'A' DIMENSION IS 3/4 OF THE PROBE DIAMETER, MINIMUM.

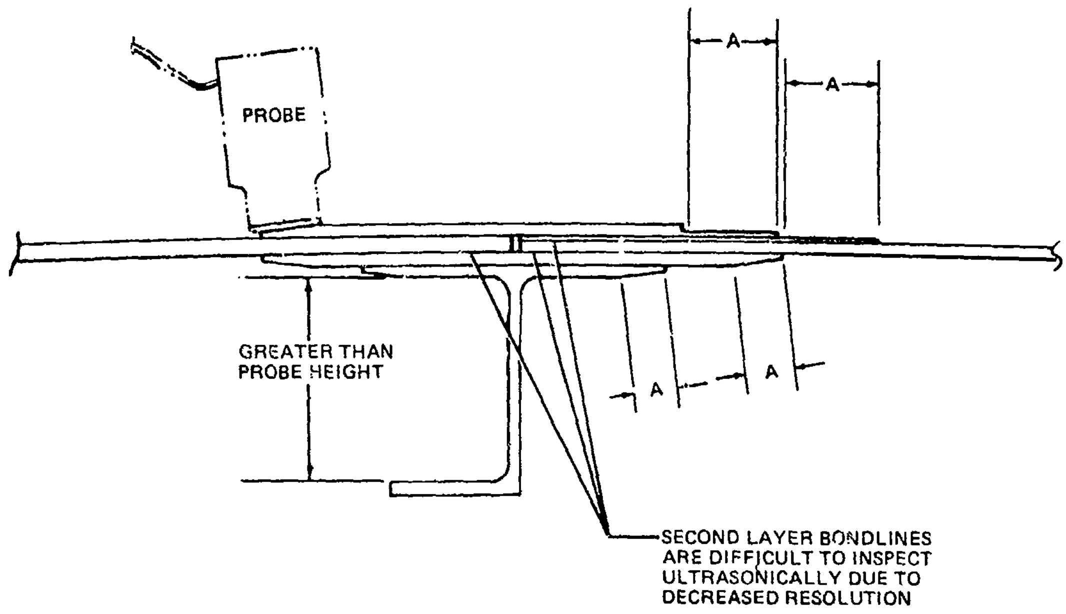
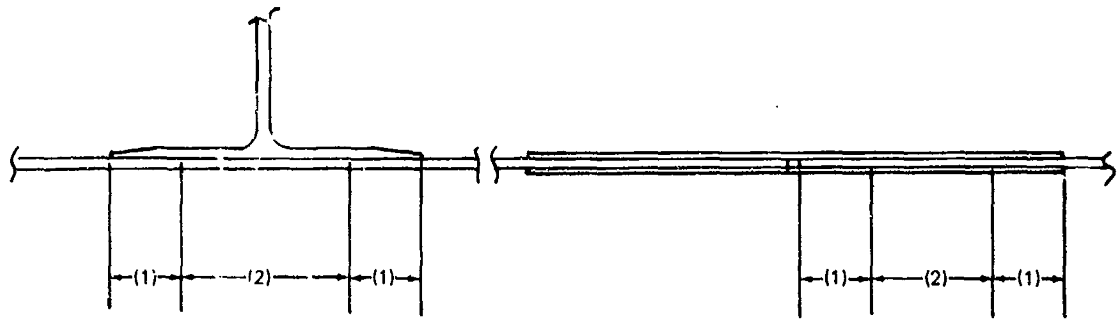


FIGURE 15 DESIGN CONSIDERATIONS FOR ULTRASONIC INSPECTION

doubler and skin will most likely be more forgiving of voids and porosity than the bondline between a stiffener and skin. Furthermore, for a stiffener, voids and porosity at the middle of the stiffener will have little effect on the capability of the structure while at the edge of the stiffener a void or porosity could cause the bondline to be critical (see Figure 16). A similar situation exists in a bonded lap splice. Most of the load is transferred through the bondline at the edges not at the middle. Acceptance and rejection criteria should be formulated accordingly.

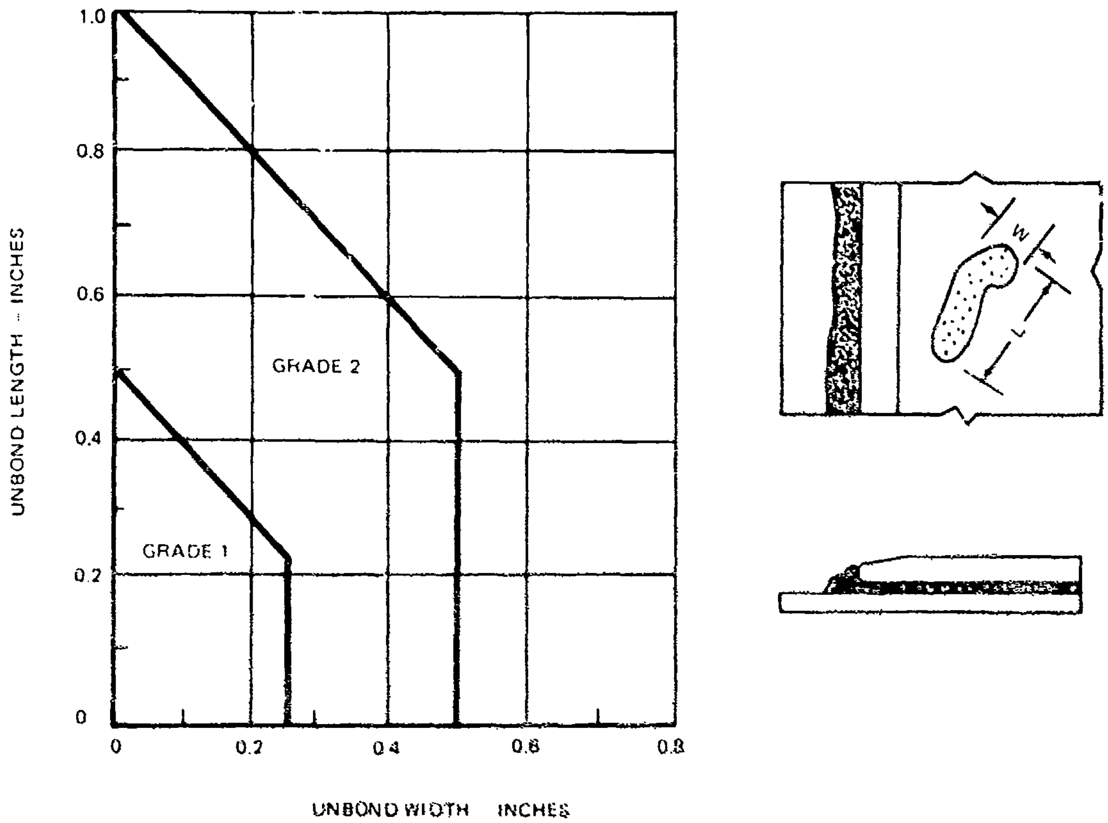
Acceptance and rejection criteria should relate to void or debond maximum area and on the frequency of occurrence. Figure 17 illustrates a typical acceptance criteria for voids or debonds. Similar criteria should be developed for porosity in the bondline. Again, these criteria are developed by the designer based on analysis of the structure under the expected loads.



(1) CRITICAL AREA – ACCEPTANCE REJECTION CRITERIA SHOULD BE RIGID.

(2) NONCRITICAL AREA – ACCEPTANCE REJECTION CRITERIA MAY BE RELAXED.

FIGURE 16 CRITICAL AREA OF BONDLINE



NOTES:

(1) "W" IS THE MAXIMUM PROJECTED UNBOND WIDTH MEASURED ACROSS THE VOID IN THE "NARROW" DIRECTION AS SHOWN IN THE SKETCH. "L" IS THE PROJECTED UNBOND LENGTH MEASURED PERPENDICULAR TO "W."

(2) THE MINIMUM VOID SEPARATION DISTANCE IS 1 INCH.

FIGURE 17. TYPICAL ACCEPTANCE GRADES FOR VOIDS OR UNBONDS

SECTION VIII

REPAIRABILITY

The life cycle cost for a structure may be significantly affected by the number and type of repairs required when the structure is damaged. Repair of bonded structure is in many cases more difficult than repair of conventional riveted structure; however, if care is taken in the initial design, the repairs may be as simple as adding fasteners to the damaged area. Bonded stiffeners should have a wide enough base to permit the addition of fasteners should the stiffener ever become disbonded (See Figure 18). The edge distance for the fasteners should be the minimum acceptable for a riveted design while the fastener diameter for a countersunk fastener may be constrained by the skin gage so that the countersink does not knife-edge the skin. Fasteners may also be added to splice members or doublers where there are flaws large enough to propagate (See Figure 19). Should fasteners be an unacceptable method of repair secondary bonding techniques may be required. Figure 20 shows how a repair might be performed on a bonded laminate that has been damaged at the edge. The damaged area is routed away as shown. Excess adhesive may be removed with carefully selected solvents and/or scraped off after applying cold; e.g., dry ice which makes the adhesive brittle or heat; e.g., heat lamps or heat gun which softens the adhesive. In cases where the adhesive is fairly uniform and thin and not contaminated, it may be left on the surface since it provides an excellent base for the new adhesive. The surfaces are then processed and the assembly is pressure bagged and put in the autoclave. As discussed previously, during this second (initial fabrication was first) bonding cycle details previously bonded should be protected from being loaded by the pressure bag in a way which might fail the bond. At the elevated adhesive curing temperature, these bonds are extremely weak. For assemblies which are not able to be returned to the autoclave other means of applying heat and pressure must be found. Heat lamps and heat blankets are available commercially and local area pressure applying bladders will work well for most light structure.

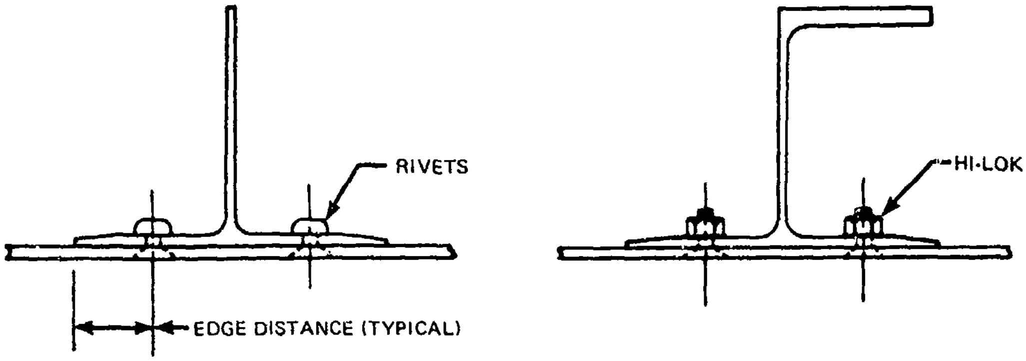


FIGURE 18. TYPICAL FASTENER INSTALLATION FOR REWORK OF DEBONDED STIFFENERS

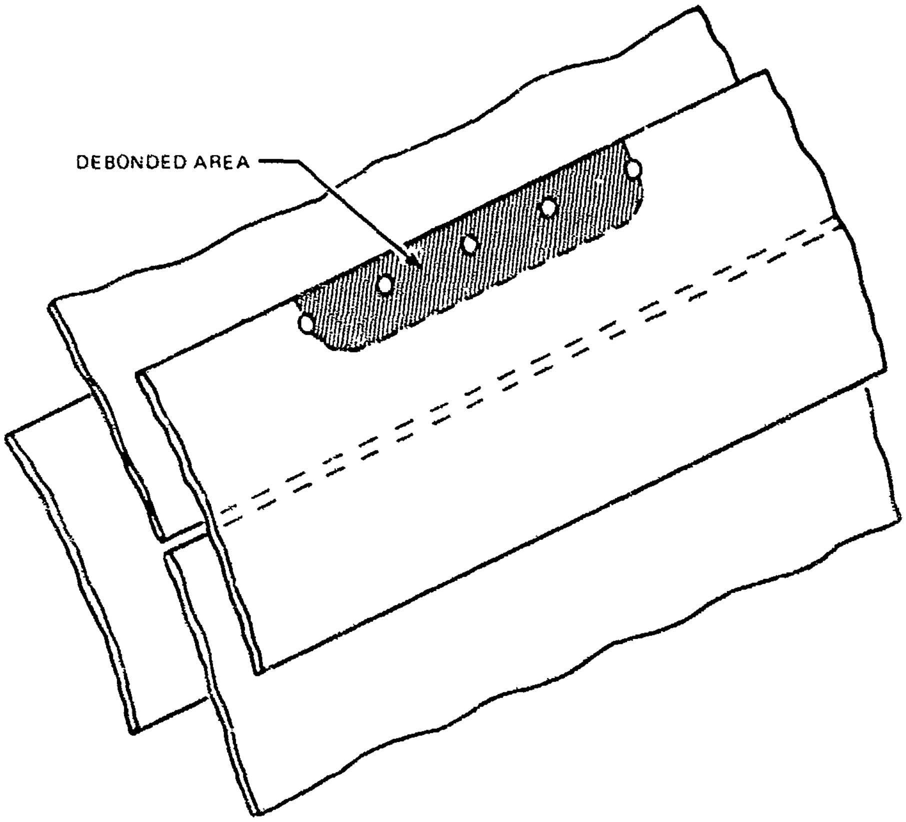


FIGURE 19. TYPICAL FASTENER INSTALLATION FOR REWORK OF DEBONDED SPLICES AND/OR DOUBLERS

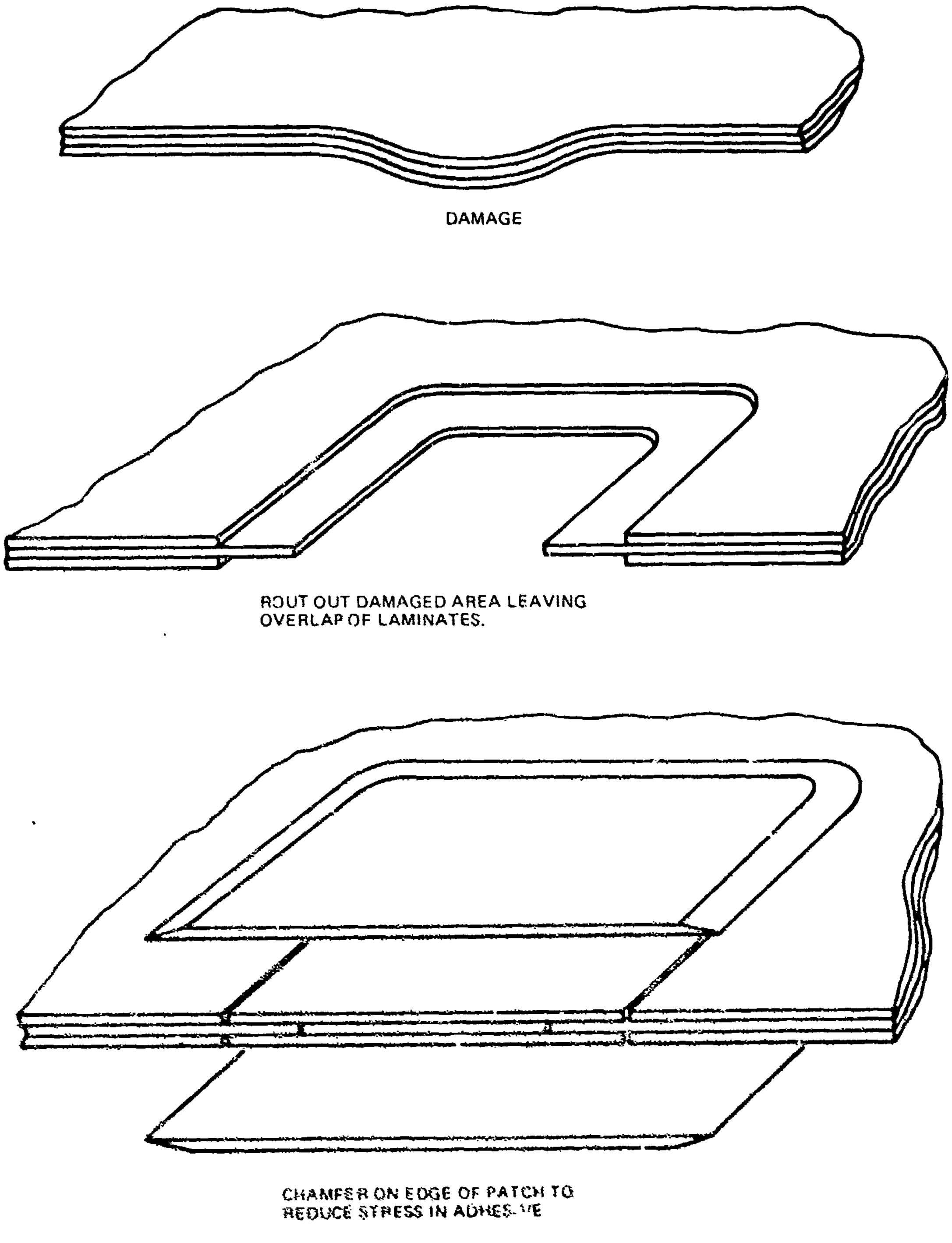


FIGURE 20. TYPICAL REPAIR OF DAMAGED BONDED LAMINATE

The repair of a large variety of typical damage to bonded structure is covered in Reference 10 which deals with the different types of damage, materials to be used, preparation of details, application of the adhesive, curing of the adhesive system and inspection and quality assurance.

SECTION IX
COST AND WEIGHT CONSIDERATIONS

Throughout all phases of design, fabrication and test, emphasis was placed on translating the adherent advantages of adhesive bonding into significant reductions in the acquisition and maintenance cost of primary structure. The goals established by the program with consideration to using bonding on the AMST class of fuselage were to achieve a 20 percent acquisition cost reduction concurrent with a 15 percent weight reduction for the participating structure. It was also desired to show a similar cost saving in the maintenance area.

9.1 Cost Analysis

The use of adhesive bonding to join primary structural components of an airframe offers some unique economic advantages in terms of reductions. The PABST approach for the manufacture of selected aircraft structure provides the basis to achieve economic gains in both the acquisition category and operating/support category of the life cycle cost of an aircraft system. Cost benefits have been recognized for some time with bonded airframe secondary structure. Initial estimates and projections of cost savings with PABST have been confirmed through detailed accounting and cost tracking of the manufacture of all of the bonded panels.

Acquisition savings can readily be estimated and projections developed. However, operating/support cost savings cannot be estimated and projected with any degree of confidence. This is due to the lack of a proper quantitative corrosion data base in the USAF from which proper cost analyses can be accomplished. But, the potential cannot be ignored since the technology has demonstrated corrosion improvements. Therefore, equal consideration is given to the operating/support cost. However, from a practical standpoint, program front-end costs or near term expenditures are more heavily weighted and emphasized than are the long term expenditures in the downstream years (contain more uncertainty). While life cycle costs are important, concrete front-end cost savings have a greater influence in the budgeting process.

Estimates of recurring manufacturing costs have been made for a section of the YC-15 fuselage structure for both conventional and PABST design and construction. These recurring costs were based on 300 production shipsets and they exclude non-recurring and all non-manufacturing receiving costs (e.g.; sustaining engineering). It is assumed that an adequate bonding facility exists and production bonding tools available for comparison with available tools for locating and drilling rivet holes. A drivematic rivet machine is assumed to be used. The comparison of costs between the conventional design/construction and the PABST design/construction was accomplished only after the baseline conventional design was modified to conform closely to the PABST design. It was reasoned that the advancements in design practices could be achieved with conventional construction and advantage could be taken of reduced part count, wider spacings and larger skins. This resulted in a conservative approach with respect to cost advantages on behalf of the PABST concept over the conventional approach. The net effect of this action is to provide a potential cost savings lower than that which could be realistically achieved with a purely conventional design (i.e.; no modifications to the baseline design). This action also provided a range of the cost savings. However, there is no conflict with the statements in the earlier paragraphs of this section wherein references are made to the conventional design and the reduction in parts - realistically this was and is the case.

Cost analyses, cost projections and cost estimates were accomplished over a three to four-year period. It was an integrated effort of multiple disciplines which were coordinated with the on-going design and construction of the PABST component and the tracking of the costs involved with this component. In order to assess the propriety of the estimates and projections of the PABST component to a production environment, a comparison was made between the actuals and the estimates associated with a first unit production. To establish the adequacy of and confidence in the estimating process, a comparison was also made between the conventional baseline approach and barrel sections of the DC-10 in terms of hours per pound. From the analyses conducted it was established that (1) the baseline estimate had good correlation with similar actuals for DC-10 production; (2) the PABST Full Scale Demonstration Component

T_1 actuals; and (3) the results overall indicate projections have a good degree of confidence. See Figure 21.

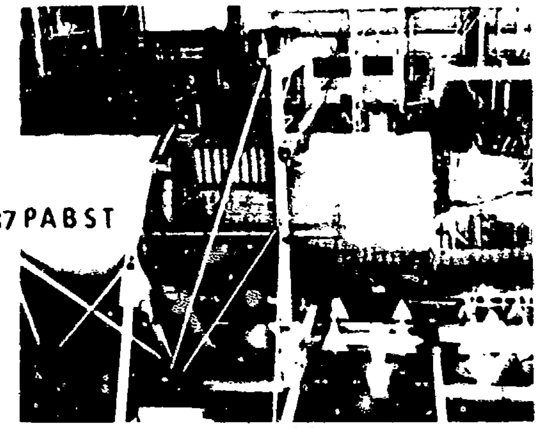
The economics of a PABST application in which the conservative approach was taken shows a minimum potential cost savings of 20 percent. Taking full advantage of the PABST concept and applying that concept over a broader application offers savings in recurring manufacturing on the order of 30 percent all of which still represents the conservative projection, given a modified baseline design. For the PABST concept, rivets are reduced 67 percent and the overall reduction in tooling by type and quantity is 29 percent. While these savings are feasible, it has been established that close coordination and control are required at each of the steps from advance design to the manufacture in order to optimize design and tooling to yield the greatest cost benefits. Additional savings in life cycle costs are to be realized in the lower maintenance actions due to the reduced incidence of fatigue cracks which reflects also in effectivness improvements due to improved turn-around times and aircraft availability.

9.2 Weight Savings

Previous paper studies have indicated that bonding of primary structure, which eliminates the rivets, and subsequent "hole-out" in the load carrying members, should produce a 15 percent savings in weight. In the early analysis of the PABST structure it became apparent that removing the rivets did not change the crack growth rate of the basic materials when the requirements of MIL-A-83444 were complied with. Therefore, early in the program it was felt that no weight saving could be obtained. In the cycle testing of the FSDC crack growth, behavior was observed. It was found that the bonded structure arrangement of the PABST FSDC would arrest a fatigue crack in the skin by completely stopping it in the bond line of the frame shear tees to fuselage skin joint. This was demonstrated with five different cracks in .060 and .050 thickness material. This new crack arrest feature is not predictable by available crack growth analysis. It appears now, at the conclusion of the program, that a weight saving of 15 percent to 20 percent in the fuselage skin material, is possible. As the designer looks to different designs, appropriate tests need to be conducted to justify the crack arresting feature of the structural arrangement being developed. In the case of the PABST fuselage structure, tests have determined that crack growth in unaxially loaded specimens will not demonstrate the same crack arrest feature that was seen in the biaxially loaded pressurized fuselage of the FSDC structure.

DC-10 BARREL SECTIONS
CUM AVG 237
2.57 HR/LB

PABST BASELINE
CUM AVG 237
2.76 HR/LB



PABST COMPONENT
CUM AVG 237
2.11 HR/LB

PABST T_1 COMPARISON	
FAB + METAL BOND	
ESTIMATE	ACTUAL
2.88 HR/LB	5.36 HR/LB
FAB + ASSY + BOND	
ESTIMATE	ACTUAL
8.75 HR/LB	7.59 HR/LB

FIGURE 21. BASELINE CREDIBILITY

SECTION X

LOADS, LOAD TRANSFER, AND FAILURE MODES

Load transfer between two elements of a structure bonded together is accomplished by minute differential displacements between the elements. Because the structural elements are elastic, non-rigid, and distort under load, the displacements across the bond lines are not uniform. Consequently, the adhesive stresses and strains vary over the bonded interfaces. Under most load conditions, a relatively small area of adhesive adjacent to the perimeter of the bonded area, or the end of a structural element, is the only part of the bond that deforms or is loaded significantly. Bonded joints cannot be designed on the basis of uniformly stressed adhesive over the entire bond area. The majority of the adhesive in bonded structures must be relatively unstressed because of the distortion of the structural elements being bonded together. Even with damaged structure, the high adhesive stresses and strains are confined to the immediate vicinity of the damaged area. In addition, a determinable area of lightly stressed adhesive is needed between the narrow effective load transfer zones, in order to develop fully the characteristic elastic troughs to restrict creep deformation of bonded joints. Further areas of bond may be needed: (1) for damage tolerance, (2) to minimize bending in the structural elements or (3) peel stresses in the adhesive caused by load path eccentricities. Otherwise, excessive bond areas are unnecessary and possibly dangerous because they suggest strength reserves which cannot be developed.

The following subsections describe the unique ways in which bonded structure and the bond itself responds to loads, load transfer and to cracking and damage failure modes.

10.1 Tension.

10.1.1 Stiffened Panels. - No Bonded Splices

Figure 22 shows a bonded stiffened panel subjected to a uniformly distributed longitudinal tension load. As the load is increased, the aluminum parts strain uniformly, then yield and finally, when the gross area stress

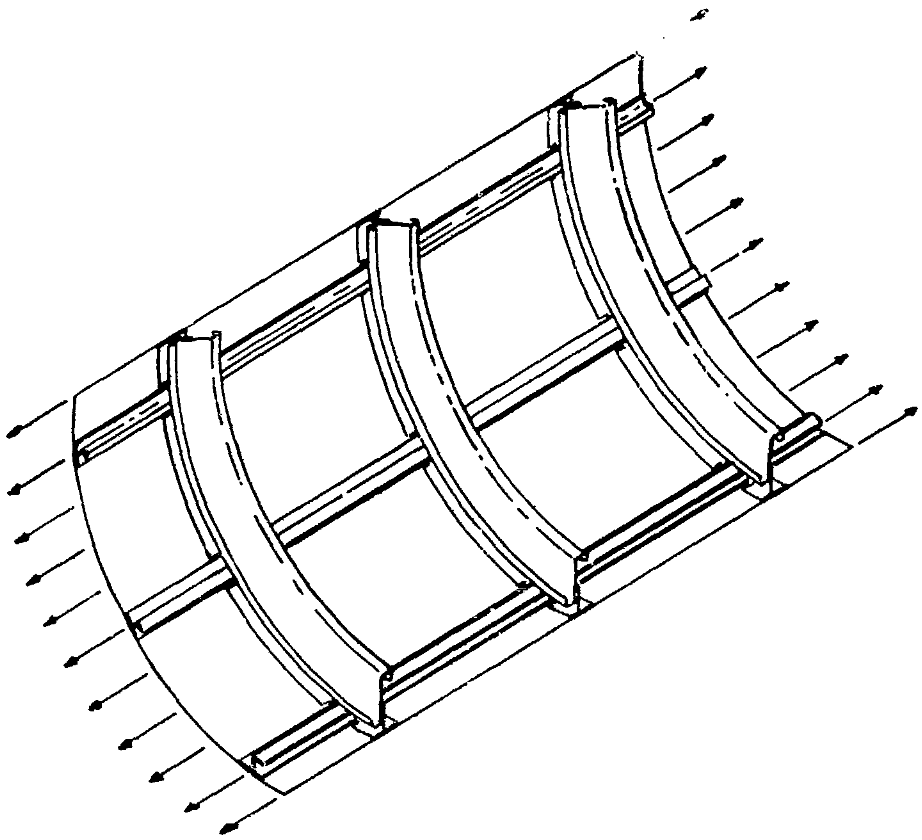
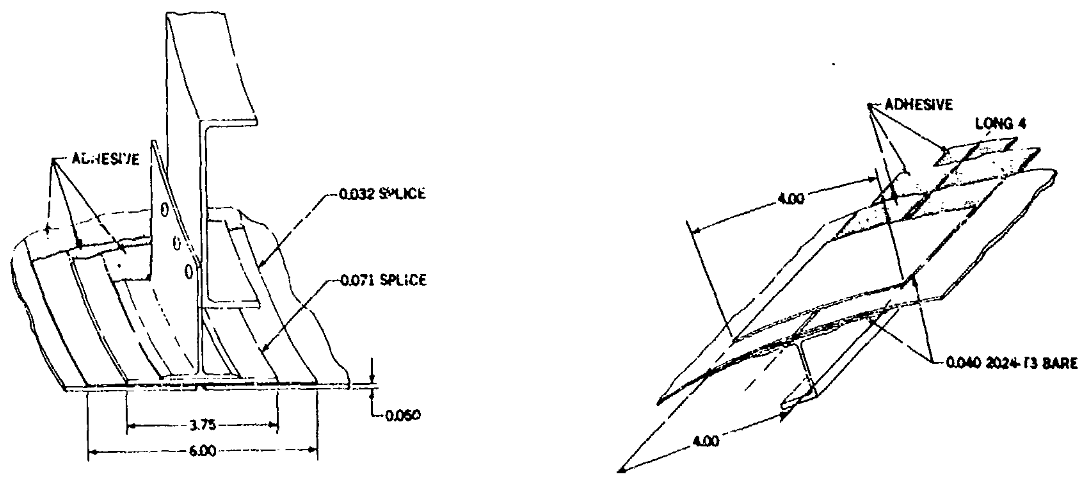


FIGURE 22. CURVED, STIFFENED, BONDED PANEL IN TENSION

reaches the ultimate stress of the metal, the skin and longeron will fail. Provided that the skin and longeron are continuous and of constant area, they strain equally and there is no load transfer across the bondline away from the load introduction at the edges. Although microscopic cracks will appear in the adhesive after the metal yields, there will be no disbonds prior to ultimate failure of the metal parts. The adhesive will not be critical for such uniform continuous structure. However, if tensile loads are applied to a panel containing structural discontinuities such as stiffener runouts and changes in skin gage, there will be high adhesive shear stresses associated with the structural discontinuities as the tensile load is transferred from one member to another. This load transfer is discussed in Section 10.3.2.

10.1.2 Stiffened Panel - With Bonded Splices

The fuselage structure that was designed, built and successfully tested to four lifetimes contained bonded longitudinal splices and circumferential splices of the fuselage skin. In spite of the fact that some of the splices contained voids, there was no expansion of the voids during the cycle testing and no metal cracks initiated as a result of the bonded skin splices. It should be noted that the skin stress levels for the fuselage, which is pressurized, were fairly low because of the damage tolerance requirements of MIL-A-83444. Pressurized fuselages operate at low tensile stresses; that is, limit loads are considerably less than the yield stress of the metal. If bonded splices were to be used in other structures (wing or empennage) care must be exercised to make sure that the sustained load stress for ultimate conditions are below yield stress of the metal at the splice. Small test specimens, under uniaxial load, have shown that yielding the adherend in the bonded joint will cause progressive failure of the adhesive. See Figure 23 . This has not been demonstrated on large panels nor has it been seen where biaxial stresses are present. Until more experience is gained at various stress levels and stress fields with bonded splices in all types of designs, development tests should be made. Figure 24 shows the skin splice designs used on the PABST F



TRANSVERSE BONDED SPLICE (STA 703 ONLY)

LONGITUDINAL BONDED SPLICE

FIGURE 24 BONDED SPLICES

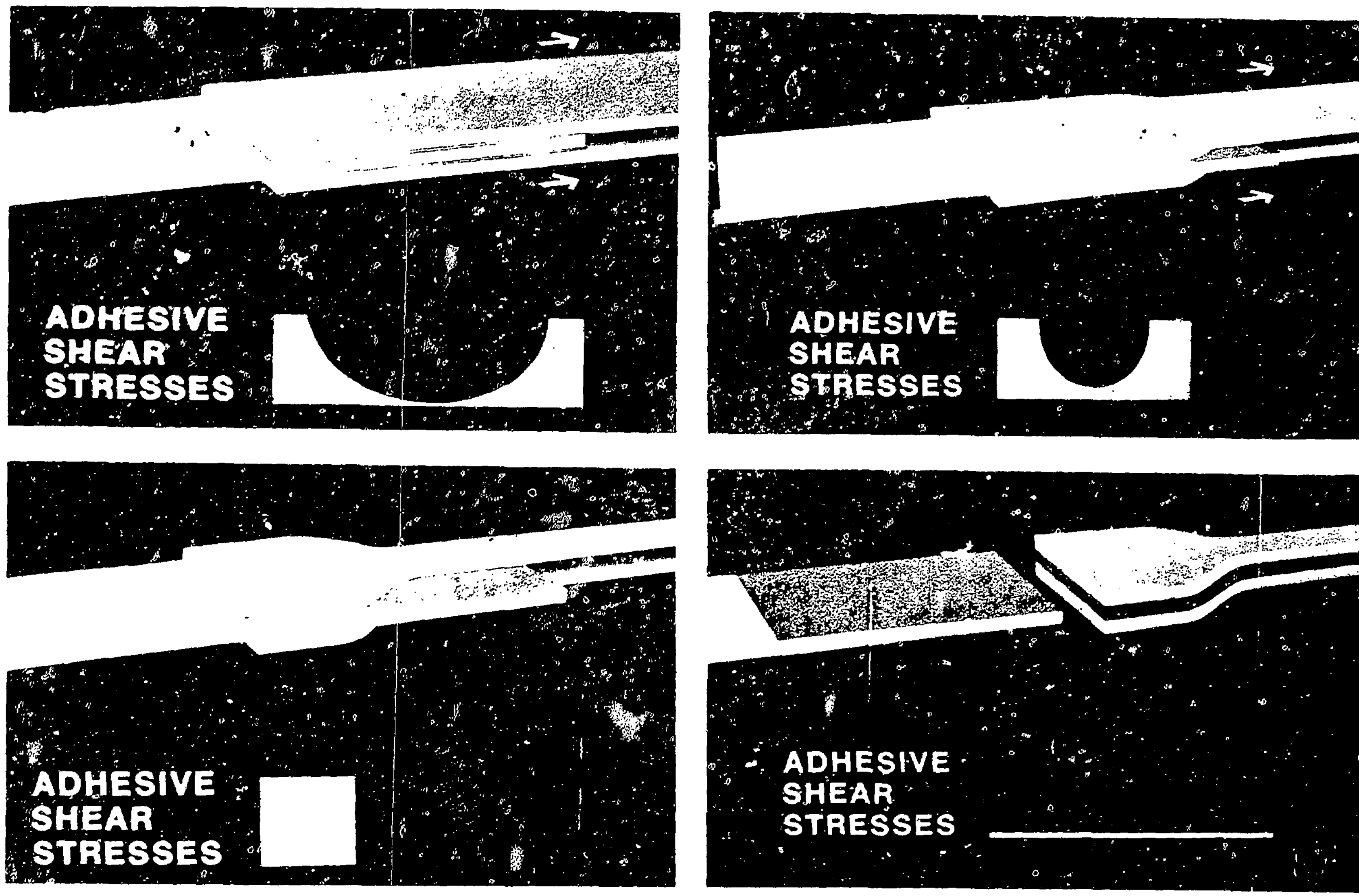


FIGURE 23 EFFECT OF YIELDING OF METAL

10.1.3 Frame Tee-To-Skin Bond Peel Loads

In a pressurized shell, the adhesive between the frame tee and the skin is loaded primarily in tension. Since the skin and the base of the frame tee deflect, shear stresses are also introduced, but they are negligible. The tension load on the adhesive is identical to the peel loads caused by shear wrinkles as discussed in Section 10.3.1. A schematic representative of these peel stresses is shown in Figure 25. Note that the peel stresses (strictly normal tensile stresses) are not distributed uniformly over the bonded area. Not only that, but nearly half the bonded area is trying to push the other half apart. This gives rise to tremendous stress amplification since the net stress is only about four percent of either the total tensile or compressive load for the illustrative case shown. The failure would start at the edges, rather than under the web of the stiffener, if the applied load be great enough. The peel stress distribution in the adhesive as shown in Figure 26 is derived from classical analysis method for a beam on an elastic foundation. Increasing the ratio of the base width to its thickness or chamfering the base will lower the peak stresses at the edge of the tee. In spite of the high calculated stresses, a specimen similar to that shown in Figure 26, was tested to failure at 1,600 pounds (at -50°F) and the calculated limit load was only 390 pounds.

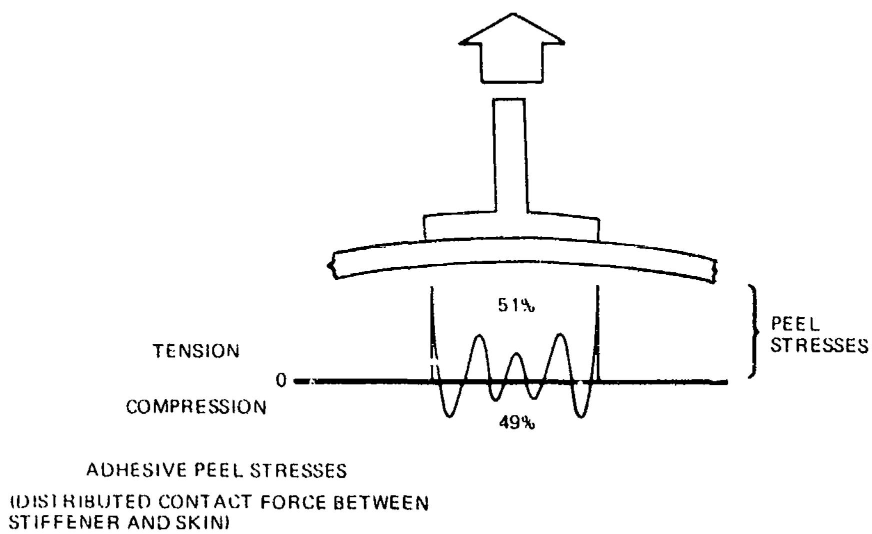


FIGURE 25 PEEL STRESSES IN ADHESIVE BETWEEN FRAME TEE AND SKIN

10.2 Compression

If the stiffened panel shown in Figure 22 is subjected to a low uniformly distributed longitudinal-compression load, the skin and longeron agains strain equally and there is no load transferred through the adhesive. However, as the load is increased, the skin stops taking more load and begins to wrinkle between the longerons. As the wrinkle approaches the base of the longeron, it produces peel stresses in the adhesive between the longeron and skin, see Figure 26. The amount of buckling required to fail the adhesive cannot be computed directly or accurately; however, the base of the longeron can be designed so that the longeron will buckle prior to a disbond. In general, the peel stresses are lower when the longeron base is thinner. Therefore, an effective way to further reduce these peel stresses is to taper the base of the longeron as shown in Figure 27. The same care should be taken when designing frame tees where frame bending will cause compression in the skin.

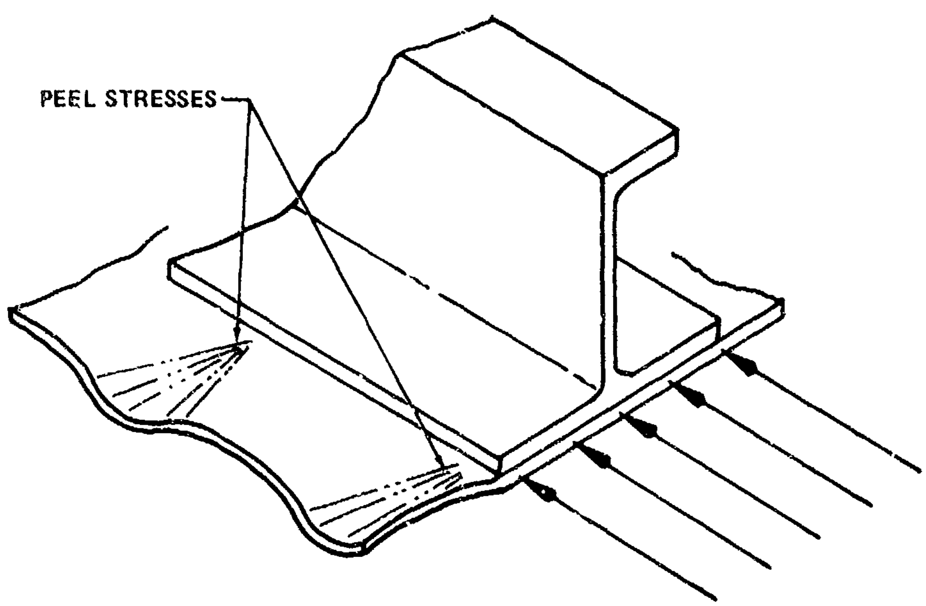


FIGURE 26. SKIN BUCKLING IN COMPRESSION

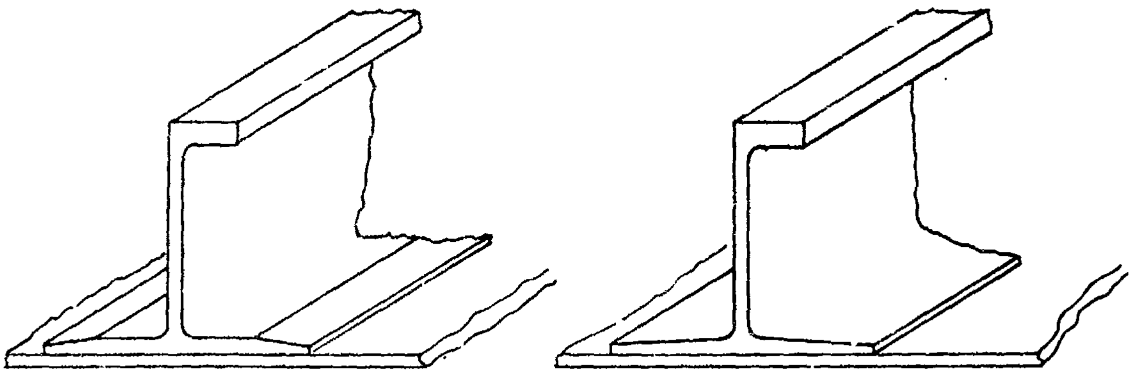


FIGURE 27. TAPERED BASES FOR STIFFENERS

10.3 Shear

10.3.1 Stiffened Panels

Shear loads on a panel also create skin wrinkles. Usually these skin wrinkles will form diagonally across a panel as shown in Figure 28. This causes the same peel stresses discussed in Section 10.1.3 at both the frame tees and the longerons, and they may be resisted in the same way. As the base of the longeron or frame tee is made more flexible, it provides a greater amount of peel resistance when the shear wrinkle attempts to cross it; hence, it will resist higher shear stresses.

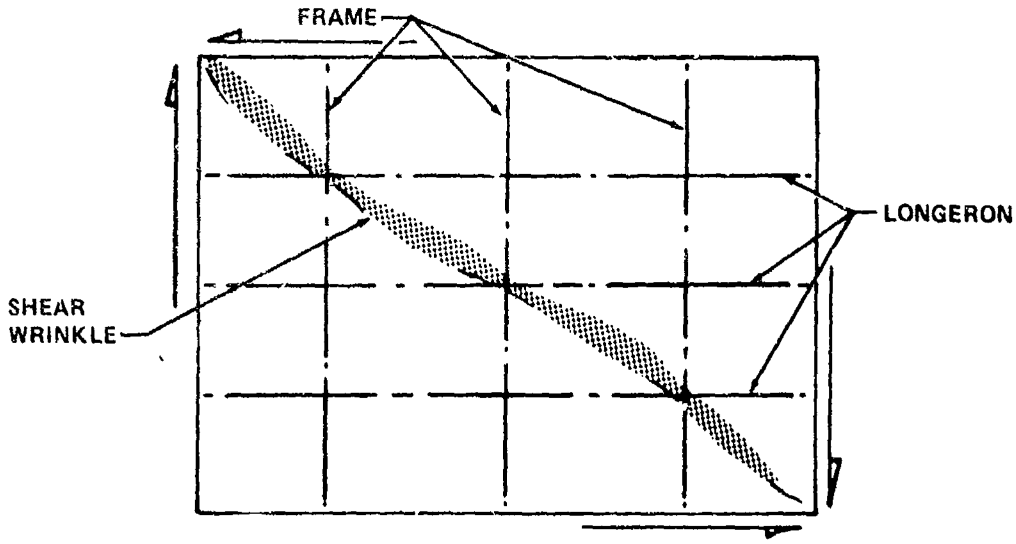


FIGURE 28. PANEL SHEAR WRINKLES

In general, a panel design using many light stiffeners will be more effective in resisting shear than one using a few heavy ones. Breaking up the panel into smaller bays decreases the amplitude of the shear wrinkle thereby decreasing the peel stresses in the adhesive.

10.3.2 Bonded Joints

There are a variety of configurations in which adhesively bonded joints are used, as illustrated in Figure 3 . However, despite that variety, there are certain basic characteristics which apply to all such joints, and these are explained in this section.

Familiarity with the basic principles of load transfer through the bondline will help the designer understand how the detail design of bonded members affects the load carrying capability of a bonded panel. The phenomenological explanation of load transfer in adhesive-bonded structure serves also to explain which geometric variables affect the joint strengths.

10.3.2.1 The Elastic Trough in Adhesive Shear Stress Distributions

The key element in understanding shear load transfer in adhesive-bonded joints is the elastic trough of lightly-loaded adhesive between the two narrow zones

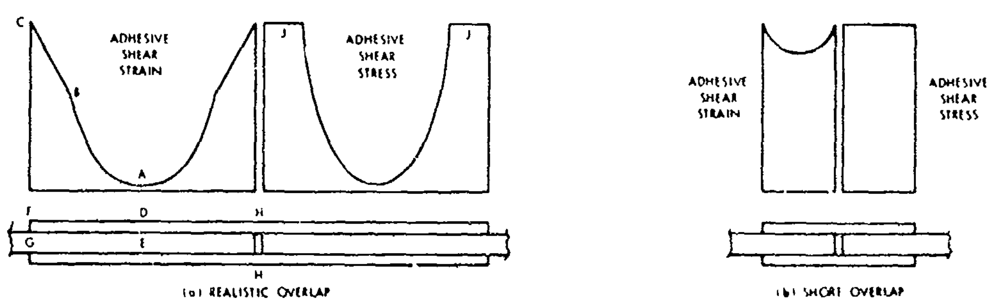


FIGURE 29. NONUNIFORM STRESSES AND STRAINS IN BONDED JOINTS

of highly stressed adhesive where the load transfer is effected. These non-uniform adhesive stresses and strains are depicted in Figure 29. The lightly-loaded trough is not inefficient since it ensures: (1) an adequate resistance to failure of the joint by creep rupture and (2) an adequate life in a possibly hostile environment. The role of the elastic trough is shown in Figure 29 (a). The total overlap must be sufficient to ensure that the adhesive shear stress in the middle of the overlap, at A, is so low that creep there cannot occur from the environment or load duration. Points A, D and E serve as a memory for the joint because there is essentially no relative motion there, no matter how high the load. The adhesive shear strains build up away from the middle of the overlap, attaining a maximum at the end, point C. If the applied load level is sufficiently high at point B, the adhesive will be loaded beyond the proportional limit. If the entire adhesive bond were strained uniformly to point C, the application of sustained load would cause all the adhesive to creep, eventually leading to complete joint failure. This is precisely what happens with short-overlap test coupons under sustained load, Figure 29 (b). This cannot happen in the realistically proportioned joint, Figure 29 (a), since there is negligible relative motion across the adhesive layer at A, from D to E. The relative motion across the bond at C, from F to G, is limited by the differential straining in the adherends. The metal stretches more from E to G than from D to F. However, unless the metal yields at G or H, the peak bond shear strain is limited by the metal distortions, no matter how long the load is maintained. Furthermore, any small adhesive creep at the ends of the overlaps is recoverable during unloaded periods because of residual stresses induced in the metal within the overlap area after unloading. Creep damage therefore, cannot accumulate. The deep elastic trough in the adhesive shear stress distribution is thus vital to ensuring a long service life of bonded structures.

Figure 30 shows the adhesive and adherend stress distributions in a bonded double-lap joint and the elastic-plastic adhesive representation used in the analysis and design of bonded joints. The extent of the end zones, through which most of the load is transferred, is defined largely by the adhesive plasticity. Strictly, the joint strength is a unique function of the adhesive strain energy in shear per unit bond area; i.e., the product of the area under

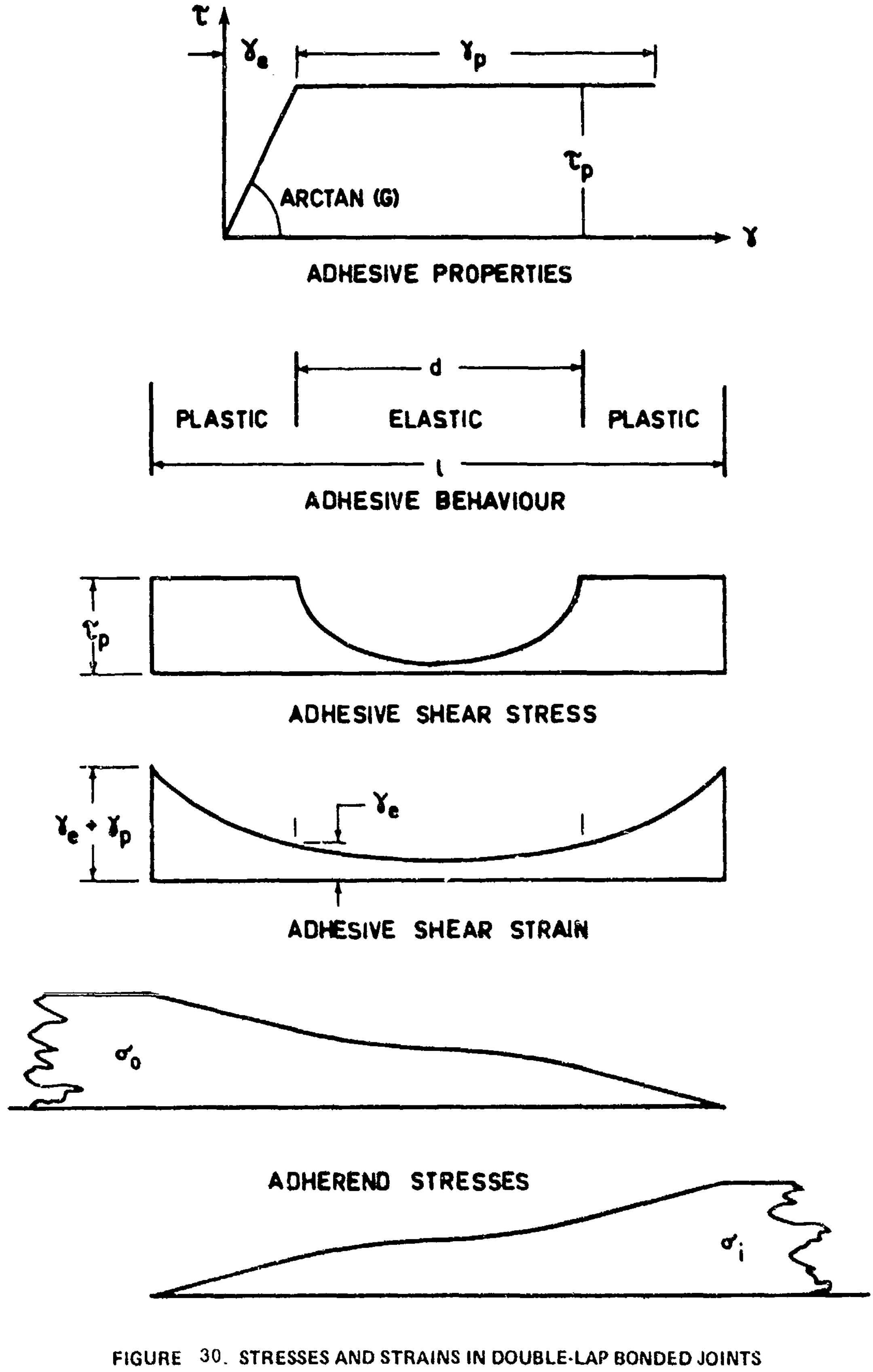


FIGURE 30. STRESSES AND STRAINS IN DOUBLE-LAP BONDED JOINTS

the top curve in Figure 10.8 and the adhesive thickness. In some circumstances, this potential shear strength may not be attained because of the prior failure of either the adhesive under peel or the adherends. Increasing the total overlap for all but very short overlaps moves the effective end zones further apart without changing the load transferred, Figure 31. The overlap must be sufficiently great to permit the elastic trough to sink low enough to prevent adhesive creep. However, further increases in overlap cannot increase the joint strength with the exception of single-lap or single-strap joints.

10.3.2.2 Shear Load Transfer in Adhesive Bonded Joints

The basic phenomena associated with shear load transfer in adhesively bonded joints due to tensile or compressive sheet loads include: (1) non-uniform shear transfer in balanced double-lap joints, (2) the influence of adherend stiffness imbalance, and (3) adherend thermal mismatch. The same local areas of high bond loads and large lightly loaded troughs apply equally to the case of in-plane shear loading, as shown in Figure 32.

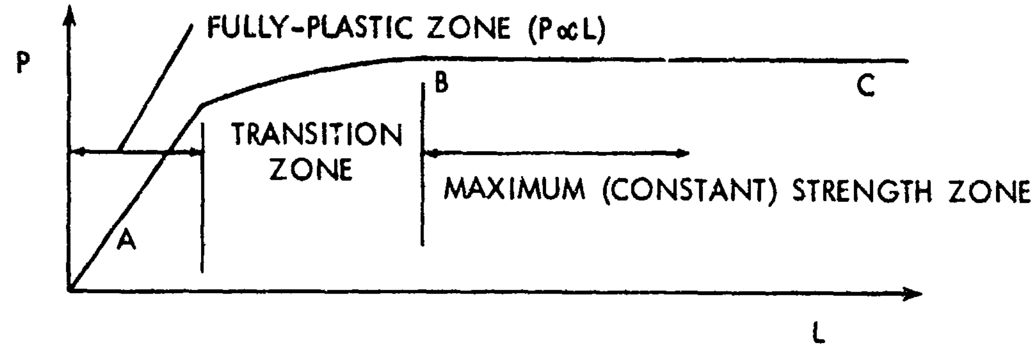
Stresses and strains induced in a bond under load arise from the differential movement of the adherends bonded together. Since the adhesive shear stress is not a constant, these three basic sources of non-uniform adhesive shear strain must be accounted for. These are discussed below for double-lap joints, but exactly the same phenomena occur in single-lap, single-strap, tapered-lap, stepped-lap, and scarf joints and bonded doublers.

(a) Balanced Double-Lap Joints

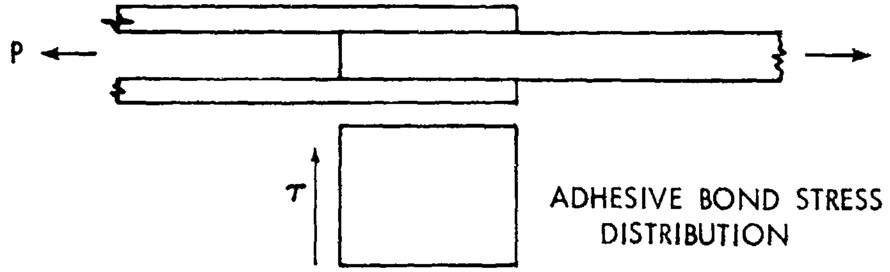
There is a strain concentration due to adherend flexibility for identical adherends, Figure 33. It can be seen that stiffer adherends promote a more uniformly loaded bond while flexible adherends have little bond-load transfer in the middle of the overlap. Figure 34 (a) explains how the uniform adherend stress in a scarf joint between identical adherends is associated with an essentially uniform bond strain and stress.

(b) Influence of Adherend Stiffness Imbalance

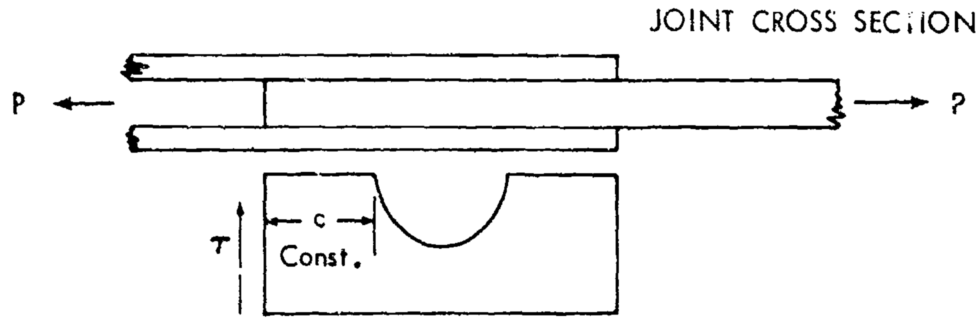
The influence of adherend stiffness imbalance is shown in Figure 35. This effect also occurs for scarf joints, as shown in Figure 34 (c). The adhesive



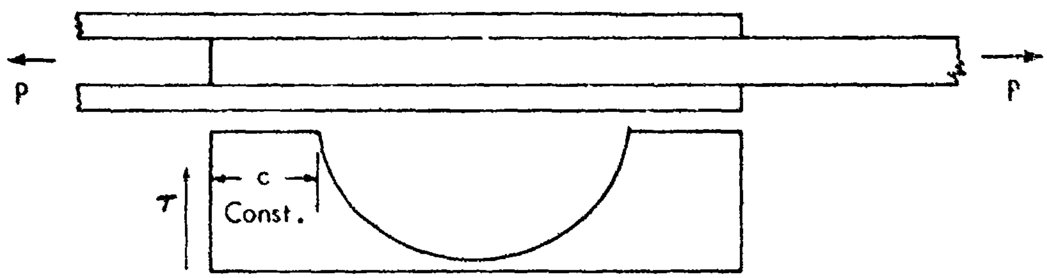
EFFECT OF LAP LENGTH ON ADHESIVE-BONDED JOINT STRENGTH



A. SHORT OVERLAP



B. INTERMEDIATE OVERLAP



C. LONG OVERLAP

FIGURE 31 INFLUENCE OF LAP LENGTH ON BOND STRESS DISTRIBUTION

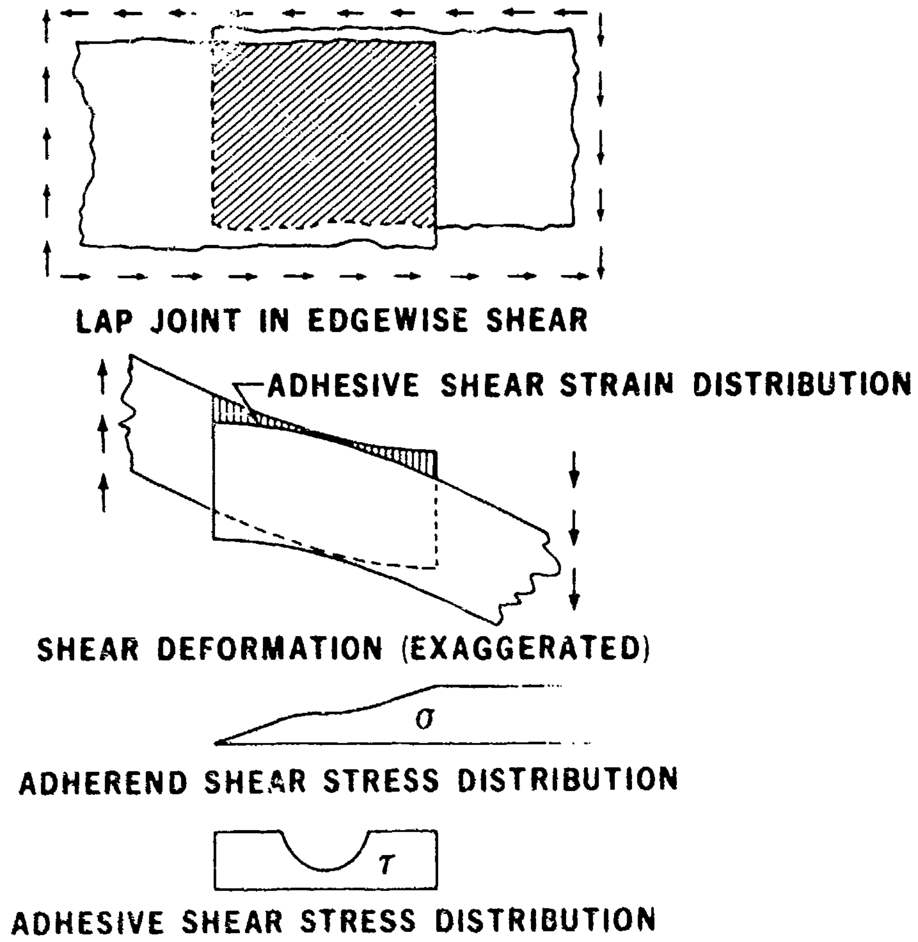


FIGURE 32 ADHESIVE-BONDED JOINT LOADED IN EDGEWISE (IN-PLANE) SHEAR

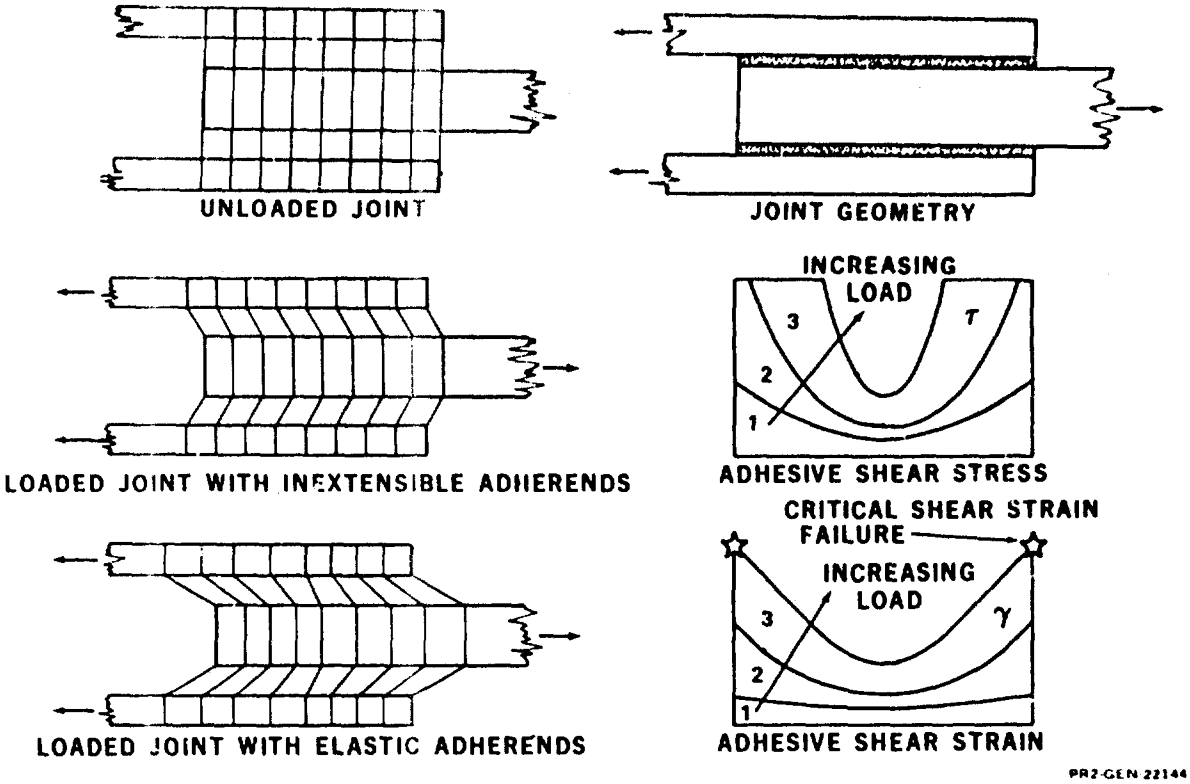
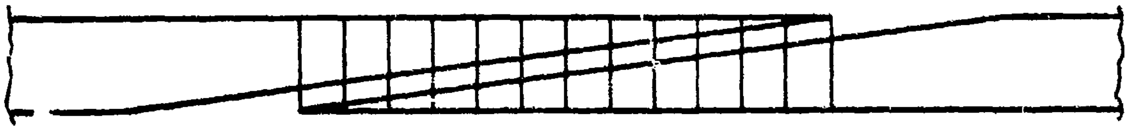
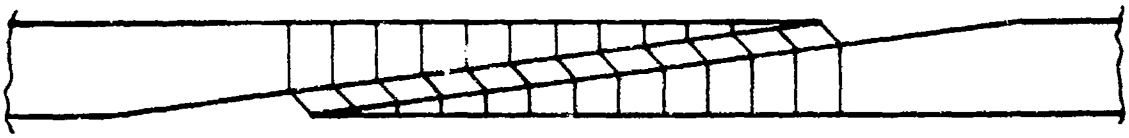


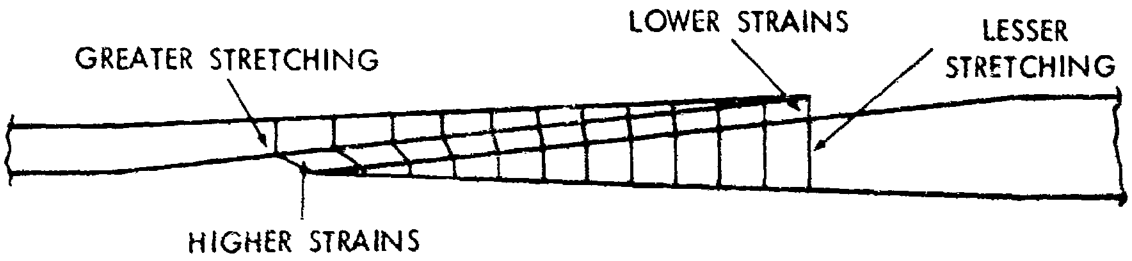
FIGURE 33. SHEARING OF ADHESIVE IN BALANCED JOINTS



(A) UNLOADED SCARF JOINT



(B) UNIFORM ADHESIVE DEFORMATION BECAUSE OF IDENTICAL STRETCHING OF IDENTICAL ADHERENDS



(C) NON-UNIFORM ADHESIVE DEFORMATION BECAUSE OF DISSIMILAR STRETCHING OF DIFFERENT ADHERENDS

FIGURE 34 SCHEMATIC EXPLANATION OF ADHESIVE SHEAR STRESS AND STRAINS, BALANCED AND UNBALANCED SCARF JOINTS

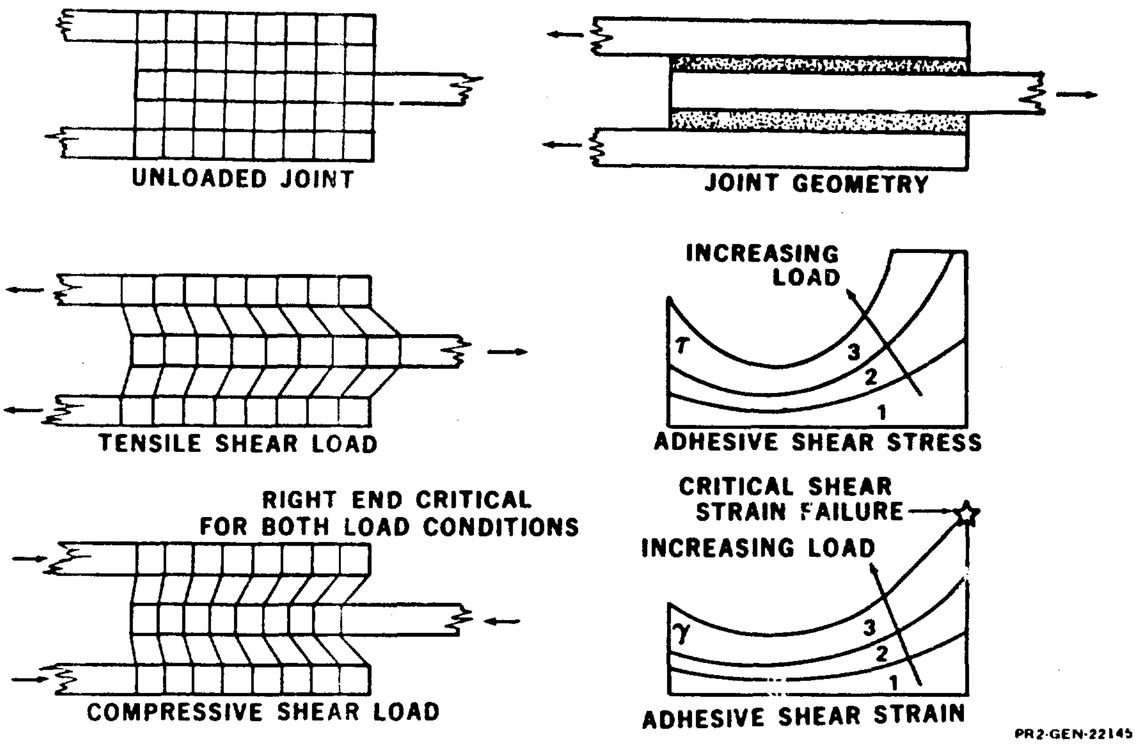


FIGURE 35 EFFECT OF ADHEREND STIFFNESS IMBALANCE ON ADHESIVE SHEAR

shear strains are intensified at the end from which the less stiff adherend(s) extend(s). The same end is critical whether the shear load is tensile or compressive. In comparison with a stiffness-balanced joint, this imbalance reduces the joint strength by unloading the less critical end. In a balanced joint there is an equally effective end zone at each end of the joint, as shown in Figure 33 .

(c) Effect of Adherend Thermal Mismatch

A further reduction in bond shear strength is caused by any adherend thermal mismatch, which is acute for some of the metal-to-composite combinations and still significant for aluminum-to-titanium joints. The problem arises be-ause high-strength adhesives are customarily cured at temperatures far above their operating temperature. The mechanics of this imbalance are shown in Figure 36 with the aluminum, which has a higher coefficient of thermal expansion, sandwiched in the middle. The aluminum shrinks more than the titanium during the cooling down to operating temperature. This shrinkage is partially resisted by the outer titanium adherends-thereby setting up residual bond stresses. The important characteristic of the so-called thermal stress imbalance is that the critical end of the joint changes with the direction of application of the load. If the overlap between the bonded dissimilar metals is short enough; e.g., up to about an inch, the adhesive creep can eliminate or, at least, minimize this problem provided that the temperature excursion of the bonded structure in service is not too great. However, for very long overlaps, the thermally induced residual stresses will remain because the adhesive cannot possibly creep far enough to relieve such stresses.

The consequence of simultaneous adherend stiffness imbalance and thermal mismatch is that the joint strength can change between tensile and compressive loading.

10.3.2.3 Peel Stress Problems and Alleviation in Adhesive Bonded Joints

The other dominant characteristic of adhesive-bonded joints is the peel stresses induced by the shear stresses. Figure 37 shows how the peel stresses balance the shift in the axis of the shear load in the bond to the stretching load in the adherend. With a thick metal adherend, the failure will

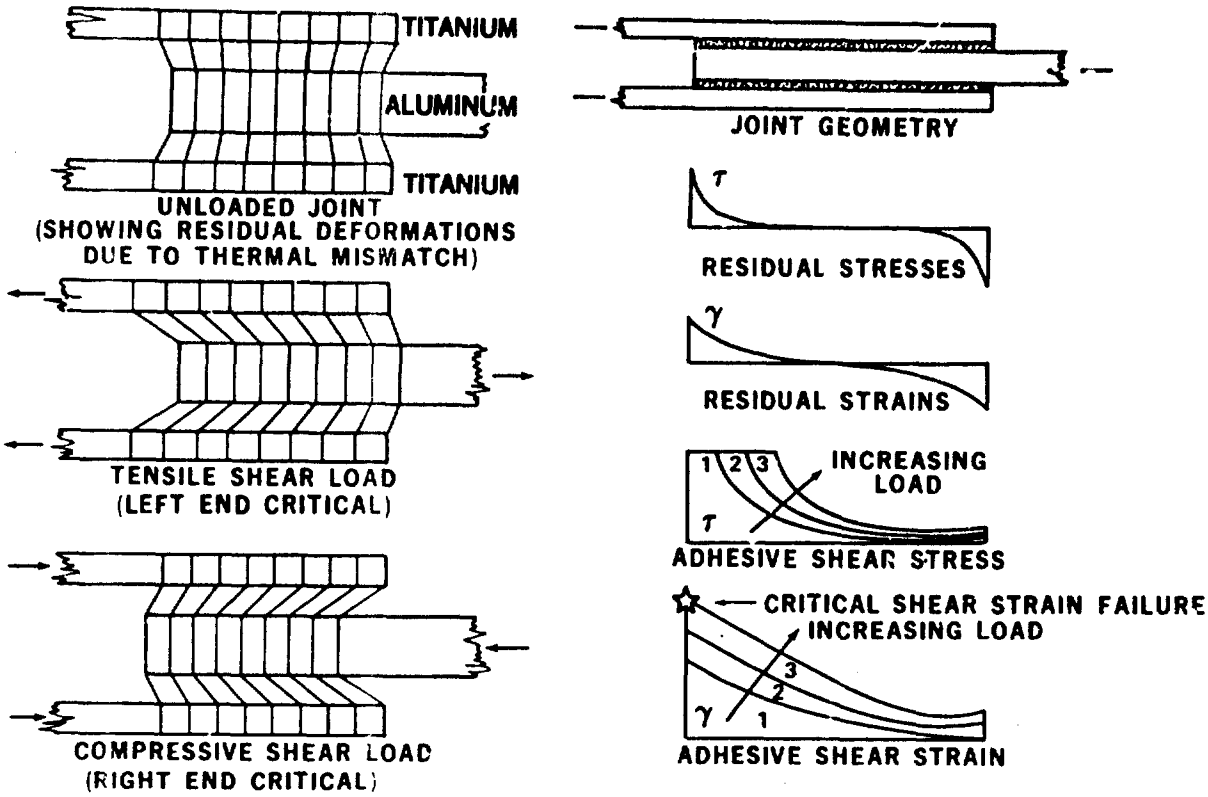


FIGURE 36 EFFECT OF ADHEREND THERMAL MISMATCH ON ADHESIVE SHEAR

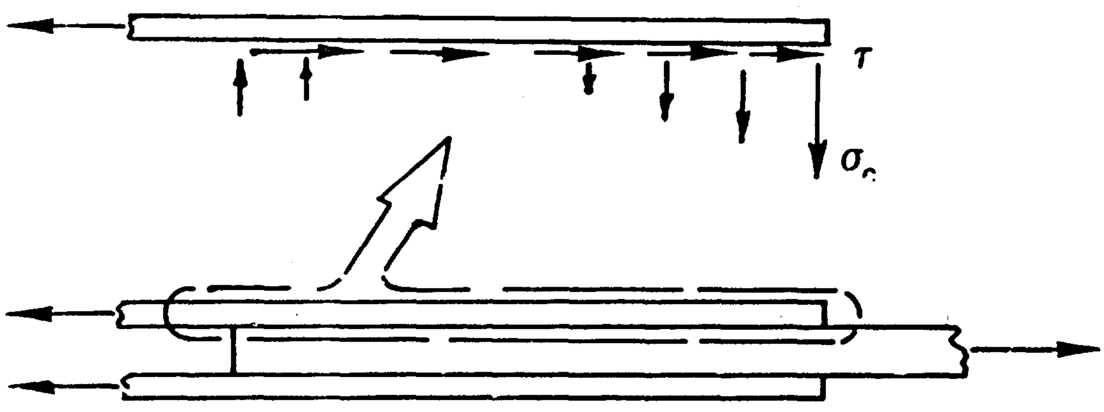


FIGURE 37. STRESSES ACTING ON OUTER ADHEREND

be peel in the adhesive preventing the attainment of the potential shear strength.

The peel stress problem is not insurmountable. For example, some of the excess shear strength can be traded off for additional peel strength as shown in Figure 38. Some shear strength at the outer ends of the joint where peel stresses are critical, is sacrificed by thinning the adherend there. This permits it to deflect with the peel stresses, thus reducing them. The loss in shear strength occurs since the ends of the thinned outer adherends can stretch more easily with the loaded inner adherend. Consequently, less shear load is transferred at that end. The uniform end of the joint, at which the adhesive normal stresses are compressive, thus becomes critical in shear first. Strain compatibility then prevents development of the full load potential at the tapered end of the overlap. The loss in shear strength referred to above can be nullified by using an adherend stiffness imbalance to counteract the uneven bond strains at the ends of the overlaps. If the splice plates are thickened in the middle, Figure 39, the relative adherend displacements there are reduced in relation to those at the outer ends. A fully plastic adhesive analysis of this problem indicates that the optimum excess thickness (really the stiffness, Et), is 32.5 percent beyond that of a balanced joint. Not only does this thickening nullify the strength loss due to the taper, it also raises the shear strength of the joint to 24 percent above the potential shear strength of a balanced uniform joint.

10.3.2.4 Effects of Joint Geometry on Bonded Structures

(a) Load Intensity

Figure 40 shows the relative strengths of the various bonded joint types and the relative range of adherend thicknesses over which each should be used. The joint configuration employed should never be weaker than the adherends being used nor cost more than is necessary. It should be noted that tapered thicknesses complicate the NDI considerably and sometimes make the task of establishing bond adequacy an impossible one. In addition, separate

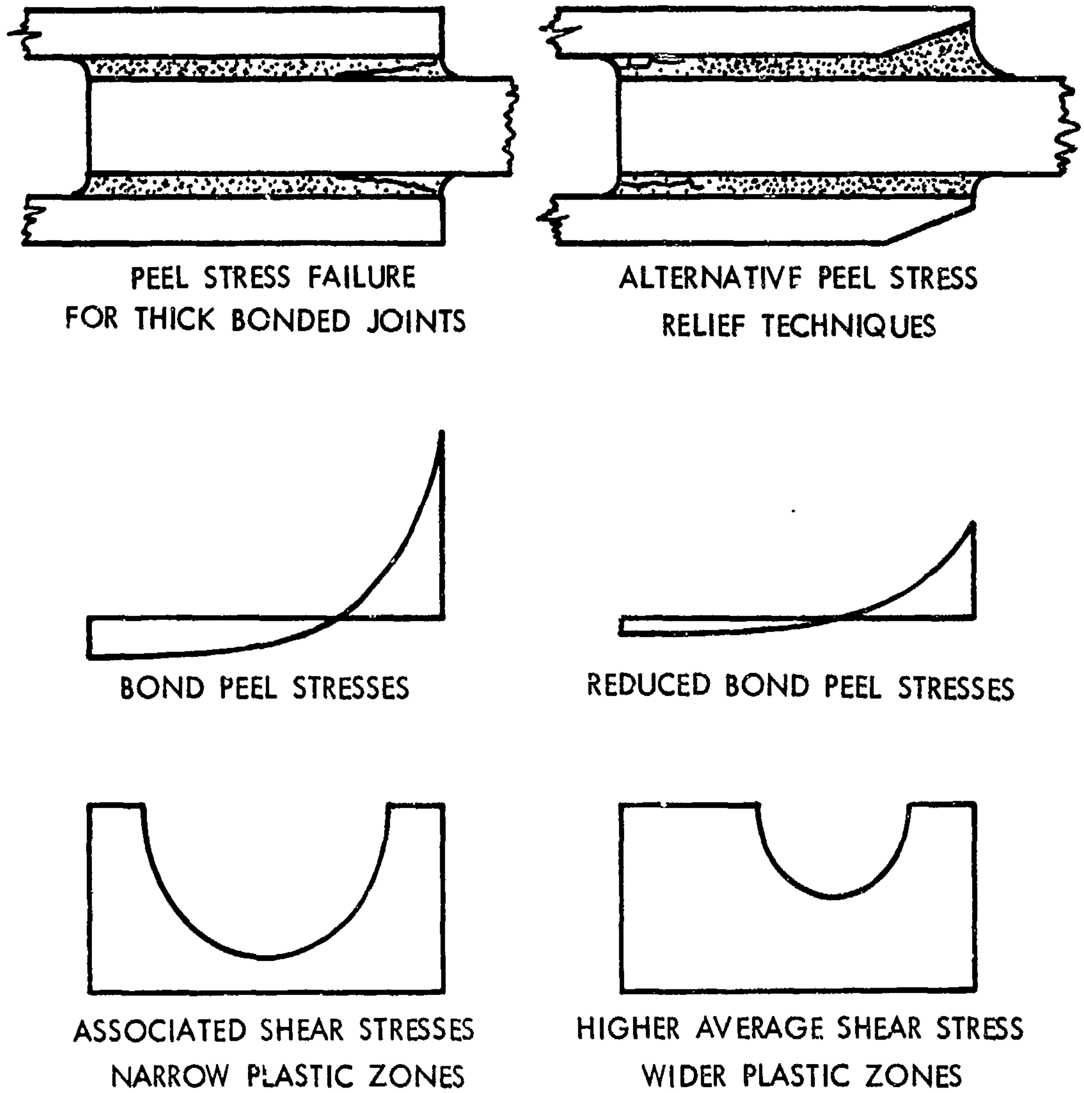
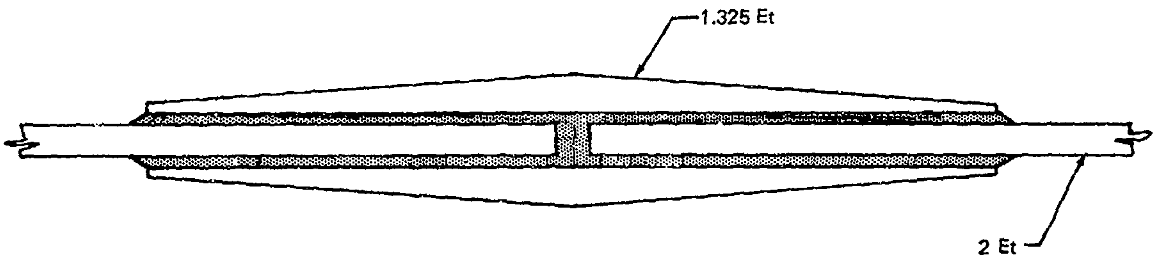


FIGURE 38 RELIEF OF PEEL STRESS FAILURE OF THICK ADHEREND BONDED JOINTS



SHEAR STRENGTH OF ADHESIVE
24 PERCENT GREATER
THAN UNIFORM BALANCED JOINT BELOW

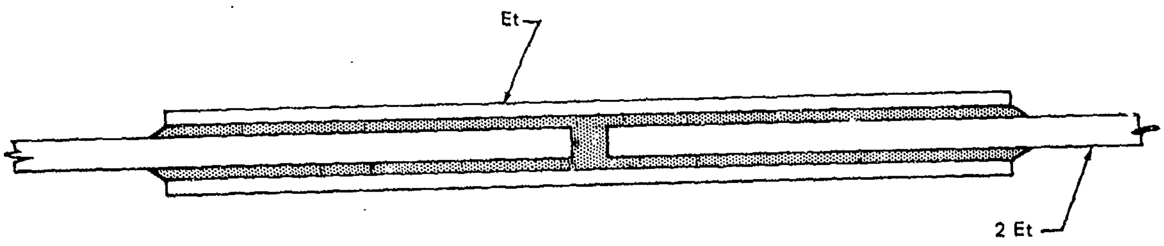


FIGURE 39 **OPTIMUM TAPERED-LAP BONDED JOINT**

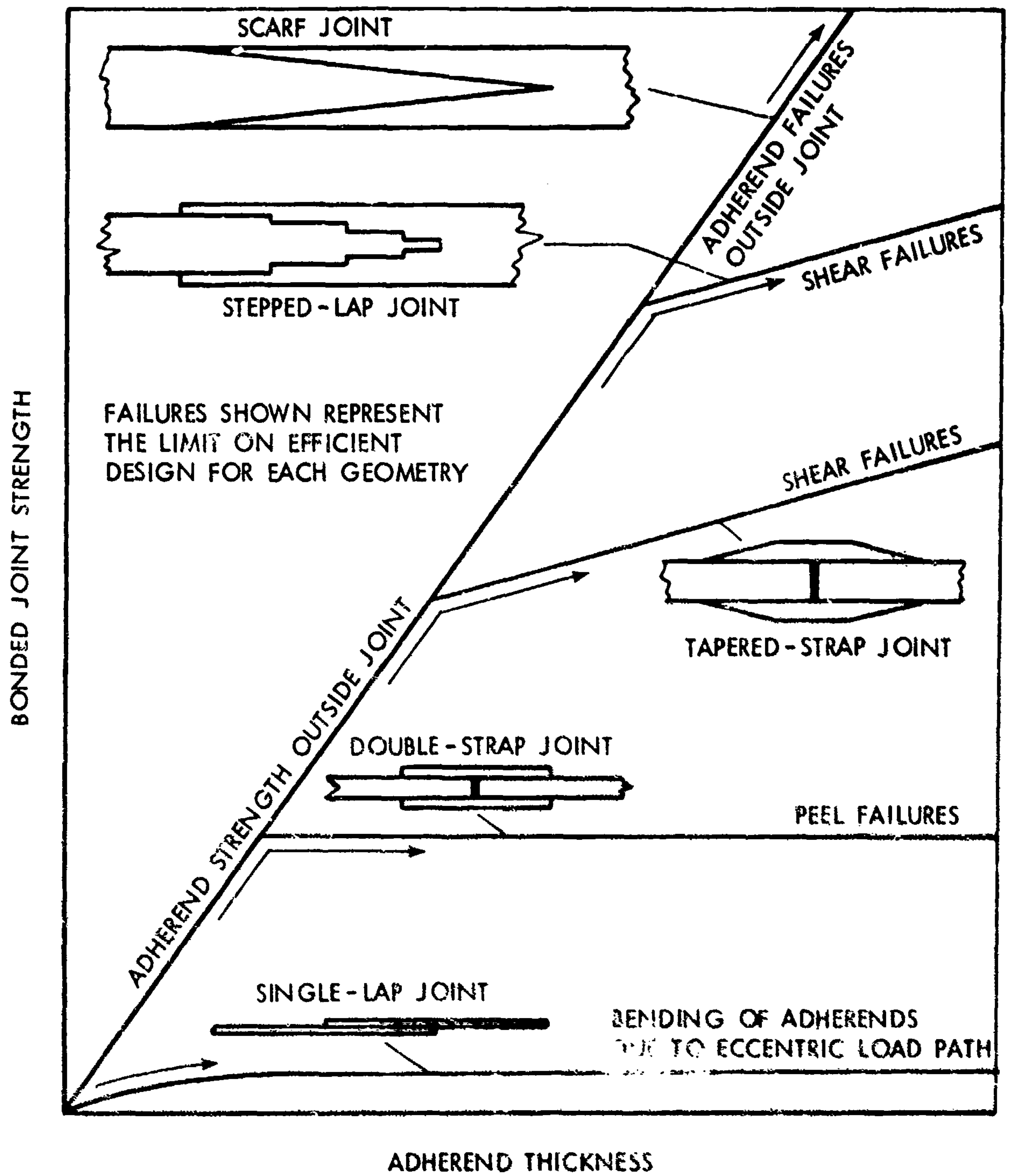


FIGURE 40 INFLUENCE OF MEMBER SIZE ON SELECTION OF OPTIMUM JOINT CONFIGURATION

calibration specimens and test machine settings are needed for each step of a stepped-lap joint. The higher strength stepped-lap and scarf joints transfer load over most of the bond surfaces, instead of just in a narrow strip at each end of the overlap; therefore, more extensive inspection is required. Such joints are more prone to failure in service by creep-rupture because of the lack of a large elastic trough.

Analysis of bonded uniformly thick adherends has established that, whereas the shear strength of an adhesive joint is proportional to the square root of the adherend thickness, the peel strength is proportional to the quarter power of that thickness, Section XI. The applicable load is limited only by the thickness itself. Consequently, for very thin uniform adherends, the joint allowable strength is the adherend strength, with a greater unrealizable potential shear strength and an even greater resistance to the induced peel loads. For thin uniform adherends, the weak link is the adhesive in shear, still with an excess of peel strength. For moderately thick uniform adherends, the weakness is in the peel strength of the adhesive, preventing the attainment of the potential adherend strength. Very thick sections should not be bonded. They should be mechanically fastened or replaced by a structure of thin laminations for bonding.

A critique of various joint configurations is presented in Figure 41 .

Figure 2 shows that the peak adhesive bond shear strain at the ends of the overlap may not attain the full potential value because the adherends(s) may not be strong enough. Similarly, the load level applied at any instant is usually less than the maximum design condition. It is appropriate, therefore, to examine the effect on the adhesive bond stresses and strains of operating at less than full capacity. Figure 42 explains how the maximum adhesive strains for partial load levels are proportionally less than the load fraction. For example, a load level of 20 percent of the maximum value is associated with a maximum strain of only seven percent of ultimate in the case shown. Therefore, fatigue failures in adhesive-bonded joints usually occur in the adherends instead of in the adhesive. Figure 10.20 uses a double-lap

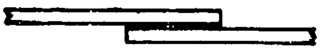
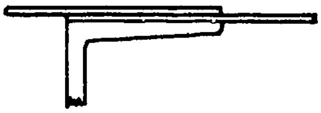
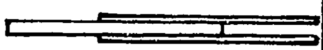
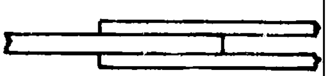
JOINT	COMMENTS
1.  SINGLE-LAP (UNSUPPORTED) JOINT	NONSTRUCTURAL JOINT HAVING LOW EFFICIENCY (FOR SHORT OVERLAPS) BECAUSE OF BENDING OF THE ADHEREND DUE TO THE ECCENTRICITY IN LOAD PATH. THICK ADHERENDS ARE ASSOCIATED WITH FAILURES BY PEEL RATHER THAN BY SHEAR. FOR THIN ADHERENDS, THESE JOINTS CAN BE GIVEN REASONABLE EFFICIENCY BY ADEQUATE (>80:1) OVERLAPS.
2.  SUPPORTED SINGLE-LAP JOINT	PRACTICAL JOINT FOR THIN ADHERENDS. NEEDS TO BE MOUNTED ON MOMENT-RESISTANT SUPPORT TO AVOID LIMITATIONS ABOVE. JOINT LOAD CAPACITY DOES NOT INCREASE INDEFINITELY WITH OVERLAP. LOAD CAPACITY IS LIMITED BY SINGLE BOND SURFACE.
3.  SUPPORTED SINGLE-STRAP JOINT	SAME AS FOR 2. SUITABLE FOR FLUSH EXTERIOR APPLICATIONS BUT LIMITED TO THIN ADHERENDS, AND NEED EITHER GOOD MOMENT-RESISTANT SUPPORTS OR VERY LARGE ℓ/t RATIOS.
4.  BALANCED DOUBLE-LAP JOINT  BALANCED DOUBLE-STRAP JOINT	EFFICIENT PRACTICAL JOINTS FOR THIN AND MODERATELY THICK ADHERENDS. SIMPLE JOINT WITH TOLERANT FIT REQUIREMENTS. JOINT STRENGTH LIMITED BY ADHEREND THICKNESSES AND INDEPENDENT OF OVERLAP BEYOND VERY SHORT (UNIFORMLY STRESSED) LENGTHS OF BOND. MAXIMUM STRENGTH LIMIT IS SET BY PEEL STRESSES FOR MODERATELY THICK ADHERENDS, RATHER THAN BY ADHESIVE SHEAR STRESSES. FOR THIN ADHERENDS, PEEL STRESSES ARE NEGLIGIBLE AND SHEAR STRENGTH USUALLY EXCEEDS ADHEREND STRENGTH.
5.  UNBALANCED DOUBLE-LAP JOINT	WEAKER THAN NO. 4 BECAUSE ONLY THAT END OF THE JOINT FROM WHICH THE THIN (LOWER Et) ADHEREND EXTENDS IS LOADED TO ITS CAPACITY. OTHER END HAS UNUSABLE RESERVE.
6.   TAPERED-LAP JOINT	EFFICIENT PRACTICAL JOINTS FOR MODERATELY THICK ADHERENDS. OVERCOMES PEEL-STRESS LIMITATIONS OF NO. 2. STRENGTH LIMITED BY ADHESIVE SHEAR STRENGTH FOR THICK ADHERENDS. BEST STRENGTHS ARE OBTAINED WITH OPTIMUM STIFFNESS IMBALANCE BETWEEN ADHERENDS TO COMPENSATE FOR SHEAR STRENGTH LOSS DUE TO TAPER. ONLY MODERATE PRECISION REQUIREMENTS.

FIGURE 41. BONDED JOINT CONCEPTS

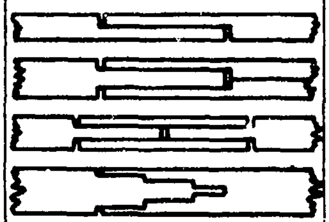
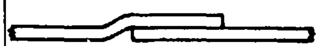
JOINT	COMMENTS
7.  FLUSH JOINTS	NONSTRUCTURAL JOINTS SUFFERING FROM NET SECTION LOSS JUST OUTSIDE THE JOINT REGIONS.
8.  JOGGLED LAP JOINT	NONSTRUCTURAL JOINT USED (BECAUSE OF AERODYNAMIC SMOOTHNESS REQUIREMENTS) ON EXTERIOR SKINS SUBJECTED TO NORMAL RATHER THAN IN-PLANE LOADS. SEE ALSO COMMENTS ON NO. 1.

FIGURE 41. BONDED JOINT CONCEPTS (CONCLUDED)

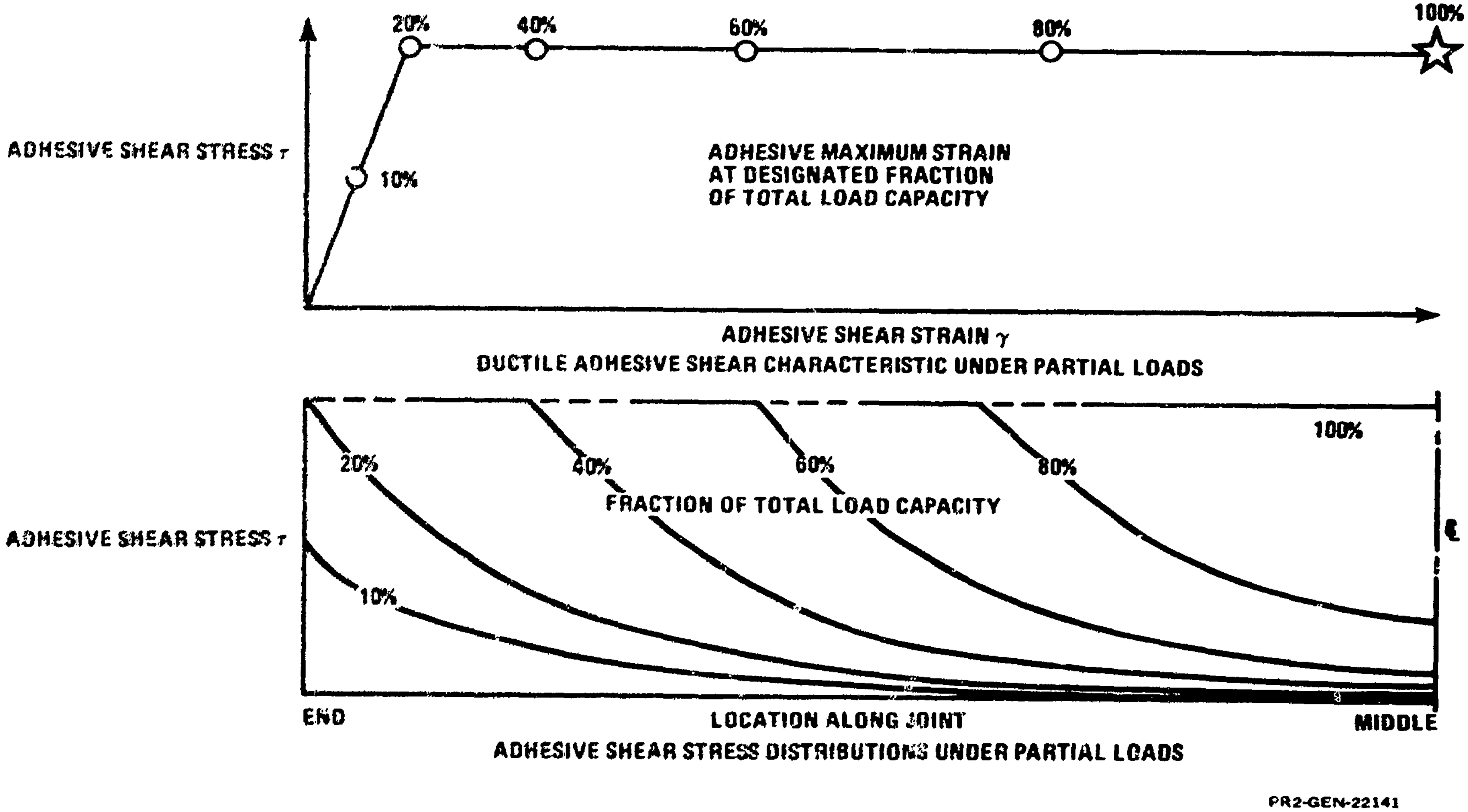


FIGURE 42. DOUBLE-LAP JOINTS UNDER PARTIAL LOADS

joint to illustrate the features of the adhesive shear stress distributions but the same characteristics apply to almost all bonded joints. The sole exception is a scarf joint between identical adherends, for which the adhesive is uniformly stressed along its length. The overlap in the figure has been selected so that the ultimate load capacity is associated with a fully plastic adhesive throughout. Realistic overlaps would retain an elastic trough, even at ultimate load. The adherends of a properly proportioned joint thus restrict the maximum adhesive strain for any given load level, preventing creep in the adhesive.

(b) Eccentricities in Load Path

Single-lap and single-strap bonded joints differ from all the other bonded joints in that these eccentric joints exhibit a marked dependence on the overlap-to-thickness ratio of the adherends. The distinctive failure phenomena are depicted in Figure 43 for ductile (metal) adherends. The failure is almost invariably initiated by the high bending moment induced at the end(s) of the overlap. The failure is usually manifested as: (1) an overloading of the adherend under the combined bending and stretching or (2) peel in the adhesive. Only rarely is the failure of such joints associated with adhesive shear characteristics. The bending moment induced in the adherends is greatly dependent on the ℓ/t ratio because the adherends can easily deflect towards the load axis to relieve the bending moment for long overlaps. Short overlaps would necessarily involve higher curvatures and bending stresses. Consequently single-lap joints should have very large overlaps, typically $\ell/t = 50$ to 100, not to decrease the adhesive shear stresses but to decrease the bending stresses in the adherends. Figure 44 quantifies this joint inefficiency as a function of ℓ/t ratio. The abscissa closely approximates the ℓ/t for aluminum alloy adherends. For example, at an ℓ/t ratio of 10, the maximum average aluminum stress that can be developed far away from the joint is about 25 ksi because, at that load level, the maximum aluminum stress is 70 ksi at the end of the overlap. In other words, the joint efficiency is only about 35 percent. Increasing the ℓ/t ratio to 50, raises the joint efficiency to 80 percent for the same maximum stress. The high adhesive peel stresses inherent with short single overlaps are also decreased for long overlaps.

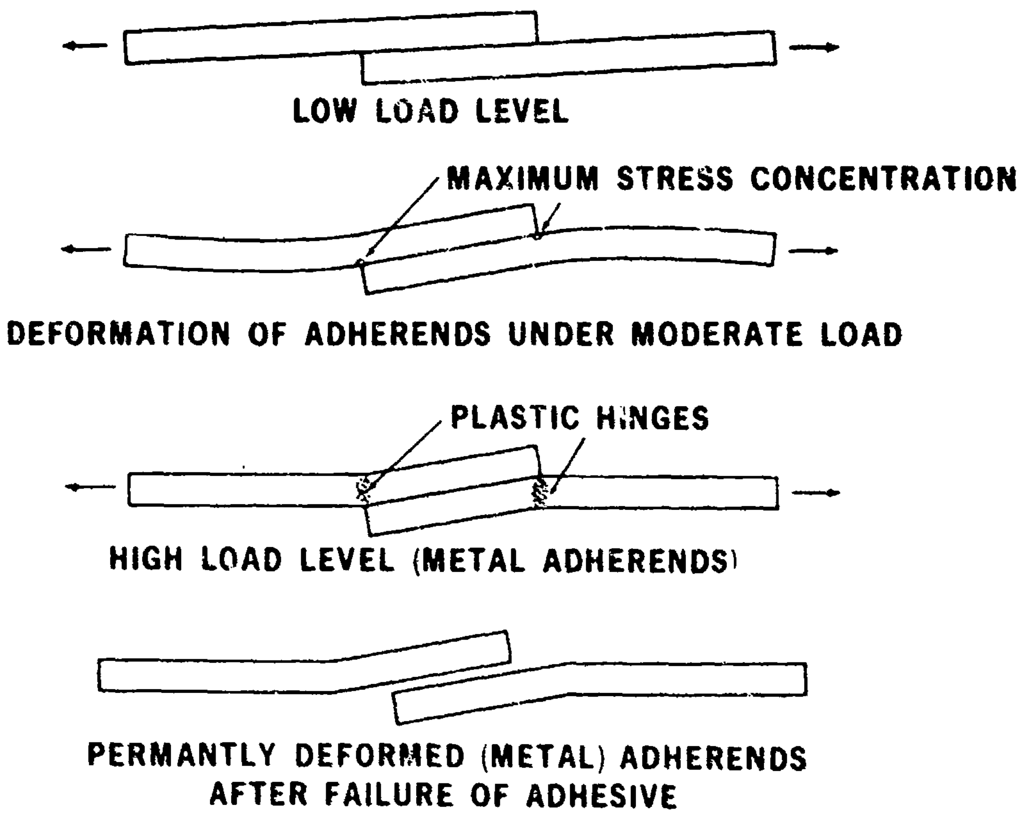


FIGURE 43. FAILURE OF SINGLE-LAP BONDED JOINTS WITH YIELDING ADHERENDS

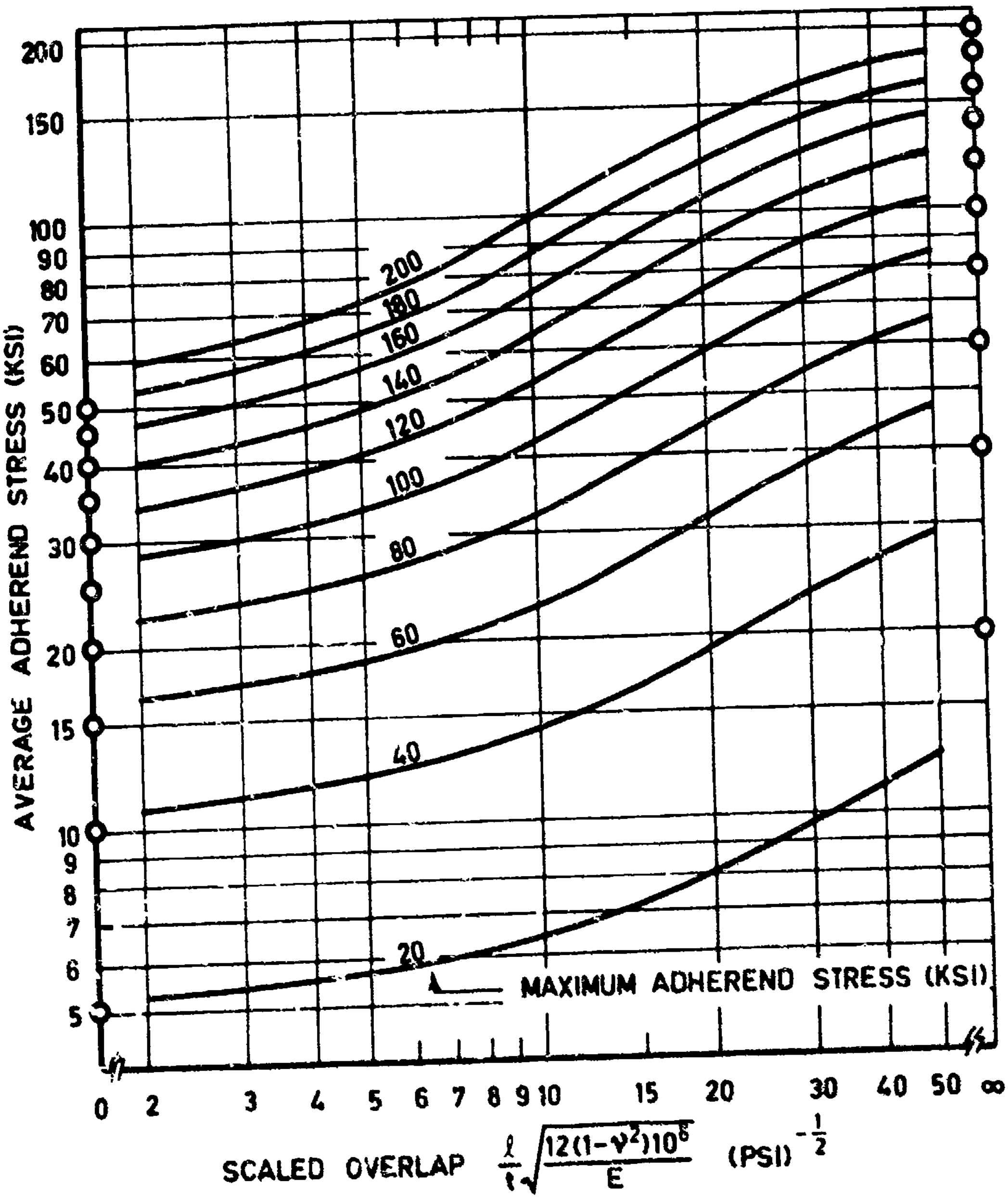


FIGURE 44. ADHEREND LIMIT LOADS FOR BALANCED SINGLE-LAP JOINTS

(c) Adhesive Properties

The adhesives used in subsonic transport aircraft have considerable ductility in shear. Since the joint bond strength is proportional to the square root of the strain energy in shear, this plasticity may represent a strength increase of from five to ten times the purely elastic strength of the joint. It is good practice to restrict the adhesive to elastic-strain only for the regularly occurring fatigue loads but extra strain capability is necessary for ultimate loads and for load redistribution around flaws.

Figure 45 shows a typical set of adhesive shear stress-strain curves as a function of temperature. It is important to note that, while a decrease in temperature makes the adhesives more brittle, the strain energy is not changed appreciably, so the joint shear strength would not be very sensitive to temperature. However, adhesives exhibit an appreciable decrease in peel strength at low temperatures, so joint strength may still decrease. It should be mentioned that FM-73, used on PABST, is so formulated that it retains a considerably higher sub-zero peel strength than older adhesives. This has been accomplished by relinquishing some of the very high room-temperature peel strength associated with the older ductile adhesives. The stress-strain characteristics of FM-73 adhesive are shown in Figure 46. At elevated temperatures, the ductile adhesives exhibit a considerable decrease in peak shear stress with an accompanying increase in ultimate elongation. This makes creep failure of the adhesive under sustained load more likely. It will also cause a decrease in adhesive shear strength if the elastic trough had been sized for a colder condition. Therefore, the highest operating temperature is usually the design condition. It requires the longest plastic end zones

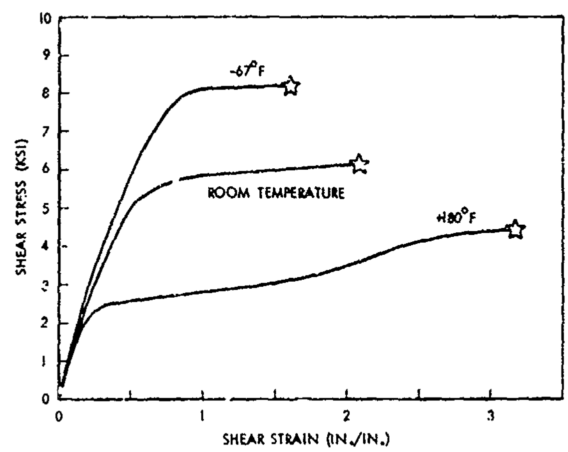


FIGURE 45. EFFECT OF TEMPERATURE ON ADHESIVE STRESS-STRAIN CURVES

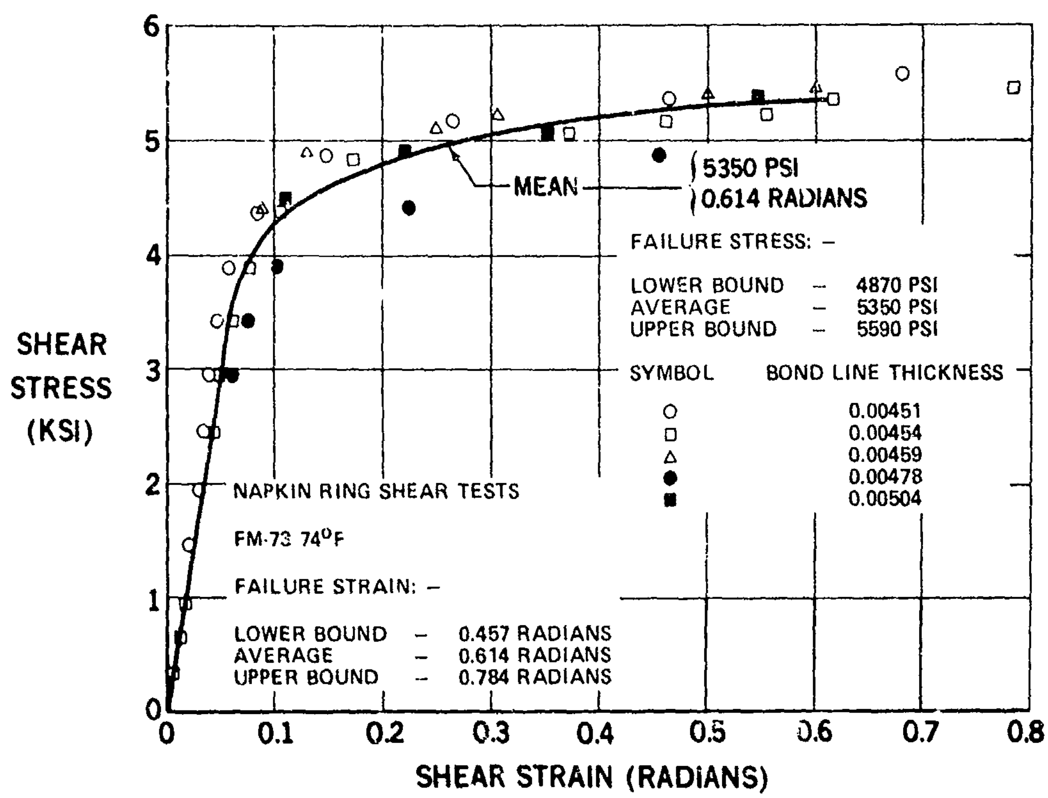


FIGURE 46. FM-73 ADHESIVE STRESS-STRAIN DIAGRAM

to transfer the load, because the peak shear stress is least; and requires the longest elastic trough, because of the lowest shear modulus.

The effect of absorbed moisture in the adhesive on the load transfer regions is important and will be explained qualitatively. An increase in moisture content changes the adhesive mechanical properties in much the same manner as an increase in temperature, see Figure 47 ; i.e., water softens the adhesive. Other long term environmental effects may permanently degrade the structure as well by attacking the adhesive-to-adherend interface. This problem can be solved by improved processing.

A uniformly wet adhesive would have wider effective end zones operating at a lower peak adhesive shear stress than a dry joint would, Figure 47 . If only the outer portions of the adhesive were wet, while the interior remained dry, the load transfer regions would be the same as if all the adhesive was wet. The elastic trough would be slightly less lightly loaded because the dryer adhesive would be more stiff than a wetter one. If the interior of the adhesive was wet, while the edges were dried out, the load transfer zones would be the same as for a completely dry adhesive and the elastic trough would be not quite so deep. However, since the interior adhesive would swell because of the absorbed moisture, the inner portions of the adhesive would induce severe peel loads at the dry periphery of the joint. Thus the peak shear and peel stresses in the adhesive would be made to coincide, reducing the joint strength below that of a dry joint. These phenomena are illustrated in Figure 47.

Good bondlines are uniform and in the range of 0.005 to 0.010 inches thick. However, with the newer environmentally resistant adhesives, the flow during cure is so great that 0.002 inch is more typical, as discussed in Section 10.6.1, with thicknesses as little as 0.0005 inch where edges and corners are pinched off. The consequent loss of bond strength must be accounted for in design and analysis. As previously stated the bond shear strength in a joint is proportional to the square root of the bond thickness.

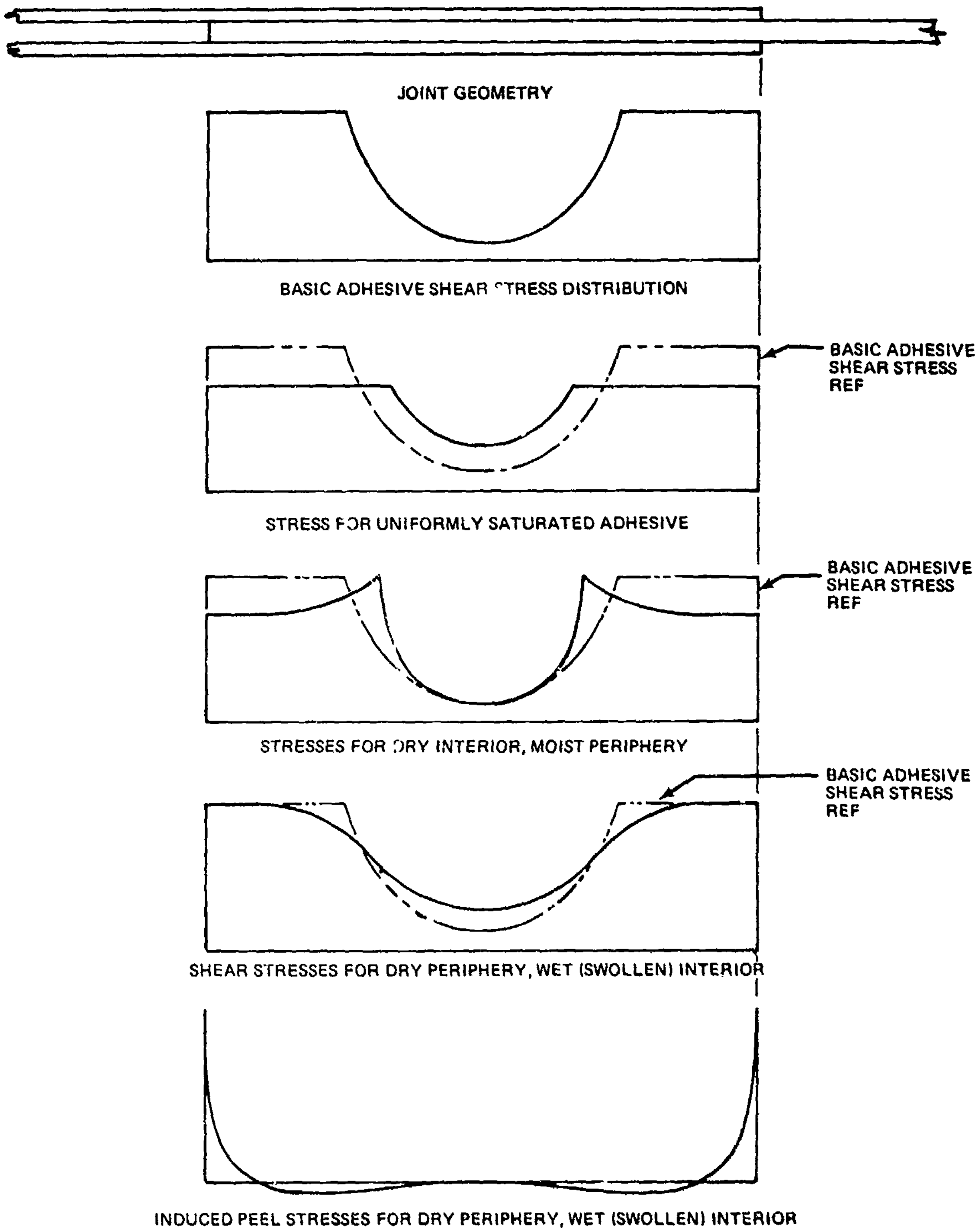


FIGURE 47 . EFFECT OF MOISTURE IN ADHESIVE ON BOND STRESS DISTRIBUTIONS

The use of adhesive to fill up the gaps where parts don't fit is incompatible with efficient transfer of load from one member to another and can be tolerated only for very thin gages of metal for which the potential bond strength is several times as great as that of the metal.

The adhesive should be looked upon as a load transfer medium which should be as uniform as mechanical fastener sizes are required to be.

10.3.3 Bonded Doublers

There is a widespread misconception about adhesively bonded doublers which warrants discussion. The belief that the adhesive in bonded doublers is less highly loaded than in bonded joints is fundamentally unsound, as explained in Figure 48 . If the various thicknesses and the applied metal stress are the same, then the peak bond stresses and strains must be the same. The bond transfers precisely the same load over the distance to the right of the left edge of the splice as it does in the same length of doubler. The significance is that (1) bonded doublers which have remained bonded in service serve as a precedent for low-risk extension of bonding to more critical applications while (2) those which have failed in service, due to inadequate environmental resistance, serve as a warning not to use bonding in that application.

Just as with single-lap and double-lap joints, it is important to minimize adhesive peel stresses in bonded doublers by restricting the doubler thickness where it joins the loaded side of the sheet. A thickness restriction of 0.040 inch; i.e., using tapered edges above that thickness, is reasonable. The effective load zones in the adhesive are of the order of 0.25 to 0.5 inch for most aerospace applications.

10.4 Bending from Eccentric Joint Loads

10.4.1 Single-Lap Bonded Joints

The key characteristic of single-lap joints, Figure 49 , is that, under no circumstances can such a joint ever be as strong as the members being bonded together since they represent an eccentricity in load path and, consequently, induce a bending moment in the adherends which adds to the membrane stresses.

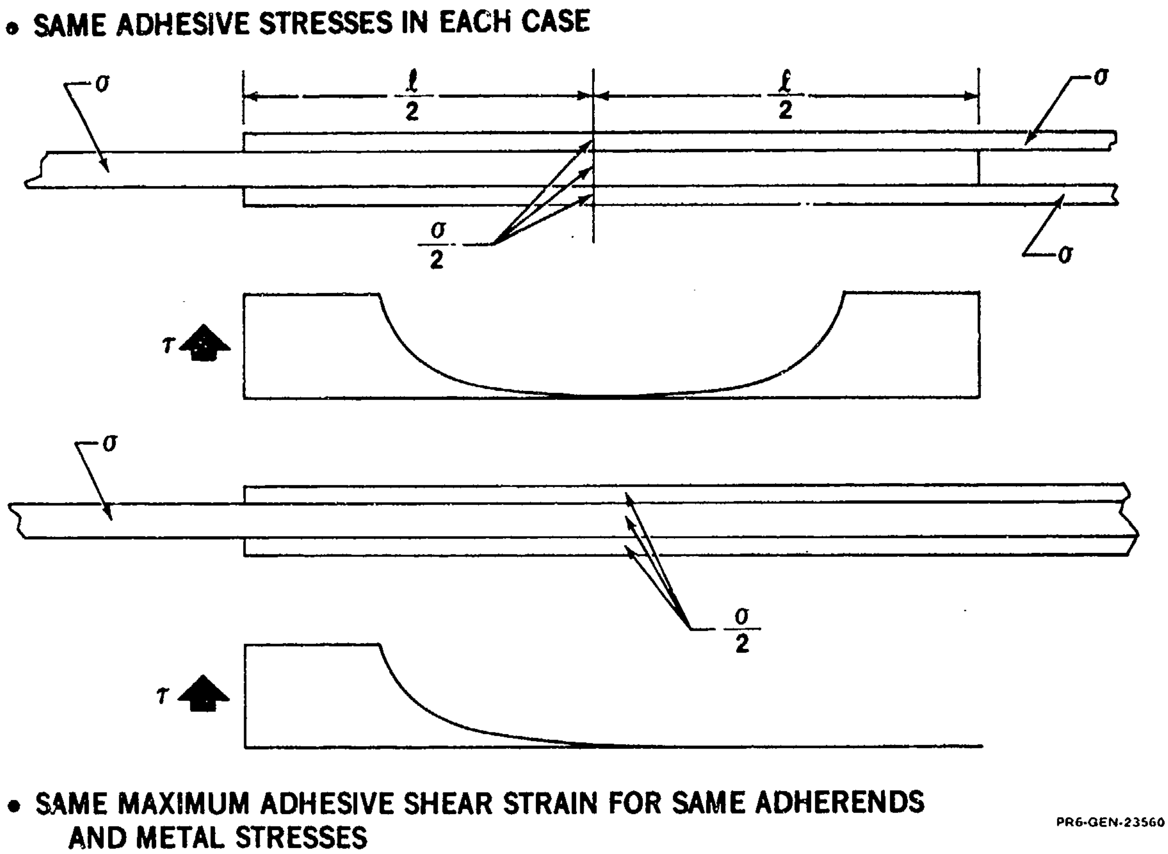
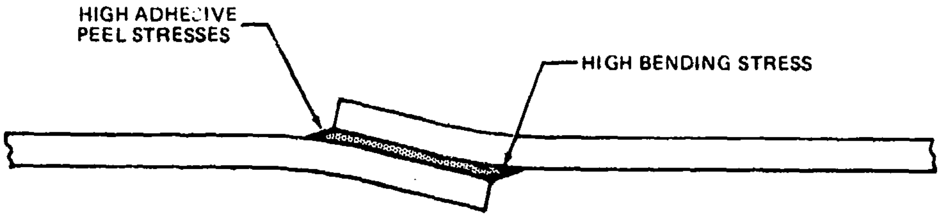
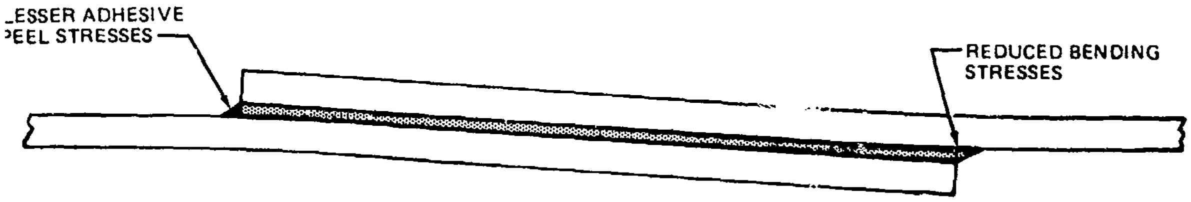


FIGURE 48. DOUBLERS VERSUS JOINTS



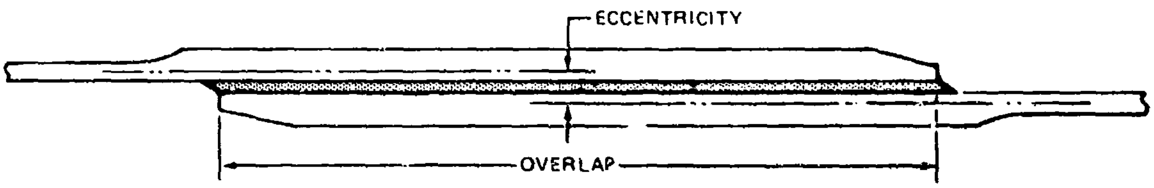
DISTORTION UNDER LOAD OF SHORT OVERLAP JOINT



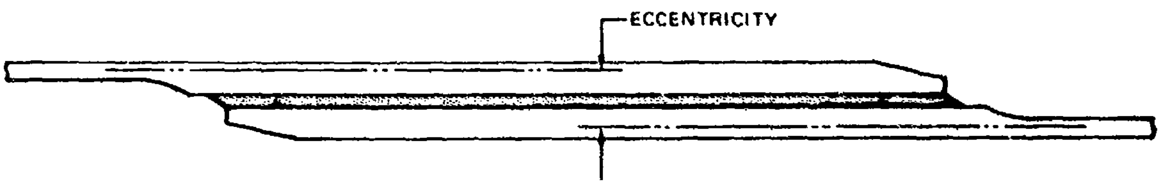
DISTORTION UNDER LOAD OF LONG OVERLAP JOINT

NOTE: ADHESIVE SHEAR STRESSES USUALLY NOT OF CONCERN
(STILL PEAK AT ENDS OF OVERLAP, LIKE DOUBLE-LAP JOINTS)

PEEL STRESS RELIEF, FOR GAUGES GREATER THAN 0.050 INCH

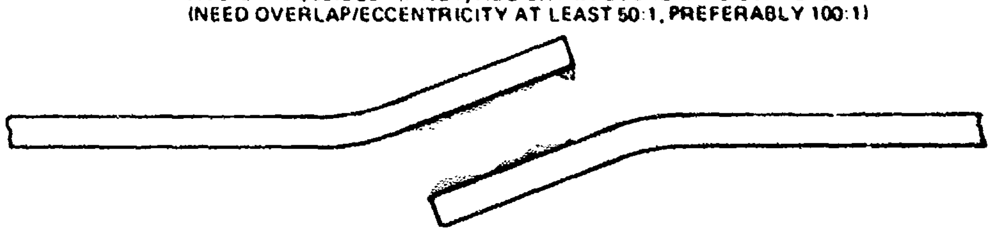


MINIMUM ECCENTRICITY IN LOAD PATH IF PANELS ARE CHEM-MILLED



AGGRAVATED ECCENTRICITY IN LOAD PATH

AVOID IF POSSIBLE. IF NOT, ADD EXTRA OVERLAP TO COMPENSATE
(NEED OVERLAP/ECCENTRICITY AT LEAST 50:1, PREFERABLY 100:1)



PLASTIC HINGES OFTEN FORM IN METAL PRIOR TO FAILURE

FIGURE 49 SINGLE-LAP BONDED JOINTS

For short overlaps, the bending stresses in the adherends, at the ends of the overlap, can approach three times the membrane stresses. Such a "joint" is, therefore, a built-in stress concentration factor of 4. Fortunately, this induced bending moment is a strong function of the overlap-to-eccentricity ratio and can be alleviated considerably by using large ℓ/t ratios; i.e., from an absolute minimum of 50:1 for non critical joints up to a desirable ratio of 100:1 for important joints. The desirability of using single-lap splices, where possible, stems from two key manufacturing considerations: (1) it eliminates the need to make and properly locate the splice plates and (2) it avoids the need for the precision trimming on assembly of the edges of skin panels where they would otherwise have to butt together. The joint weights are about the same as for designs with an $\ell/t = 50$, so the weight penalty for single-lap splices is not severe.

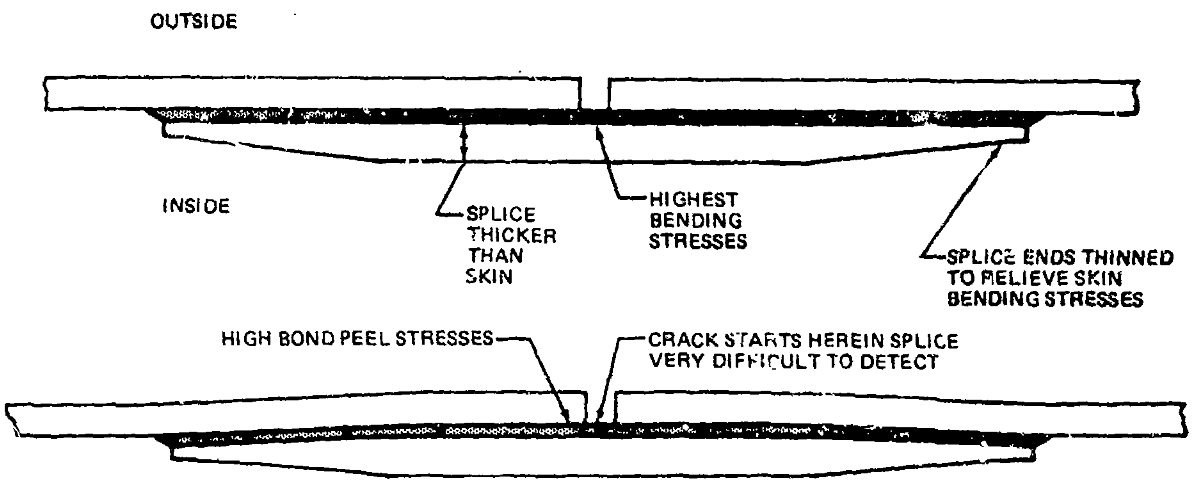
Figure 49 identifies the key characteristics of single-lap bonded joints. Only rarely is the strength of such joints limited by the shear strength of the adhesive, see Section 10.3.2.2. If the load intensity is sufficiently high and the overlap-to-eccentricity ratio sufficiently low, the adherends will yield at the plastic hinge formed in the adherends at the ends of the overlap. After this, a failure of the adhesive under combined peel and shear loads becomes inevitable, but the primary failure will be yielding of the metal. The more ductile is the adhesive, the closer the failure load corresponds with the ultimate strength of the metal rather than with its yield strength.

10.4.2 Single-Strap (Flush) Bonded Joints

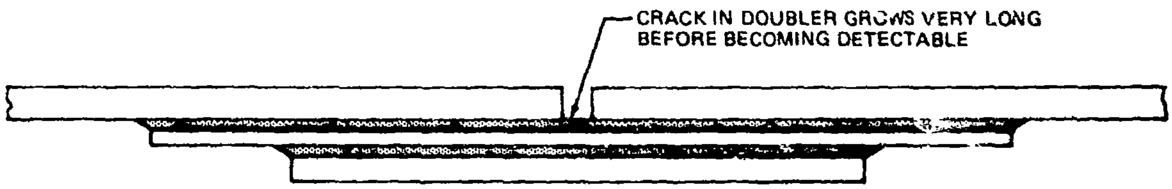
Neither double-lap nor single-lap joints should be used for circumferential fuselage splices. Aerodynamic drag requirements dictate that such splices be flush on the outside, leaving the only acceptable joint as one with one or more bonded straps on the inside.

The correct non-linear analyses of such flush joints reveal that they should not be considered as two single-lap joints mounted back-to-back. In the single-strap joint there is no length to provide any flexibility, by smooth deflections, at the middle of the splice. The skins can deflect smoothly at the outer ends of the splice, however. Therefore, the bending moment induced in the middle of the splice is inevitably more severe than the bending moments in the skins at the edges of the splice. Therefore, the splice plate must necessarily be thicker than the skins if it is to have the same strength. Increasing the overlap remains a powerful technique to minimize the bending stresses induced due to the eccentricity. Additional techniques are discussed below. The basic problem in designing these joints is that the very extra thickness needed to increase the bending strength of the splice adds to the eccentricity in load path which gave rise to the need for reinforcement in the first place. The use of an extra overlap to relieve the bending moments is, therefore, preferable. The critical location in the joint is the outside surface of the splice adjacent to the ends of the skins. This area is virtually uninspectable. A splice skin crack can grow a long way along the seam before it becomes visible on the inside, Figure 50 . If such a splice is tied to a frame on a pressurized fuselage, the tendency of the splice plate to expand more than the frame will counteract these bending stresses due to the eccentricity, as long as the fuselage is pressurized. However, pressurization moments are additive to moments caused by eccentricity in the skin at the outer edge of the splice.

The laminated-strap flush splice in Figure 50 contains both positive and negative features with respect to the tapered single-strap flush splice. The taper presents a real problem for the NDI of the adhesive but is necessary both to: (1) prevent very high peel stresses in the bond at the edges of the splice plate, and (2) minimize the bending stresses set up in the skin at the same location. The laminated strap overcomes all of these problems but introduces a new one of its own. When the crack initiates in the thin doubler and grows along the skin seam, the bond will delay the initiation of the corresponding crack in the thick splice until the first crack is quite long and still undetectable. When the second crack finally starts to propagate in the splice, it is then driven by a greater eccentricity in load path and will attain critical size more rapidly than for an unlaminated splice. The crack will reach the inside of the splice sooner for a single splice than for a laminated splice and, therefore, offers a better chance of detection and repair before the crack on the outside of the splice grows to a critical

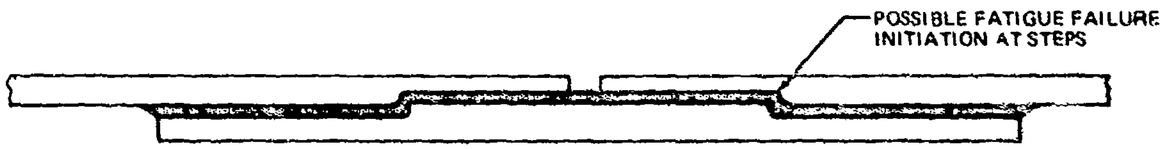


(a) DEFLECTION UNDER LOAD TO RELIEVE BENDING STRESSES

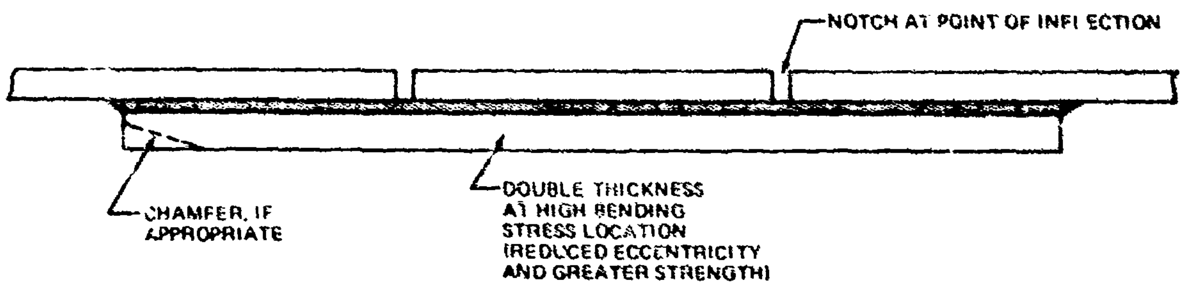


(b) LAMINATED-STRAP FLUSH SPLICE

(UNIFORM DETAILS EASIER FOR BOND INSPECTION BY NDI)



(c) TWO-STEP FLUSH JOINT HAVING REDUCED ECCENTRICITY



(d) MODIFIED FLUSH JOINT TAKING ADVANTAGE OF DISTORTION UNDER LOAD

FIGURE 50. SINGLE-STRAP (FLUSH) SPLICES

length. This problem has no parallel with riveted construction, for which the laminates deflect individually, like the leaves of a spring, without inducing high bending stresses. Currently, insufficient experimental evidence is available to choose between these two bonded splices. For test data on single-strap bonded joints, refer to Section 11.3

Figure 50 (c) and (d) show techniques which may be used to alleviate the eccentricity problems of joints (a) and (b). Of the two, the joint (d) is undoubtedly more reliable and easier to make. However, it is heavier than the other designs since it requires about twice the overlap but at only a single increase in gage above that of the skin, rather than about a 50 percent increase in thickness. The virtue of the joint in Figure 50 (d) is that analysis predicts such a dramatic reduction in bending stresses that the increase in life, with respect to the other designs shown, is proportionally far greater than the increase in splice weight. Since difficult machining is not needed, it is also the least expensive design shown.

10.4.3 Bonded Doublers

Figure 51 provides some real evidence for the concern about skin cracks induced by the load path eccentricity in one-sided bonded doublers. Skin cracks developed at the end of nearly every finger and in between some adjacent fingers as well in the design shown. The location of the cracks is exactly where a non-linear analysis predicts them to be. Figure 51 also indicates one distinct advantage of finger doublers over tapered doublers. All the cracks shown have yet to grow across the gaps between the fingers because the bending stresses in the skin are much lower there. Consequently, the cracks must grow slowly through an area in which they can be easily detected before failure can occur. With a straight-edged doubler (tapered or not), there is no such provisions and catastrophic failure from the joining up of undetected skin cracks is far more likely to occur.

Skin cracks grow more slowly when restrained by a bonded doubler than would be the case for a chem-milled panel of the same total thickness. Further information pertaining to bonded doublers is to be found in Section 10.3.3.

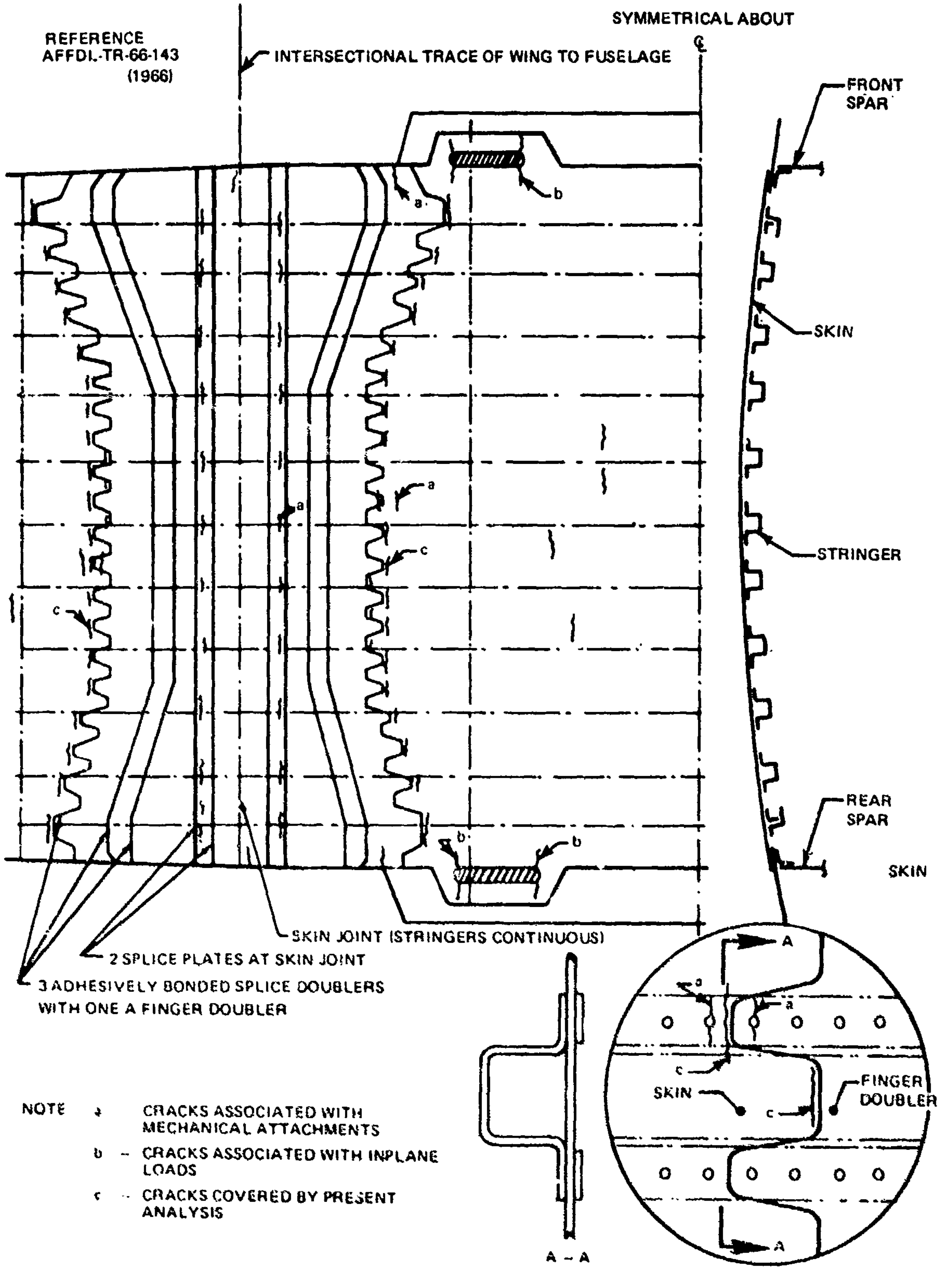
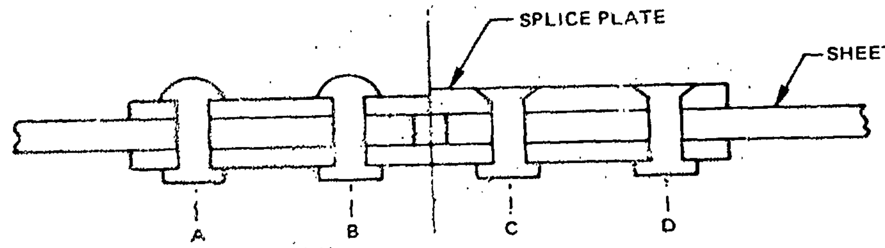


FIGURE 51 . FATIGUE CRACKS DEVELOPED DURING TESTING OF AIRCRAFT STIFFENED WING PANELS WITH BONDED DOUBLERS

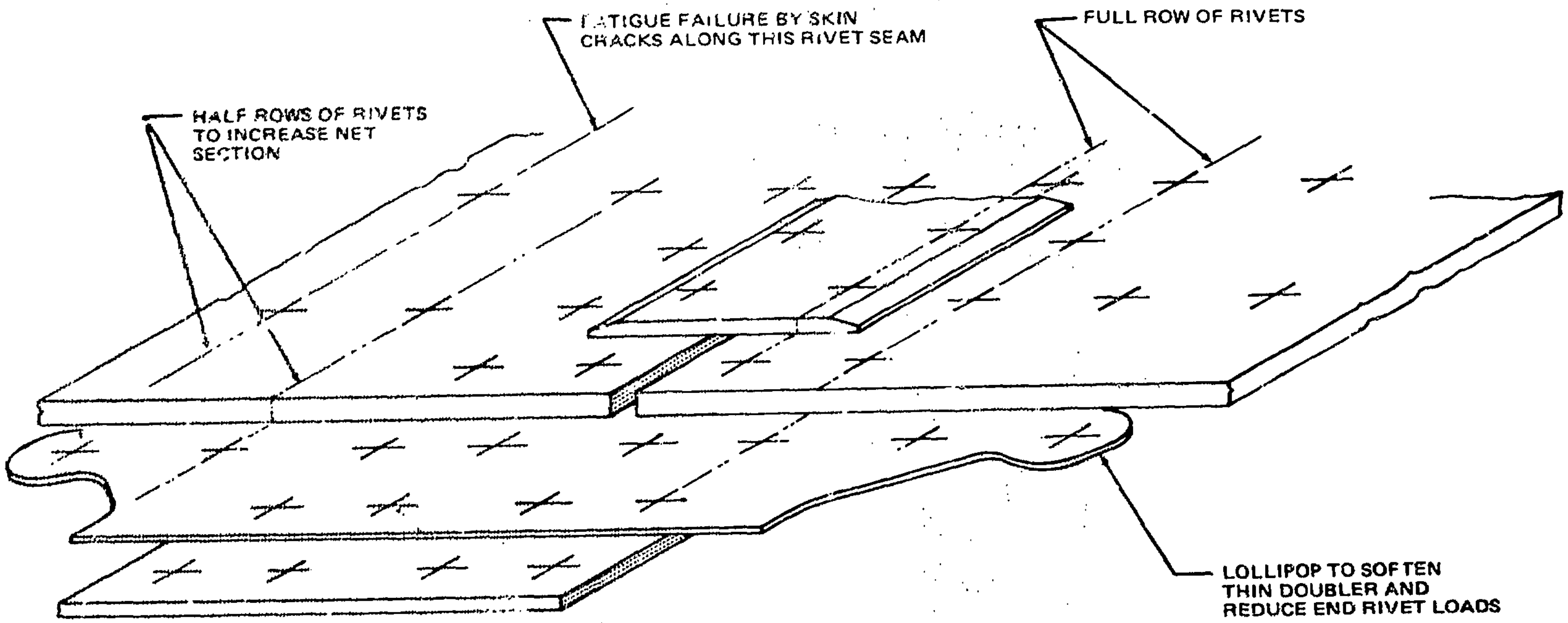
10.5 Combining Bonding with Mechanical Attachment

In order to provide total integrity in a structure it is necessary to consider the environment, corrosion resistance, acoustic fatigue, fatigue life, ultimate strength, fail safety, damage tolerance, and residual strength. There are applications where neither adhesive bonding alone nor mechanical attachments alone can provide total integrity in a structure. For instance, the use of adhesive bonding may be necessary to provide a sufficient fatigue life for a structure with adequate ultimate strength. Mechanical fasteners to provide a fail safe path may be necessary to provide damage tolerance to a bonded structure with marginal residual strengths. In addition, bonding can provide excellent corrosion resistance to a mechanically fastened structure subject to a hostile environment. Therefore, the judicious use of bonding and mechanical fasteners can provide a light-weight low cost structure with maximum structural integrity.

Prior to discussing the load sharing between the adhesive and the fasteners, it is appropriate to identify the high and low stress areas in a mechanical multi-row splice shown in Figure 52 (a). This configuration has equal total strength and stiffness in the sheet and plates, in which each rivet row transmits an equal load. The joint is equally likely to fail in the sheet at row A or in the splices at row B. The sheet at row B and the splice at row A are less critical because there is no tension load at those holes to be reacted elsewhere. If row B had flush rivets, the weak link would be in the splices there since the sheet at A would be subject to a much lower bearing stress even though the tension stresses would be identical. Countersinking the fasteners at row A does not usually add a further potential failure location (even if the fasteners at B were not countersunk) because of the absence of a tensile load in excess of the bearing load. It would be preferable to force the first evidence of fatigue failure into the splices at A from considerations of inspection, residual strength and ease of repair. Usually the splice plate is thicker, as at C and D in the upper right of Figure 52 (a), for flush fasteners. This both decreases the bearing stress in the splice and transfers some load from row C to row D as the result of stiffness imbalance between the sheet and splices. This is undesirable, since even a small increase in bearing stress at the outer row D, of fasteners reduces the fatigue life of the sheet there.



(a) SIMPLE CONFIGURATION

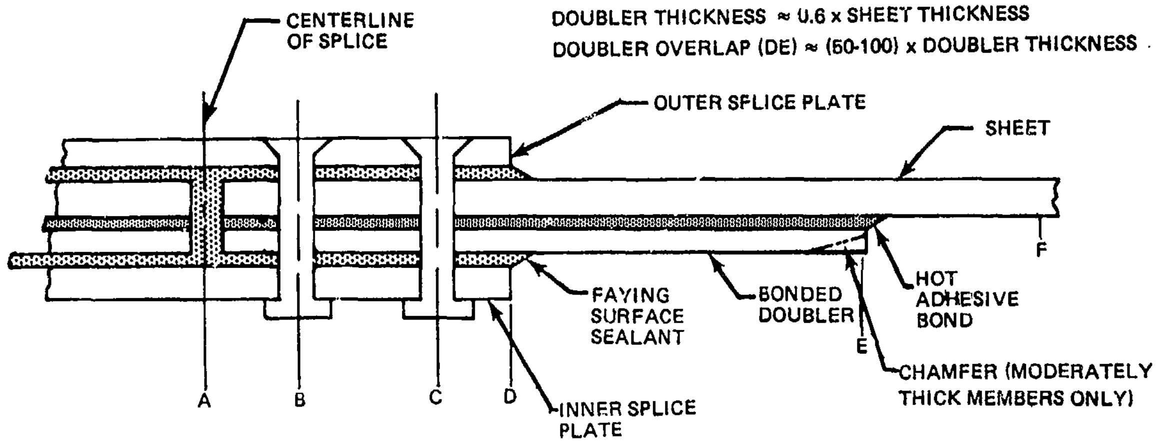


(b) REFINED CONFIGURATION

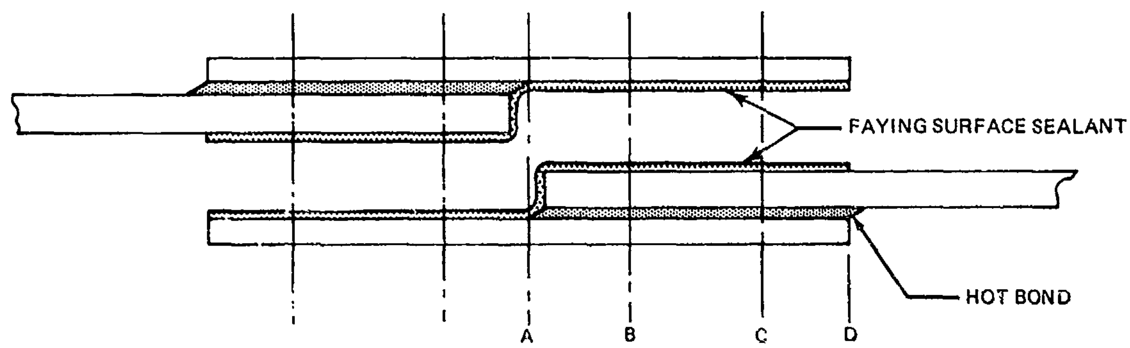
FIGURE 52 . DOUBLE-STRAP MECHANICAL SPLICE

The complexity usually added to enhance the fatigue life of the simple joint is indicated in Figure 52 (b).The tension load at the outer rows of fasteners is fixed,so the design objective is to decrease the bearing load on the end fastener.. This is particularly important when that fastener is countersunk into the sheet.

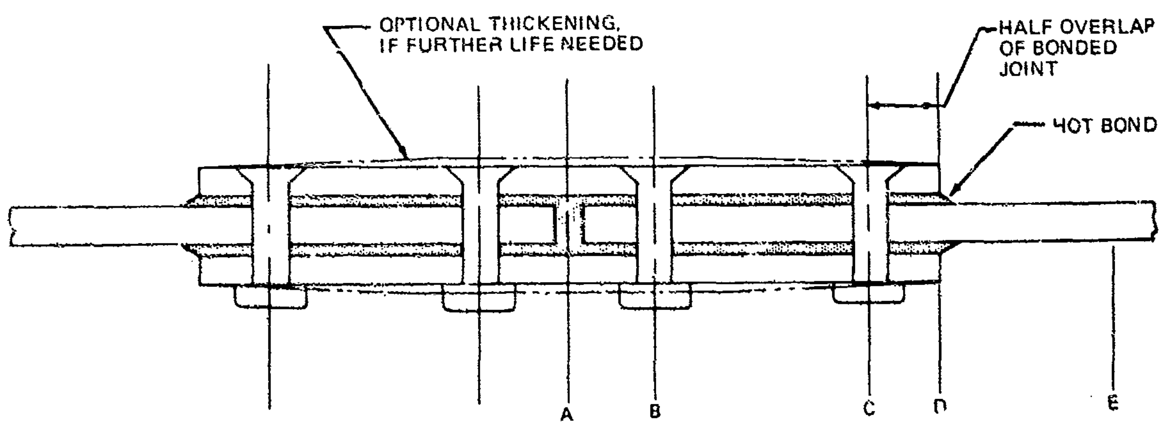
With an understanding of the basic load transfer in mechanical joints, one can discuss improving the situation by using hot bonding as well. Two basic approaches for this are described in Figure 53 . The choice between the two will usually be dictated by manufacturing considerations. The customary approach, Figure 53 (a) requires trimming the sheet/doubler combinations where the subassemblies are butted together on assembly. The key to this approach is the reduction of the sheet stress at station C with respect to that at F away from the splice. The overlap from C to E should be adequate to transfer sufficient load from the sheet to the doubler so that the stress in both is the same as at station C. Actually longer overlaps are needed to prevent introducing a bending moment in the sheet at E because of the eccentricity in load path there. The distance from the end of the splice, at station D, to the end of the doubler at E, should be at least 50 times the doubler thickness, but need be no more than 100 times that thickness unless the fit of other details demands it. The splice plate thicknesses, widths and edge distances are those used for conventional riveted construction. The outer splice plate will be thicker than the inner plate if the fasteners are countersunk. The total thickness of the splices will reach, or slightly exceed, the combined thickness of the sheet/doubler combination in order that each full row of fasteners will be equally loaded. The middle splice, Figure 53 (b) is considerably lighter than splice (a) and retains most of the benefits. The sheet is reinforced by one or the other doubler before the load reaches the outer row of fasteners, as at C. Thus, the distance CD should be longer than for a riveted-only-design, being one half of the overlap for a double-lap bonded joint between the same members. Because of the bonded splices,each acting like bonded doublers on one side of the joint, the fastener loads are roughly halved. Also, the tension stress in the sheet at E is reduced to about 2/3 of that value by the outer row of fasteners, as at C. The first potential failure locations are no longer in the sheet, but occur in one splice



(a) TRIM-ON-ASSEMBLY APPROACH



(b) TRIM-ON-ASSEMBLY APPROACH



(C) PREFIT AND TAKE-UP TOLERANCES ELSEWHERE, APPROACH

FIGURE 53 . DOUBLE-STRAP BONDED-RIVETED SPLICES

plate or the other at the row of holes adjacent to where the sheets butt gogether. The splices could be thickened at their center as shown, to compensate for this, if necessary. The joint in Figure 53 (c) is probably preferable from the viewpoint of structural efficiency, but poses potential fit-up problems and needs greater care in handling because of the thinner edges. Figure 54 shows how the sheet stress is reduced by the bond between the edge of the doubler and the first row of rivets. Indeed, there would be practically no load in the rivet unless the bond had been destroyed by environmental attack or by yielding of the metal. This characteristic of the bond being stiffer than a rivet extends also to other rivet-bonded joint configurations such as the single-lap joints discussed later.

The typical rivet-bonded joints shown in Figure 55 identify qualitatively some of the guidelines for rivet-bonded single-lap splices. In using such information, one should seek the least complex joint having adequate strength and life, consistent with applicable manufacturing considerations. With reference to the lower figure in Figure 55 and remembering the preceding assessment of mechanical joints, the critical element to be protected is one sheet or the other at the first (outer) rows of fasteners, A and D. If there were no outer doubler, the first failure location would be at the countersunk rivets in the outer sheet in A. The next potential failure would be the inner skin at D. Therefore, if an outer doubler were installed with faying surface sealant only, instead of the hot adhesive bond, the elimination of the countersink and high bearing stress in the outer sheet at A would leave that area no worse than the inner skin at D. Therefore, this arrangement can develop significant fatigue lives even without the hot bond. If, after hot bonding the outer doubler, the inner sheet does not develop an adequate life and cracks at D, the stiffener could be moved, as shown, to provide a hot bonded reinforcement at that location and improve the life. For additional information, see Reference 2 . Figures 33, 34 and 35 , explain why it is hard to transfer more of the load through the inner rows of fasteners (B and C) to unload the outer rows (A and D). It is important to understand that final failure is preceded by small crack growth at some or all of the fasteners. Therefore, using smaller fasteners at the same pitch as the inner rows does not provide a reasonable opportunity to detect such cracks prior to failure. Therefore,

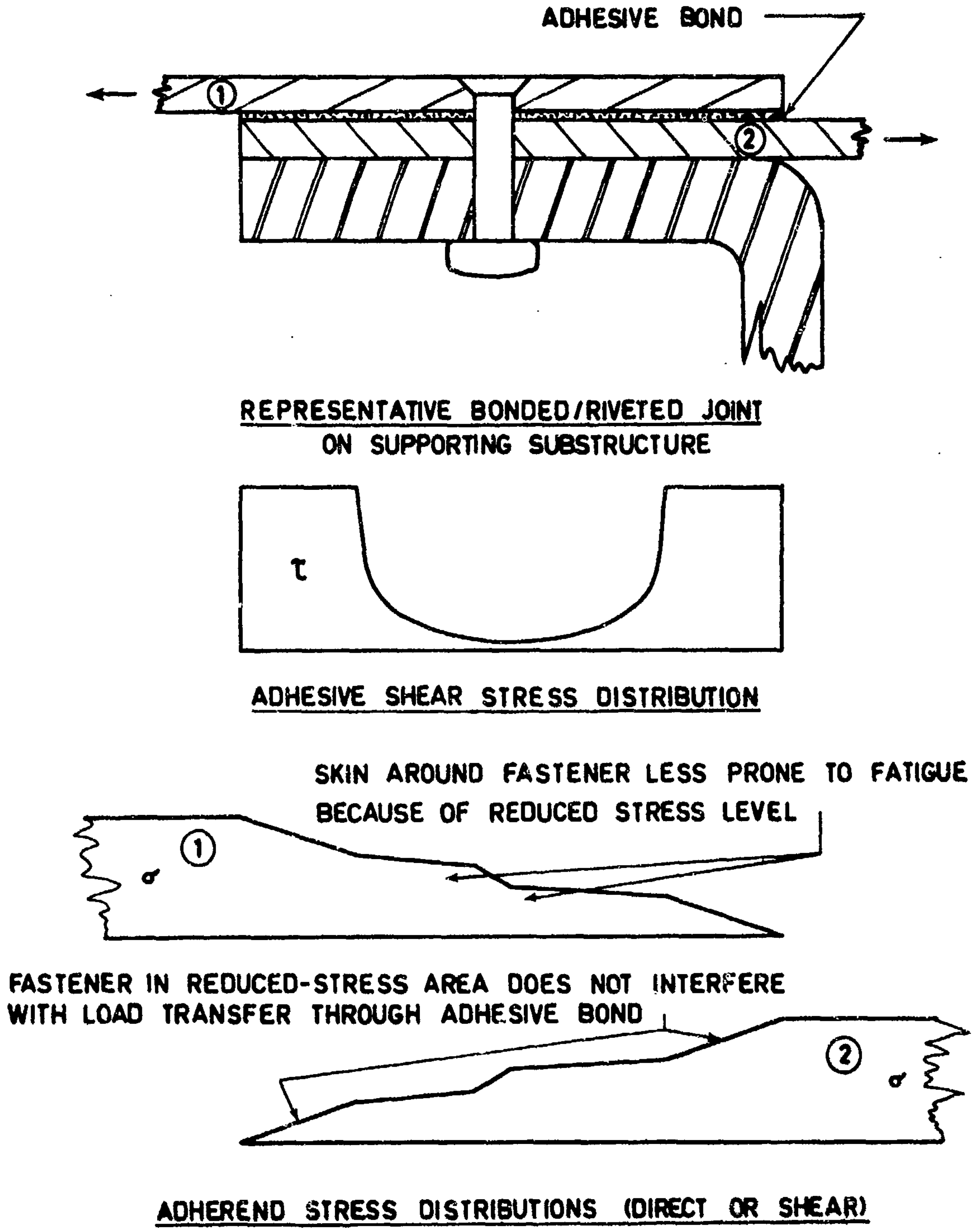
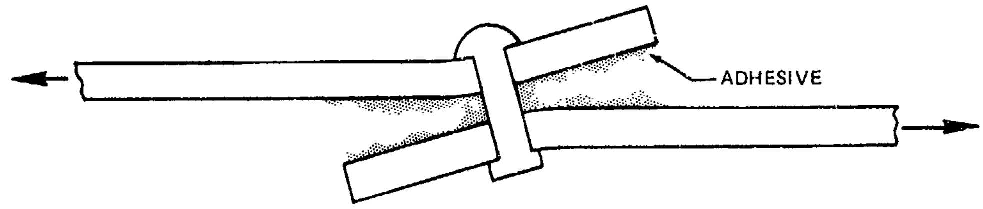
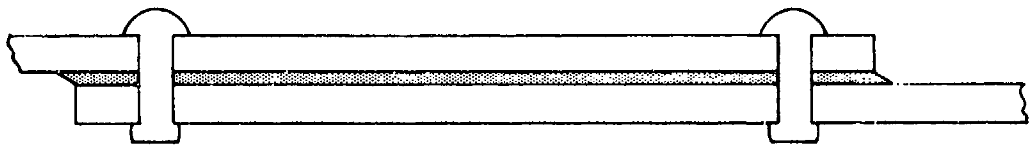


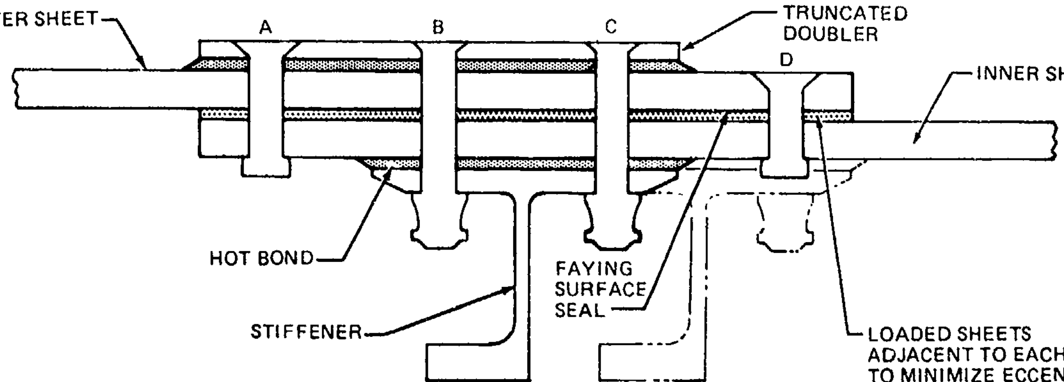
FIGURE 54 . EXPLANATION OF LOAD TRANSFER IN RIVET-BONDED CONSTRUCTION



DON'T – HIGH BENDING MOMENT, OVERLAP TOO SHORT, BOND BROKEN BY PEEL LOADS.



DO – LONG OVERLAP (50 TO 100t) RELIEVES BENDING MOMENT DUE TO ECCENTRICITY. RIVETS IN REDUCED STRESS AREA BECAUSE OF HOT BOND ADHESIVE. VERY LITTLE RELATIVE MOTION IN CENTER OF OVERLAP SO FASTENERS COULD NOT PICK UP LOAD THERE. RIVETS PROVIDE FAIL-SAFE LOAD PATH AND DAMAGE TOLERANCE IF BOND FAILED LOCALLY BY YIELDING OF METAL.



DO – LONG OVERLAP TO ALLEVIATE ECCENTRICITY.
STIFFENER TO STABILIZE SPLICE AND PREVENT WRINKLING.
BONDED EXTERNAL DOUBLER TO REDUCE OUTER SHEET STRESS AND AVOID COUNTERSINKING SHEET AT A.
HALF-ROWS OF FASTENERS AT A AND D DO AS TO LEAVE LOAD FOR FASTENERS AT B AND C AND MAXIMIZE NET SECTION AND PERMISSIBLE CRACK LENGTH AT A AND D.

DON'T – EXTEND HOT BONDED EXTERNAL DOUBLER TO ROW D SINCE THIS WILL INCREASE RIVET LOADS AT ROW D AND LEAD TO EARLIER FAILURE OF INNER SHEET THERE.

OPTIONAL – LOLLIPOP INNER SHEET AT A, OR DRILL WITNESS HOLES THERE IN INNER SHEET ONLY BOTH TO UNLOAD FASTENERS AT A TO MAKE CRACKS IN OUTER SKIN LESS LIKELY AND TO IMPROVE INSPECTABILITY. IF CRACK DEVELOPS AT A, IT WILL BE IN OUTER SHEET, NOT INNER.
OMIT FASTENER ROW D, FOR BASIC STIFFENER LOCATION, SO THAT ROW C BECOMES FIRST ROW WITH INNER SHEET REINFORCED BY HOT-BONDED STIFFENER.
RELOCATE STIFFENER AS SHOWN IF NECESSARY TO AVOID INNER SHEET FAILURES AT ROW D.

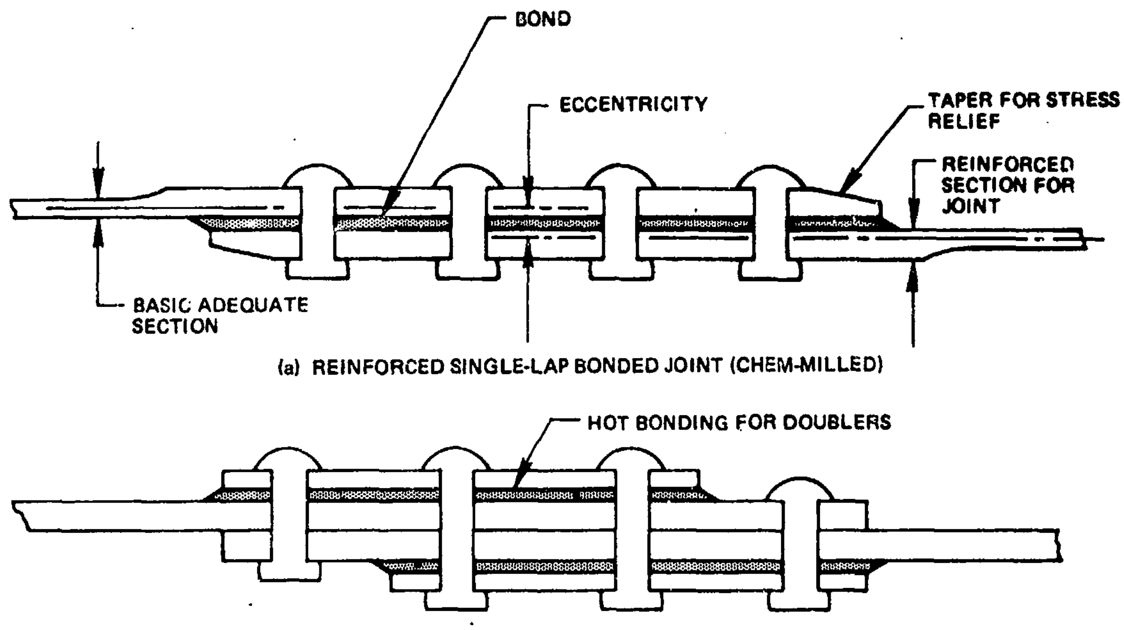
FIGURE 55. GUIDELINES FOR RIVET BONDING

the outer rows usually omit every second fastener. Consequently, the fasteners in the outer rows must be soft so as not to attract extra load. Longitudinal skin splices of the type shown at the bottom of Figure 55 were developed for and successfully tested during the PABST program.

Figure 56 contains further information pertinent to raising the structural efficiency of single-lap rivet-bonded joints. Figure 56 (a) explains how to minimize the eccentricity by suitably placing any local reinforcements to decrease the stresses in the joint area. Figure 56 (b) shows a further refinement to transfer the load more evenly between the fasteners. By deliberately omitting any reinforcement for the "last" row of rivets, the skin is softened there, permitting it to stretch more easily, so that load is shed from the outer rivets and transferred to the inner pair. The consequent decreased bearing stresses at the "first" row of rivets thus increases the fatigue life of the skins.

Longitudinal skin splices can use single-lap joints or double-strap joints because the deviations from a smooth loft surface run longitudinally and do not cause too much of a drag penalty. However, circumferential splices must be flush, requiring the use of a single-strap joint. Such a joint is inherently weaker than either of the joints mentioned above and is, therefore, not usually used for highly-stressed longitudinal splices.

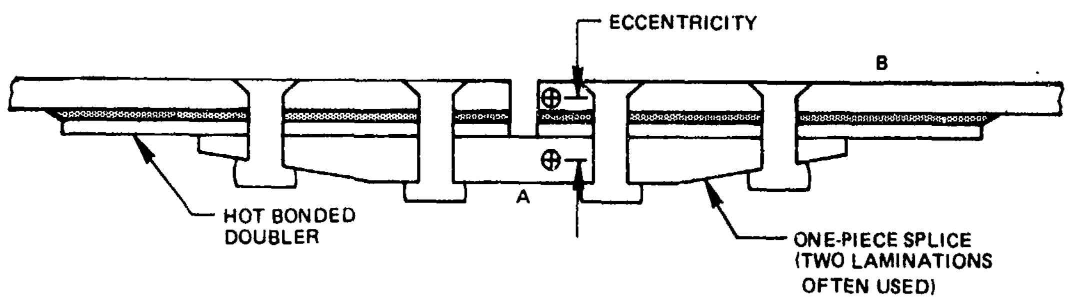
The single-strap splice has an inherent load path eccentricity and the thicker the reinforcement, the greater is the eccentricity in load path necessitating the reinforcement. The other joints have no similar problem. The key difference between a single-lap and single-strap joint is that the latter has no long length of splice in the middle to deflect smoothly and alleviate the bending stresses. The single-lap joint has that capability at both ends. As shown in Figure 57, the potential weak link is the bending stresses induced in the splice plate under the sheet ends. To improve upon a basic mechanical splice, a hot bonded doubler would be made continuous to reinforce the splice, Figure 57 (a). Also, an extra row of rivets would be added, joining the sheet and thin doubler only, to reduce the load on the outermost row of rivets. The configuration of Figure 57 (b) both minimizes the eccentricity in the



(a) REINFORCED SINGLE-LAP BONDED JOINT (CHEM-MILLED)

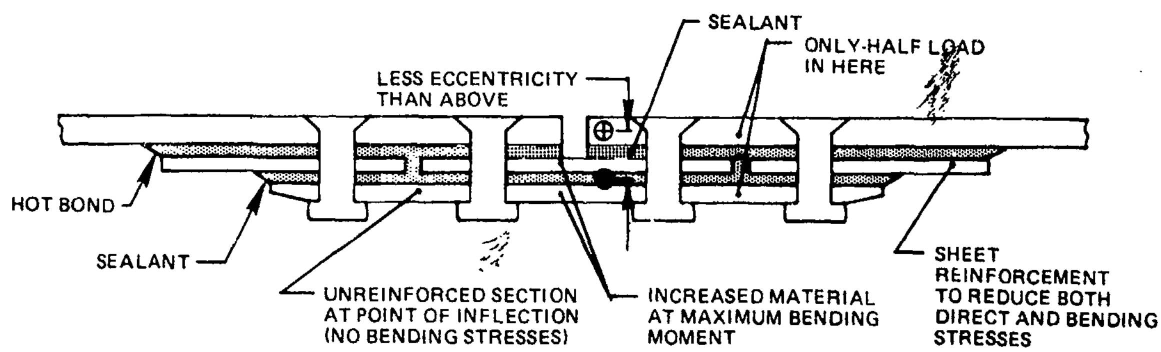
(b) BONDED REINFORCEMENT TO MINIMIZE ECCENTRICITY IN LOAD PATH

FIGURE 56 . MINIMIZATION OF ECCENTRICITY IN SINGLE-LAP RIVET/BONDED SPLICES



(HIGH BENDING MOMENTS AT A AND B CAN BE RELIEVED ONLY BY LONG OVERLAPS)

(a) REINFORCED SINGLE-STRAP RIVETED/BONDED JOINT, SUFFERING FROM GREAT ECCENTRICITY IN LOAD PATH.



(b) REINFORCED SINGLE-STRAP RIVETED/BONDED JOINT, WITH MINIMIZATION OF ECCENTRICITY EFFECTS.

FIGURE 57. FLUSH (SINGLE-STRAP) RIVET/BONDED SPLICES

load path and reinforces both the sheet and splice where the stresses are highest. All countersinks are in a low stress area, either because of hot bonded reinforcement or because half of the load has already been transferred elsewhere.

An important point about such flush joints is that, if circumferential splices are stabilized by attachment to a frame, the hoop stress is reduced locally to only about half of that in the skin away from the frames. Furthermore, the bulging of the skin under pressure between the frames tends to nullify some of the eccentricity in the axial load path at the splice. Therefore, such a joint should perform somewhat better in service than would be indicated by flat coupon tests.

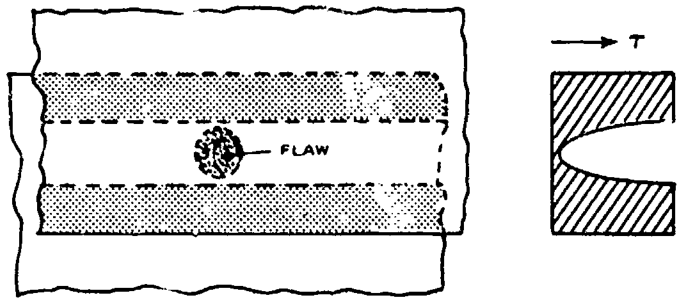
10.6 Cracking and Damage Failure Modes

10.6.1 Effects of Adhesive Flaws, Damage, and Variable Thickness Bondlines

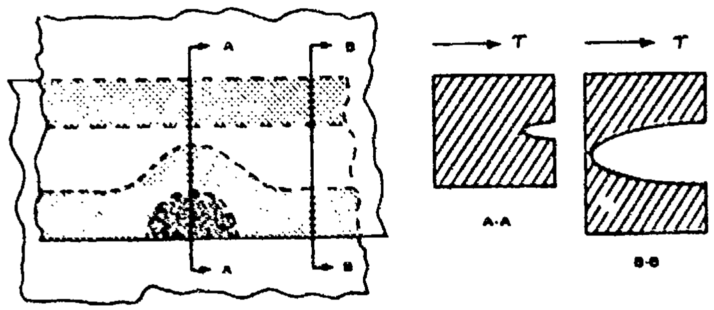
Imperfectly bonded structures can be separated into two categories. The first has flaws that are sufficiently small, or located in relatively insensitive areas, so as not to induce a load redistribution in the structure outside the bond area. The second has flaws large enough to cause a basic redistribution of loads beyond their immediate vicinity which, in turn, can lead to a drastic restriction in remote load levels if the initially localized defect is not to propagate rapidly. Analysis methods for each of these types of flaws are currently in preparation. The significance of such flaws must be assessed in conjunction with their influence on the metal as well as on the bond, and in terms of the remote metal stresses. Consideration of damage tolerance of the metal can sometimes over-ride the effects of flaws in the adhesive.

Figure 58 explains two circumstances for which small bond flaws can be ineffective. In Figure 58 (a), the flaw is located in an area which will never have a high shear stress even with an intact bond. Such a flaw should be ignored or, at most, inspected periodically to see if any growth occurs. Any attempt to repair such a flaw could permit moisture to enter the flaw and start corrosion. The flaw in Figure 58 (b) is small enough to merely transfer a small increment of load into a previously lightly loaded elastic trough. Such flaws should at least be sealed to prevent water from penetrating the open crack and growing the crack from a freeze-thaw cycle. A sealant should be used in preference to a cold set adhesive.

Significant bond flaws act like cracks in sheet metal, Figure 59 . Small bond flaws merely redistribute the load within the bond without affecting the stress distribution outside the joint but a large bond flaw causes a large redistribution of stresses far beyond its immediate location. Figure 59 shows that, at same load level, the adhesive will become loaded to its maximum capability next to the flaw. At that same load level, the bond stresses far away from the flaw will not be as highly loaded as those adjacent to the flaw. The reason is that the intact bond near the flaw must carry both the applied load and the load diverted around the flaw. Consequently,



(a) INEFFECTUAL FLAW IN LOW-STRESS AREA



(b) REDISTRIBUTION OF LOAD TRANSFER AROUND SUB-CRITICAL FLAW

FIGURE 58. SMALL DEFECTS IN BONDED JOINTS

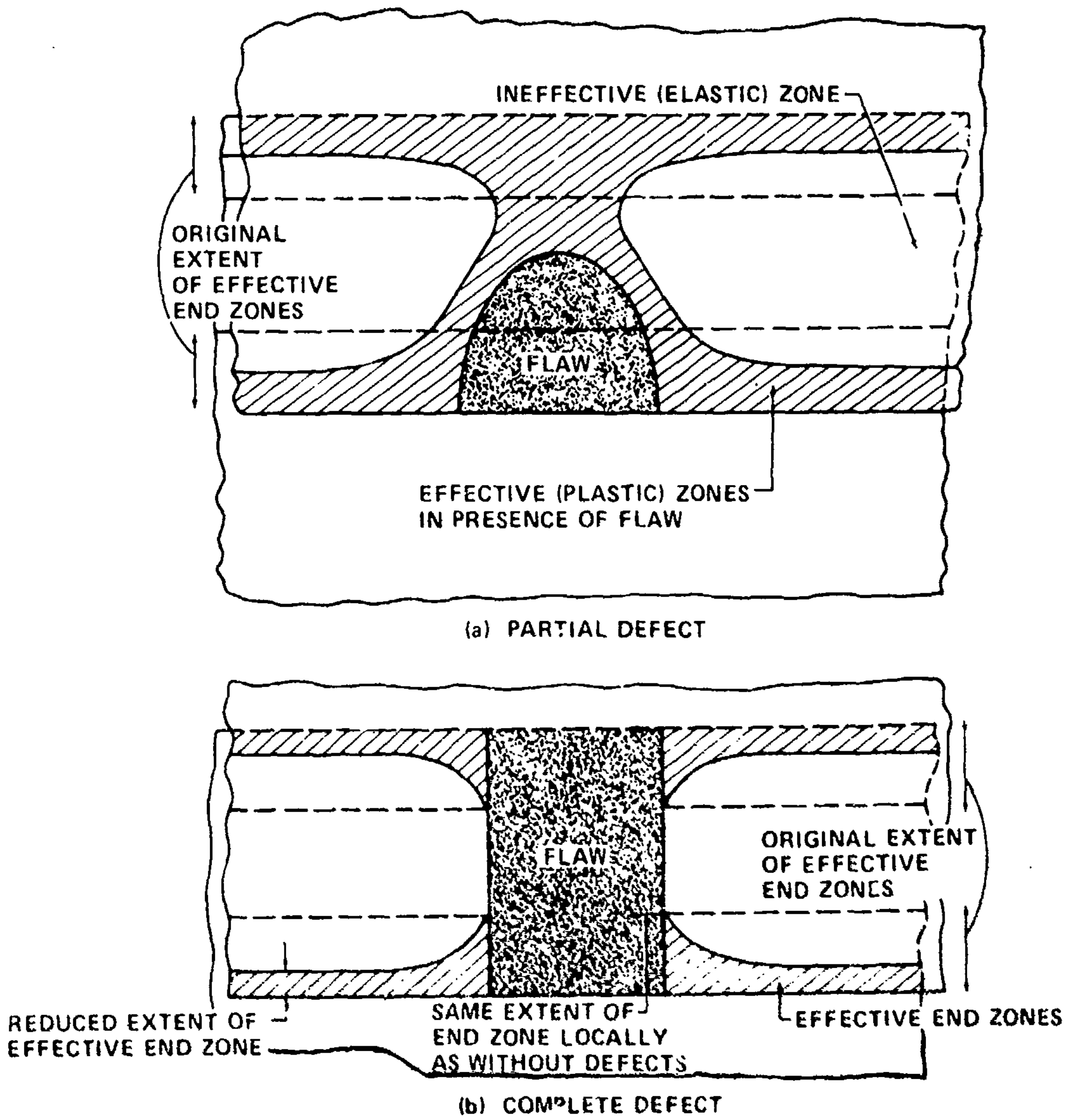


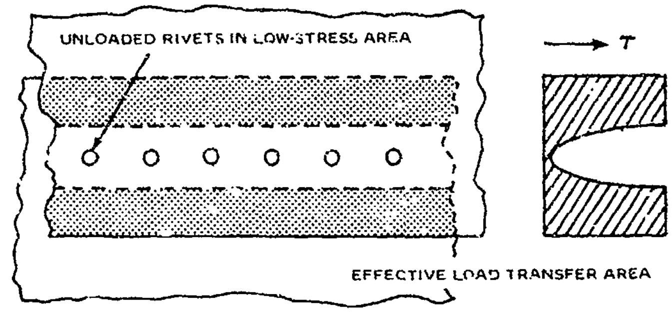
FIGURE 59 . MINOR AND MAJOR FLAWS IN ADHESIVE-BONDED JOINTS

the average load level in the sheet must be restricted to eliminate bond flaw propagation. Considerations of metal damage tolerance appear to over-ride this condition for transport aircraft fuselages, based on the PABST experience. Larger flaws require lower metal stresses if the bonded joint is not to fast fracture and fail catastrophically. This behavior bears a direct analogy to cracking metal sheet loaded by tension perpendicular to the crack. Since the load in the metal is proportional to the load in the bond at the same point, a sub-critical disbond will probably induce fatigue cracks in the metal rather than fail the bond since the intact bond is often stronger than the metal.

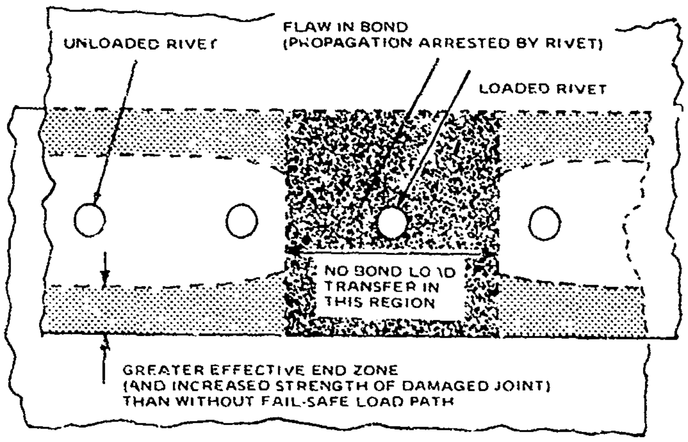
Alternatively, for a one-time-only overload of some highly stressed component, the metal may yield at the edges of the disbond and, if the load is maintained, total disbonding could occur. In short, once a bond flaw becomes large enough to be significant, its effects extend far beyond its immediate vicinity.

It may be desireable in the more highly loaded (stress) joints to provide some alternative fail-safe load paths to prevent some bond flaws from propagating instantaneously over the entire length of the panel. One solution, shown in Figure 60, is to install a seam of rivets or fasteners capable of sustaining limit load in the middle of the overlap, where the metal stress is only half of that outside the joint. As long as the bond is intact, such fasteners neither accept load or induce cracks in metal. If the bond were to fail locally, however, those rivets would pick up the load which would otherwise have been transferred around the edge of the disbond causing it to spread. Concerning the loading on the rivet holes, pillowing and skin bending stresses are induced by the pinching-in at each frame, Figure 61. Therefore, the longitudinal load is more critical along the frame than it is along the longerons. Likewise, the hoop load is more severe at a longitudinal splice than around a frame because the frame locally restricts the hoop stress in the skin. Therefore, the use of rivets as a fail safe load path in bonded skin splices would not degrade the strength or life with respect to pure bonded tensile test coupons. It should be noted that, for the environmentally resistant adhesive/primer/surface treatment system used on PABST, there was no indication of needing such a fail-safe load path.

A related two-dimensional bond problem is that of fatigue from variable thickness bondlines. When two sheets are bonded together by parallel thick (soft) and thin (stiff) bonds which vary in thickness throughout the joint, both the adhesive and metal stresses vary along the splice. Since the bond strength



(a) SOUND JOINT WITH NO FLAWS



(b) CRACK-ARREST OF DEFECTIVE BOND BY FAIL-SAFE LOAD PATH (RIVET OR SPOT-WELD)

FIGURE 60. DAMAGE CONFINEMENT BY RIVETS THROUGH ADHESIVE-BONDED JOINTS

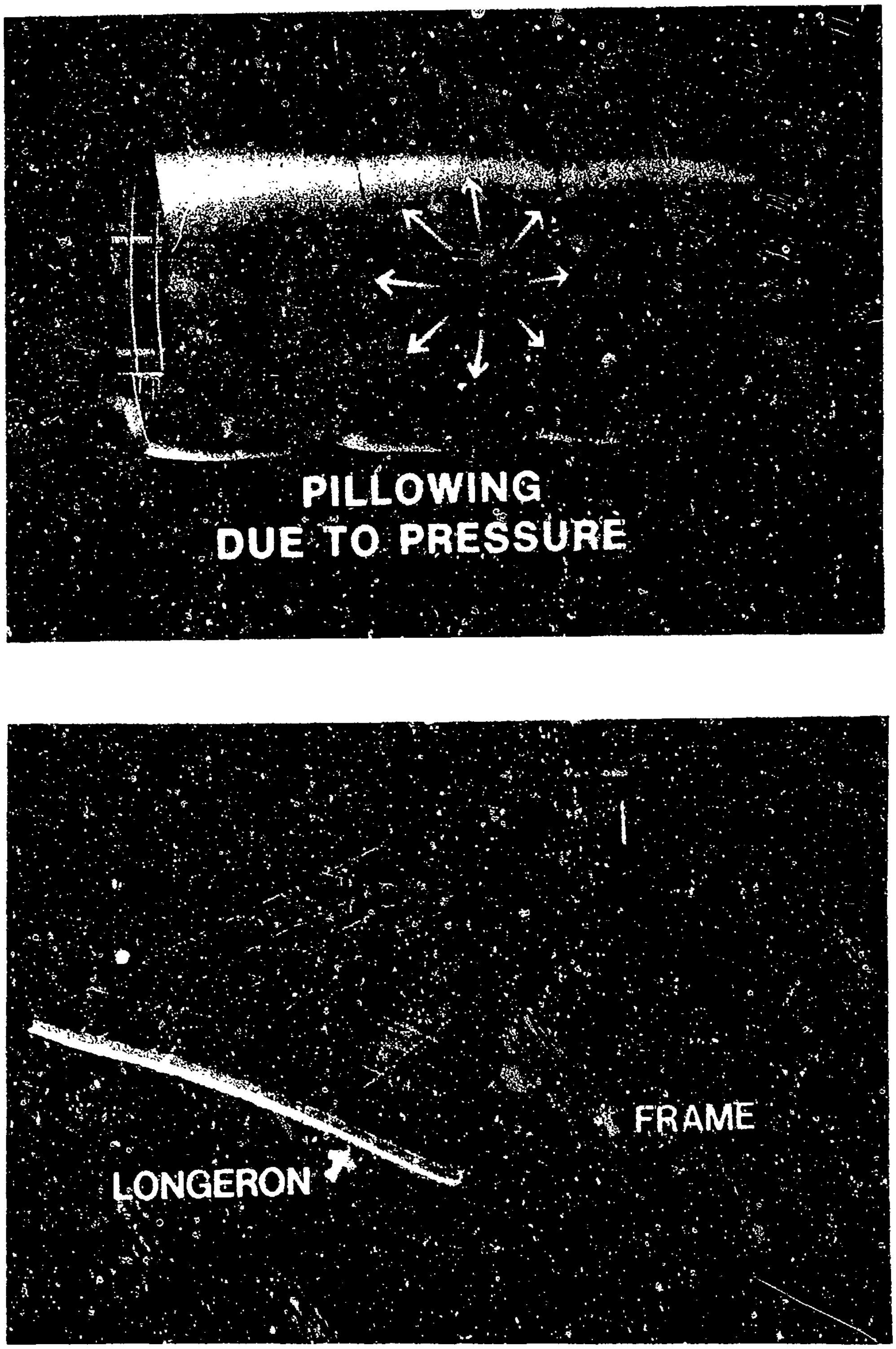


FIGURE 61 . EFFECTS OF PRESSURE PILLOWING IN A STIFFENED CYLINDER

is proportional to the square root of the bond thickness, the thinner bonds are weaker than the thicker bonds. The net result of variable thickness bondlines is, therefore, that the majority of the load is attracted to the weakest, thinnest bond areas while the load is shed from the strongest, thickest bonds. As a consequence of this local variation in intensity of load transfer, the stresses in the metal adjacent to the bond overlap vary likewise. The variable thickness bond may induce a premature fatigue failure in the metal because of the stress concentrations it causes. A failure in the metal, rather than in the bond, is not sufficient evidence of an adequate bond. The quality of the bond should, therefore, be judged on the basis of a comparison of the cycles to failure for the bonded joint and for an identical part without a joint to ensure an adequate part. There are at least two approaches which can be used to control the variable thickness bondline problem; neither approach is sufficient by itself. One is the selection of a low flow adhesive. The other is associated with the application of pressure for bonding such that the pinch off is minimized.

The experimental evidence with regard to bondline flaws and variable thickness bondlines is too sparse to draw many conclusions. Testing has confirmed that flaws do not grow in the lightly loaded areas of adhesive bonds. Furthermore, it is exceedingly difficult to make a disbond propagate under shear loading alone. A substantial peel stress is necessary to promote disbond growth. Efforts to induce bond flaw growth in test specimens of realistic thicknesses have almost invariably led to a metal fatigue failure. The only consistent exception is the single-strap (flush) joint. This joint has high enough peel stresses, where the skins butt together, to cause bond failure even for perfect bonds if the overlap is not made long enough. Therefore, artifically thick test specimens have been used to prevent a metal fatigue failure from masking the affect of a flaw on the adhesive. This has led to disproportionately high peel stresses in conjunction with the applied shear stresses. Whereas variable thickness bonds occur in normal manufacture, all efforts so far to deliberately produce a test specimen with a controlled variable - thickness bond have failed.

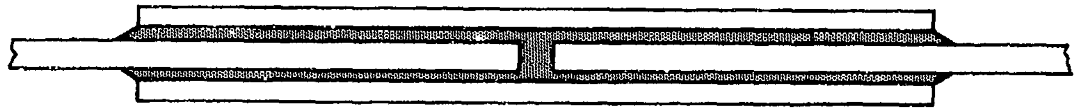
In summary, the structural details needed to promote small bond peel stresses; i.e., flexible parts, automatically lead to good quality bonds in all required areas provided that: (1) the adhesive does not flow out and (2) the parts are held together properly during cure.

10.6.2 Fail Safety in Double-Lap Bonded Joints

Double-lap bonded joints possess a limited amount of fail safety as the result of having two distinct layers of adhesive. A crack or disbond progressing in one layer is retarded by the other layer, in terms of shear load. However, once a substantial disbond has developed on one side, the remaining side represents a sharply eccentric splice with very high peel stresses where the sheets butt together. Furthermore, the remaining splice plate, which is subjected to both twice the normal load and to a very high bending moment, cannot be made strong enough to carry the load alone. If the splice plates are each made thick enough to carry the load alone when the other plate becomes detached, the thickening of each splice plate will unbalance the joint and almost halve the bond shear strength. In addition, a considerably greater overlap would be needed so as not to fail the bond in peel. These issues are explained in Figure 62.

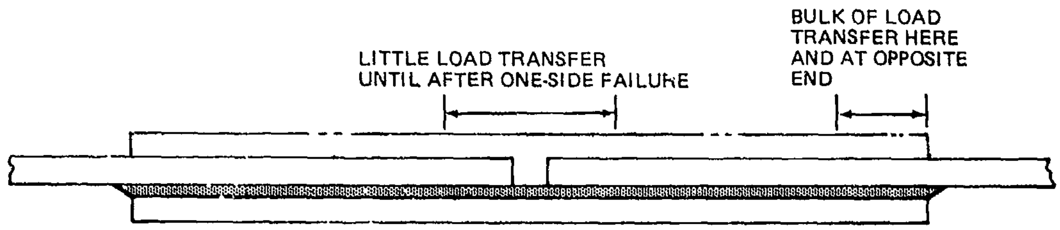
10.6.3 Adhesive Bond Stresses at Discontinuities and Cracks in the Metal

In well-designed adhesively bonded structures, the weak link is usually located in the metal rather than in the adhesive. The one classical exception is the environmental degradation of the bond-to-adherend interface associated with inadequate or improper surface preparation and moisture-sensitive adhesives and primers. This problem can be eliminated by using environmentally resistant adhesives and primers, such as FM73 and BR127. Any potential stress concentrations in the bonds for intact structures can be alleviated by careful detail design employing such techniques as tapering, fingered doublers, and additional bond area, where appropriate. However, bonded structures may be damaged or broken in service and there is no way to relieve these potential bond stress concentrations because the precise location of the structural damage cannot be predicted. Therefore, the design must permit metal elements to be broken at any location, without causing the adhesive to disbond catastrophically under fail-safe load levels. Consideration of fail-safety tends to drive



NORMAL DOUBLE-STRAP BONDED JOINT

RESIDUAL STRENGTH IF ONE SIDE FAILS



MODIFICATIONS NECESSARY TO CREATE RELIABLE FAIL-SAFE LOAD PATH THROUGH UNFAILED BOND

- INCREASED OVERLAP TO MAKE KINK LESS ABRUPT
- INCREASED THICKNESS TO ABSORB GREATER MEMBRANE AND BENDING STRESSES
- DECREASED BOND SHEAR STRENGTH

FIGURE 62. POOR FAIL-SAFE CHARACTERISTICS OF DOUBLE-STRAP JOINTS

the configuration to one of many elements which are individually small enough so that no single element failure can ever overload the bond.

Figure 63 shows the geometry and the adhesive shear stress distribution associated with one discontinuous member bonded to a continuous one. The peak adhesive stresses and strains occur immediately adjacent to the discontinuity and are a strong function of the ratio of cut area to bond width, as well as of the adhesive properties. In the simple one-dimensional case, once the load has reached a high enough intensity to start failing the bond, the adhesive will disbond instantaneously along the entire length.

In real structure, such as shown in Figure 64, the disbond may, and often does, self-arrest. This occurs because as the flaw grows, a progressively greater length of highly loaded skin between the intact bond stretches more and more. This relieves the bond shear strain at the crack tip. This benefit can occur only when the intact member is wide, with the strain level restricted far away from the crack and disbond. On the other hand, the configuration shown in Figure 65 does not exhibit any tendency for disbond arrest. Once it starts, the adhesive will disbond instantaneously along the length of the stiffener whether the stiffener yields or not. This is the only damaged structure geometry to demonstrate complete disbonding without prior warning. In every other case, as in Figure 10.42 or the corresponding cases in Figures 66 and 67 where a finite skin crack is held shut by bonded stiffener(s), an increase in applied load is necessary to propagate the disbond after it has self-arrested.

References 5 and 9 contain several quantitative examples of the influence of individual parameters on the residual strength of the damaged bonded structure. Each such curve has the same qualitative form as shown in Figure 68 . As the disbond grows, it eventually becomes so long that the partially disbonded stiffener is no longer able to hold the crack tip shut and the sheet then fast fractures and the structure fails.

In aircraft service, a crack in one member will probably induce a fatigue crack in another member before gross disbonding occurs. Referring to

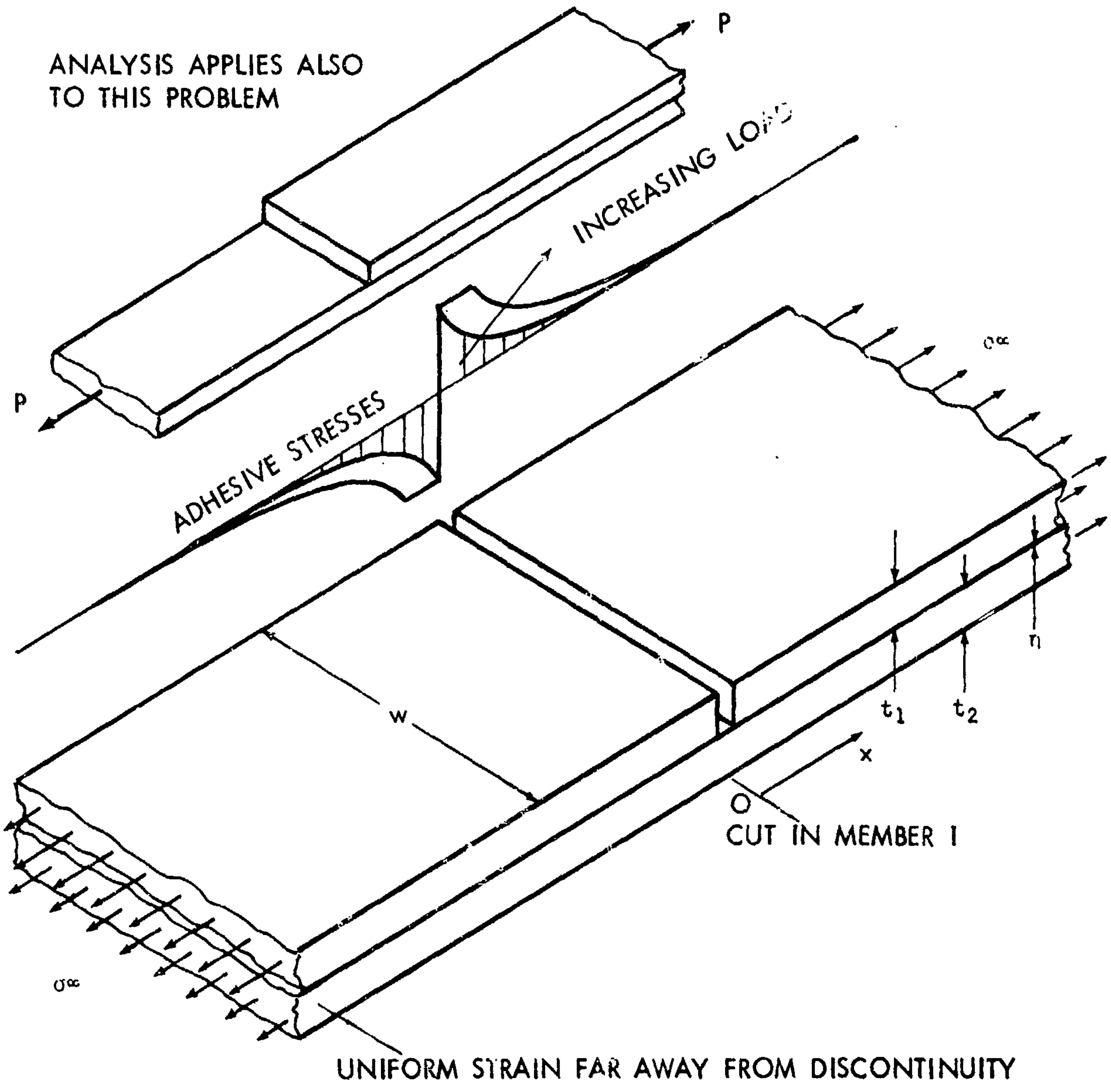


FIGURE 63. GEOMETRY AND NOMENCLATURE FOR BOND STRESSES AT STRUCTURAL DISCONTINUITY

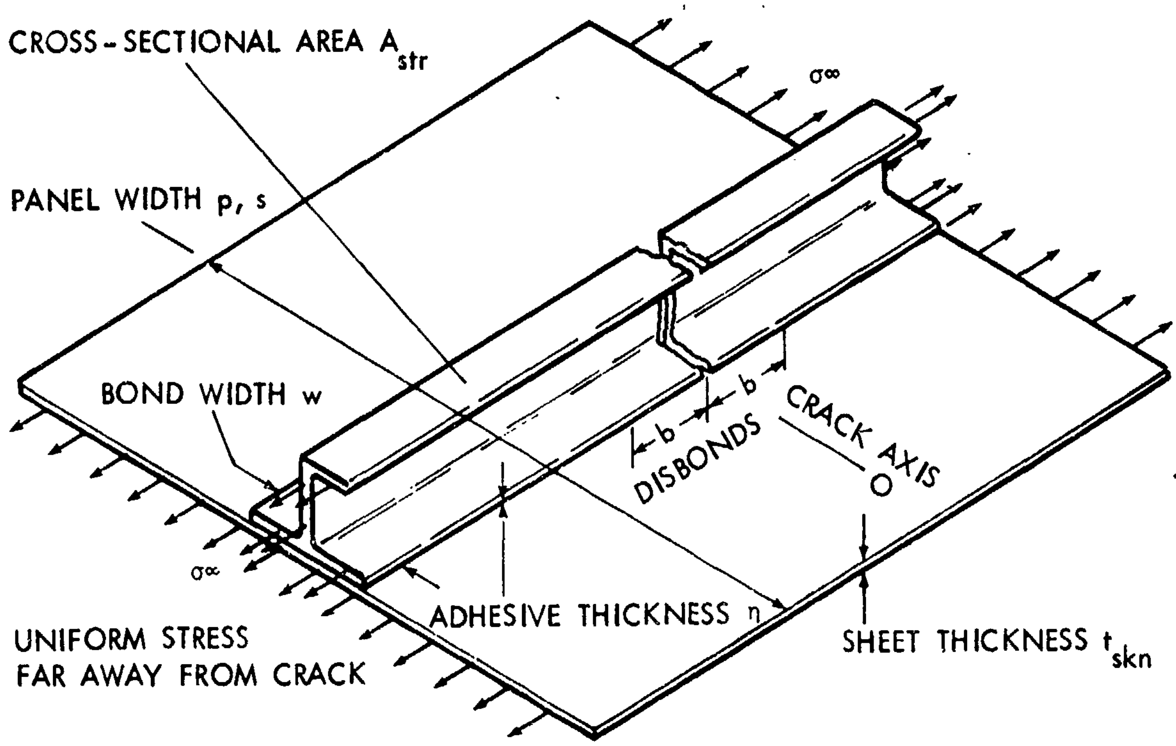


FIGURE 64. SHEET BONDED TO CRACKED STIFFENER

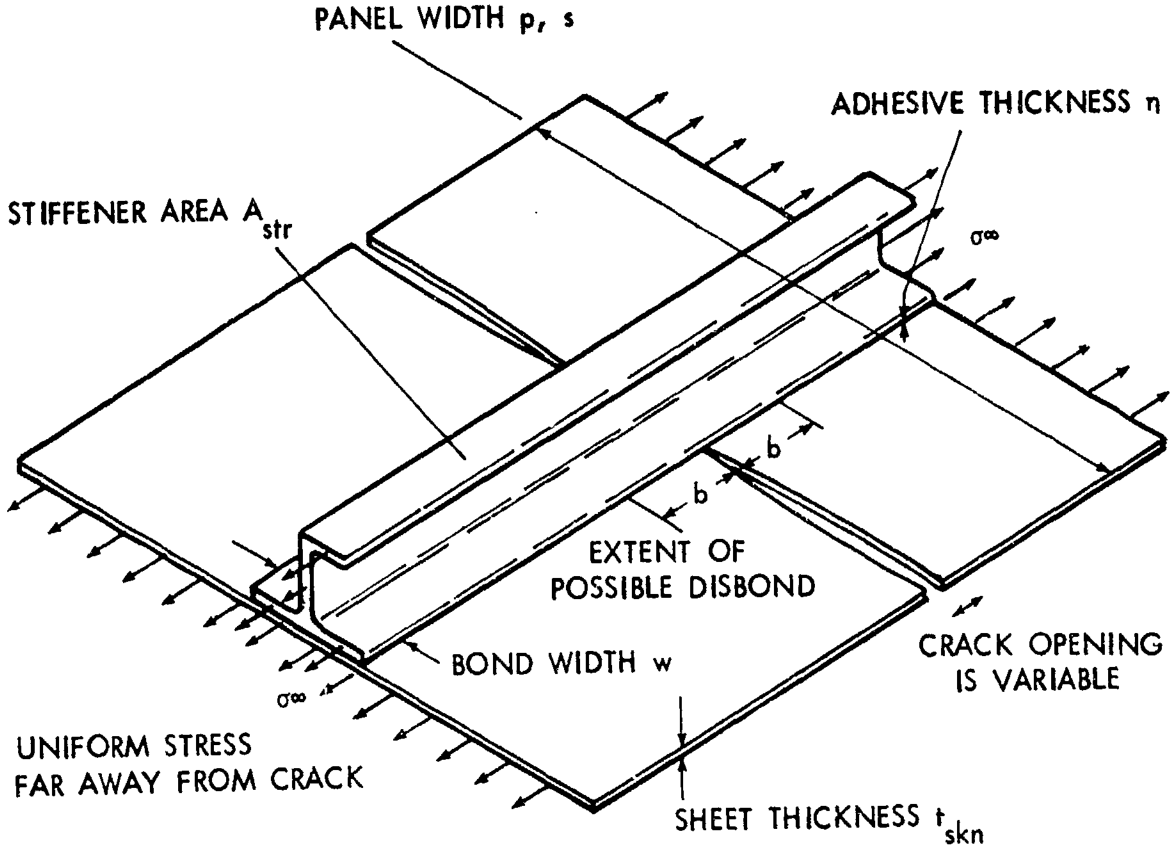


FIGURE 65 . STIFFENER BONDED TO FULLY CRACKED SHEET

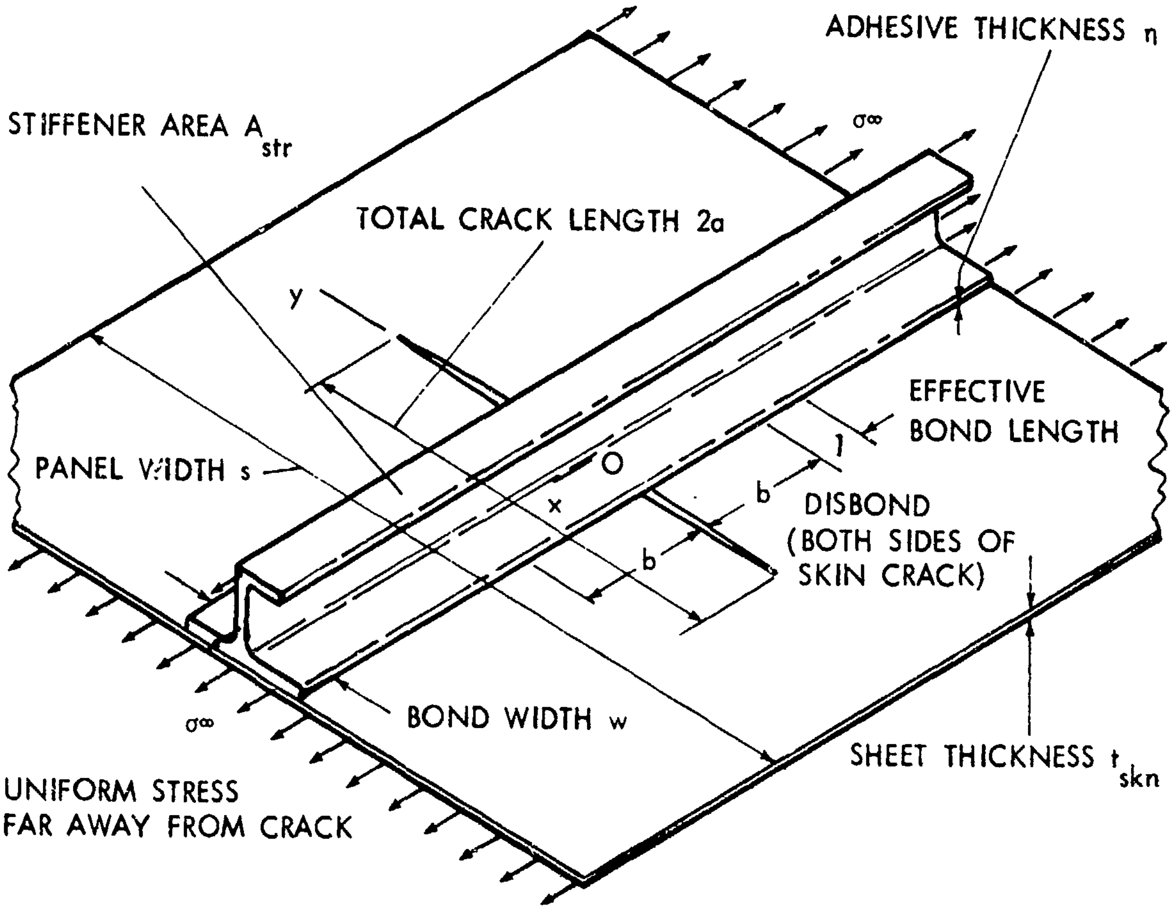


FIGURE 66 . TWO-BAY SHEET CRACK RESTRAINED BY BONDED CENTRAL STIFFENER

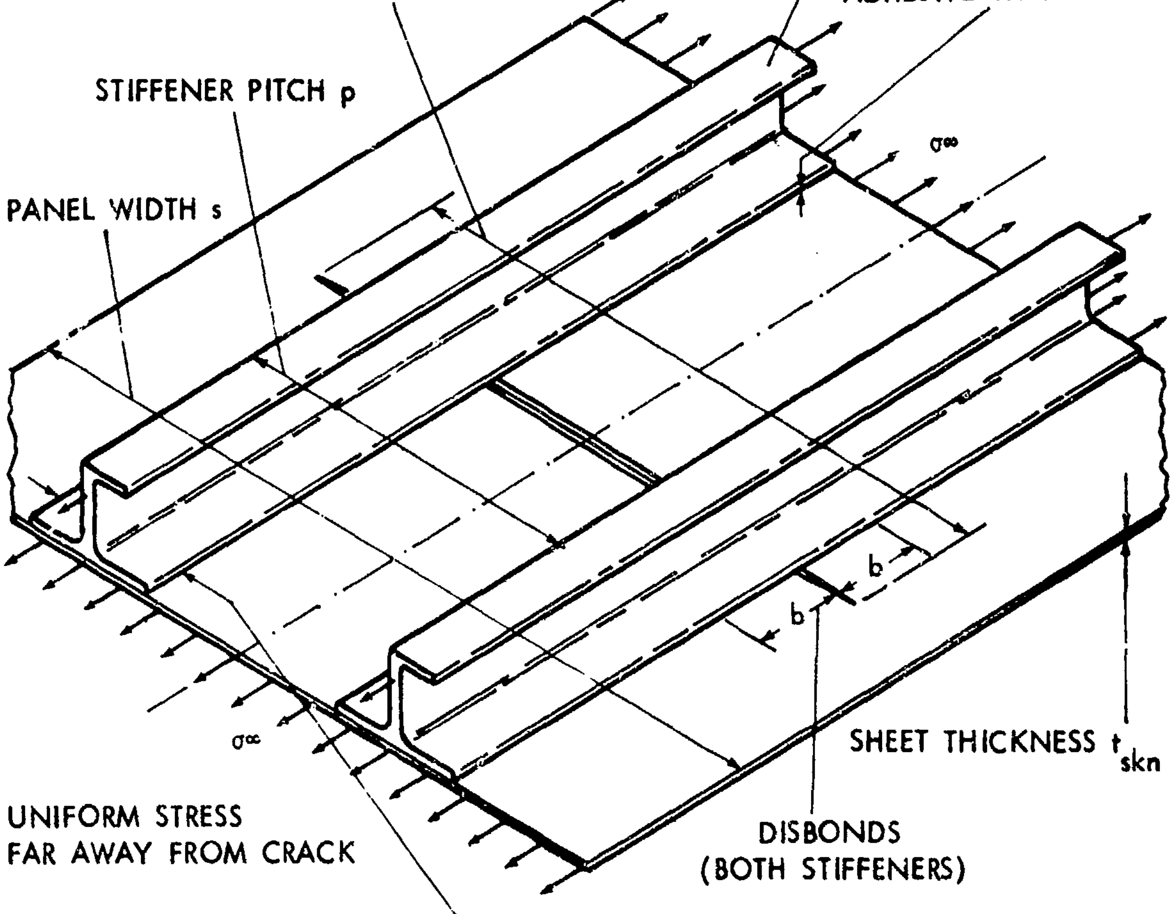


FIGURE 67. ONE-BAY SHEET CRACK RESTRAINED BY BONDED STIFFENERS

- BONDS MORE LIKELY TO FAIL AT ABRUPT CHANGES THAN AT WELL-DESIGNED SPLICES
- NEW ANALYTICAL ELASTIC-PLASTIC SOLUTIONS PROGRAMMED
- 2.0 CENTS PER CASE
- CHARACTERISTIC BEHAVIOR BOTH FOR SKIN CRACKS AND BROKEN STIFFENERS

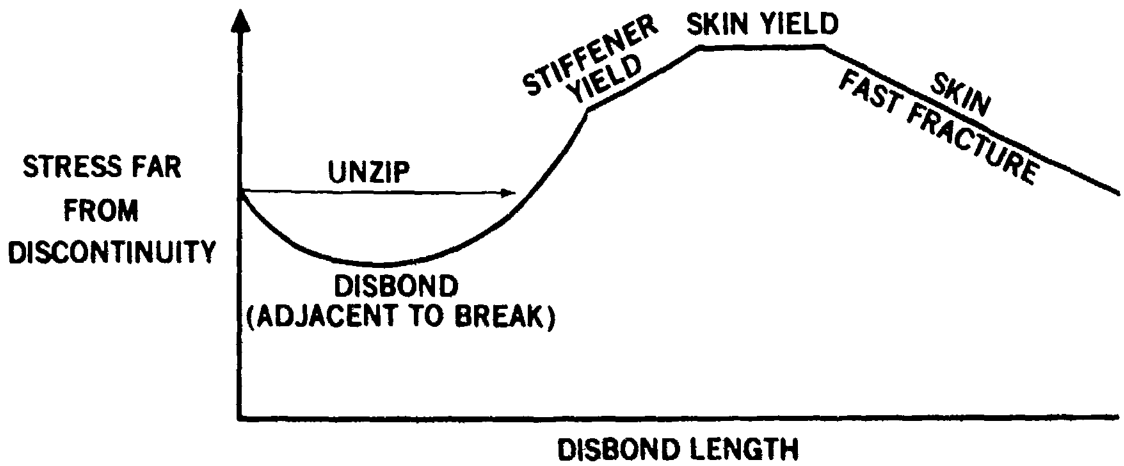


FIGURE 68 . DISCONTINUITIES AND CRACKS IN BONDED STRUCTURE

Figure 64, the stiffener shape is such that the web and unbonded flange transfer load: (1) through the bond, (2) into the skin, and (3) through a short length adjacent to the stiffener crack or disbond, over only a small fraction of the total bond width. This induces a very high local stress in the skin that is the basis of the frame-shear tee cutout problem, in which one stiffener is cut abruptly without adding adequate skin reinforcement to accept the added load, Figure 69.

When discontinuities cannot be avoided, as in the frame-shear tee cutout problem, care should be taken to relieve the load transfer through the bondline at the very end of the bonded member. A typical properly designed frame-shear tee intersection with a reinforcing doubler is shown in Figure 70. If this is not done, fatigue failures may initiate in the skin adjacent to the end of the part (see Figure 71. One obvious way to relieve these high adhesive shear stresses at the end of the longeron is to taper the longeron cap and web and chamfer the base as shown in Figure 72. If the longeron ends adjacent to another member, which is often the case, the gap between the stiffener and the other member can be bridged with a bonded or mechanically fastened gusset as shown in Figure 73.

A high local stress can also result from a broken stiffener, Figure 74. Therefore, in high stress areas, one panel design condition is to assess the effect of a broken stiffener or the damage tolerance capability of the skin. One approach is to use an elastic/plastic finite element analysis to obtain a stress concentration factor. The factor and stress level, together with crack growth data, will determine if there is a damage tolerance problem. It should be noted that the local stress level is also a function of the amount of disbond that may exist.

10.6.4 Cracking in Single Lap Bonded Joints

An additional concern is the early detection of any incipient failure of single-lap bonded joints in pressurized fuselages. The highest stresses in the metal, where skin cracks usually start, develop on the bond side of the sheets precisely at the ends of the overlap. At such a location, the crack is effectively undetectable. If the structure is not painted, the crack will cause a visible crazing in the adhesive fillet before it grows to a critical

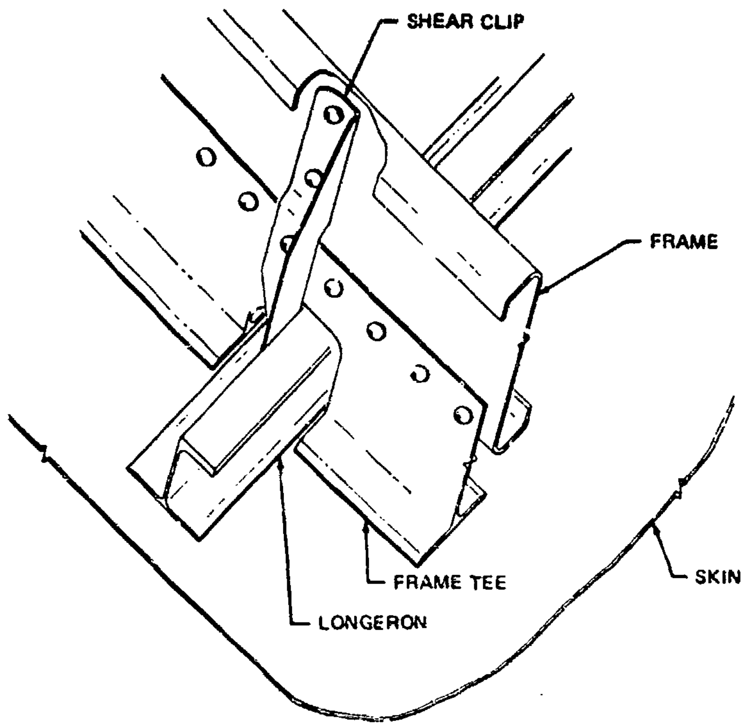


FIGURE 69 . POORLY-DETAILED TYPICAL FRAME AND INTERNAL LONGERON INTERSECTION (DISCONTINUITY IN FRAME TEE)

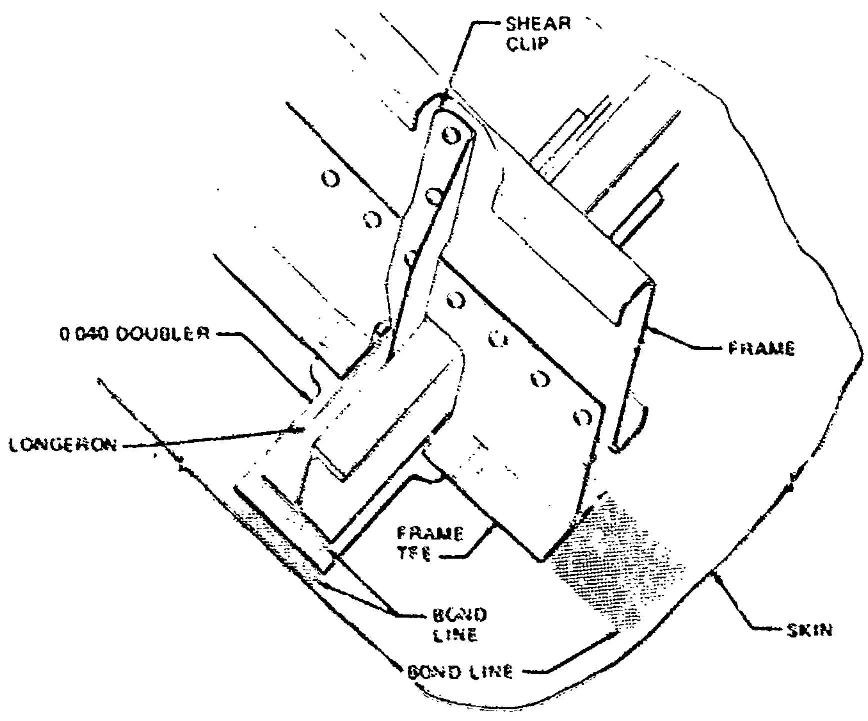


FIGURE 70 . TYPICAL FRAME AND INTERNAL LONGERON INTERSECTION WITH DOUBLER TO PROVIDE STRUCTURAL CONTINUITY

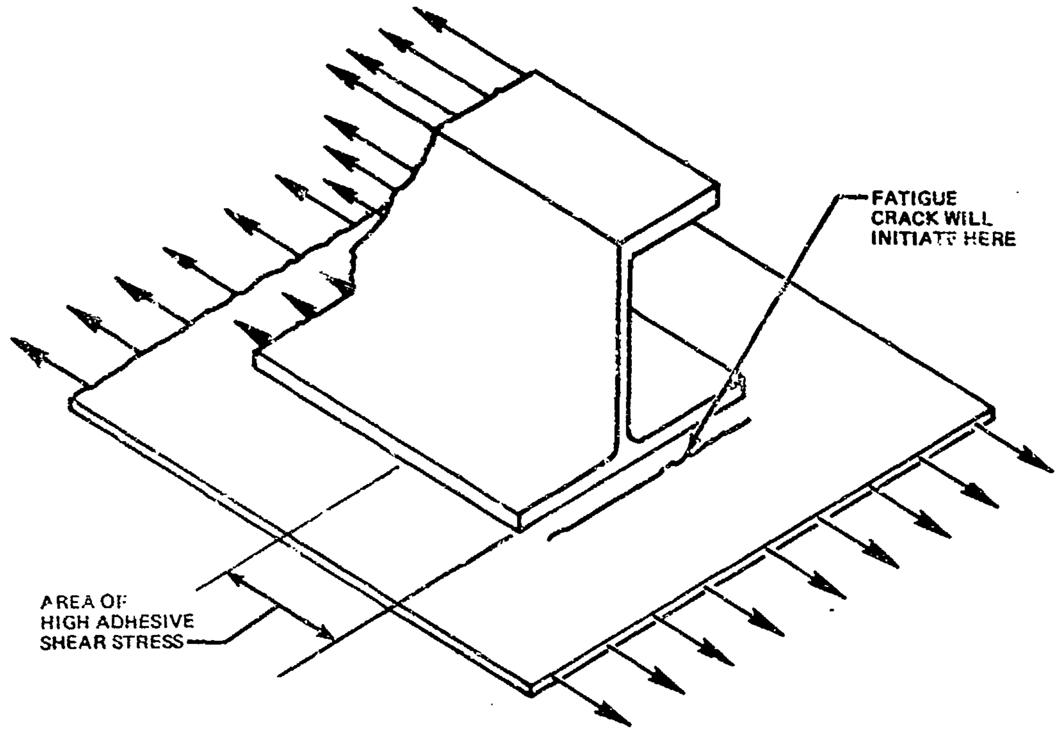


FIGURE 71 . FATIGUE CRACK DUE TO DISCONTINUOUS STIFFENER

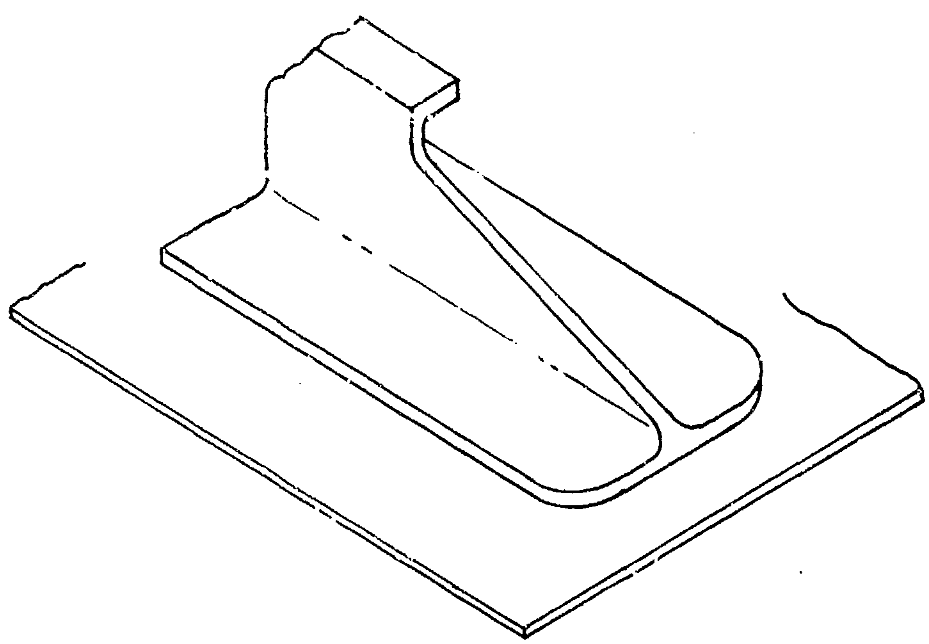


FIGURE 72 . TAPERED STIFFENER REDUCES ADHESIVE SHEAR STRESS NEAR DISCONTINUITY

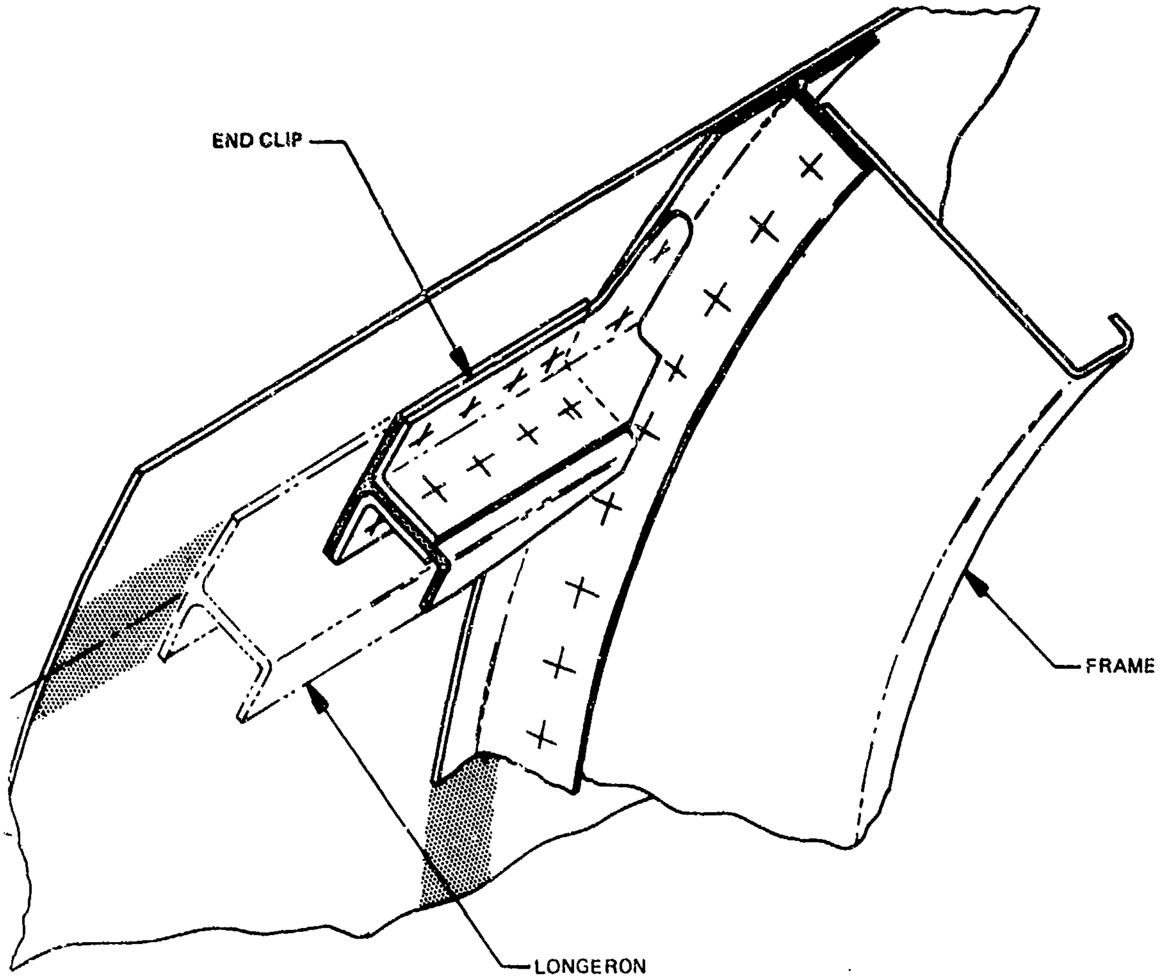


FIGURE 73 . LONGERON TERMINATING AT FRAME TEE

size, but such an inspection aid is not always available. The solution is to use a fingered edge on the inner sheet, as shown in Figure 75 . The interaction between fuselage pressurization and the eccentricity in load path is such as to always make the outer sheet more critical. The fingers both: (1) decrease the bending moment by adding greater flexibility and (2) ensure that any skin crack growth starting at the end of one or more fingers will be forced out into an open inspectable area before it can grow to critical size in an uninspectable area. This refinement may not always be necessary, but is a simple way to obtain a long-lived easily-inspectable structure.

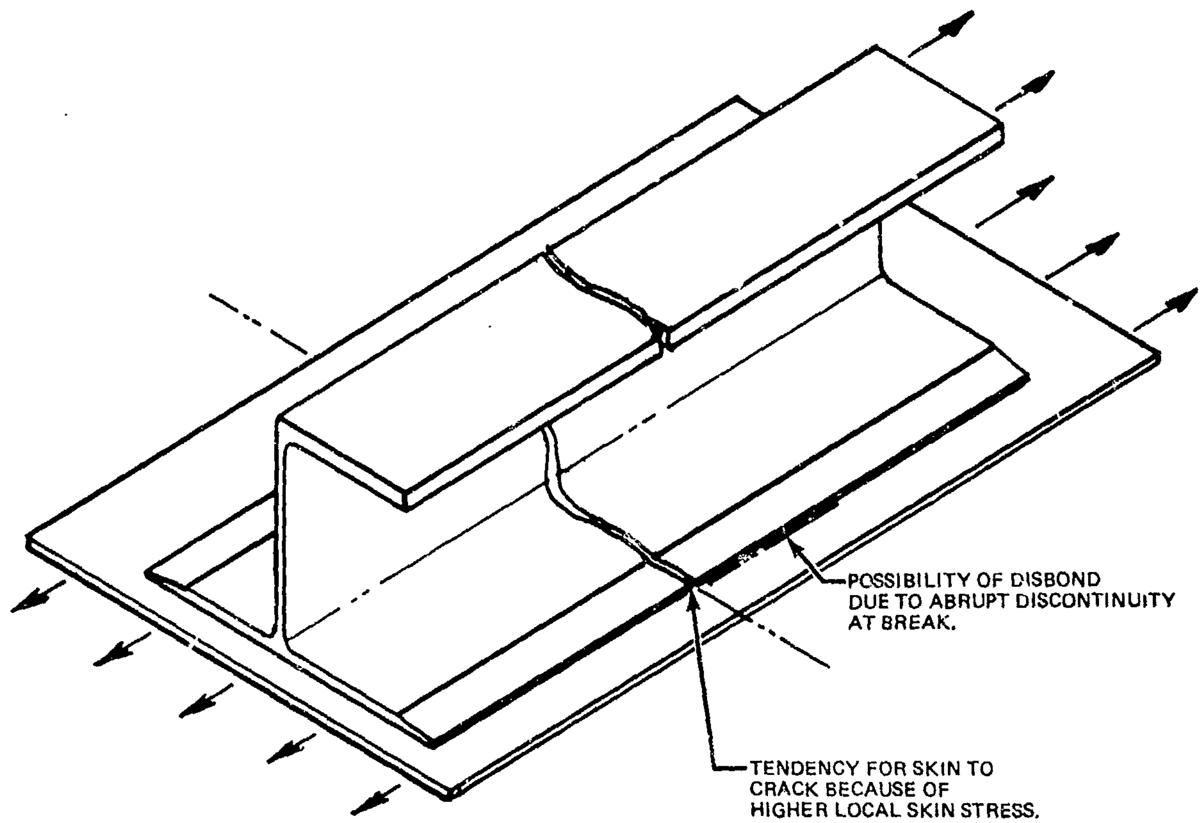


FIGURE 74 . STRESS CONCENTRATION AT BROKEN STIFFENER

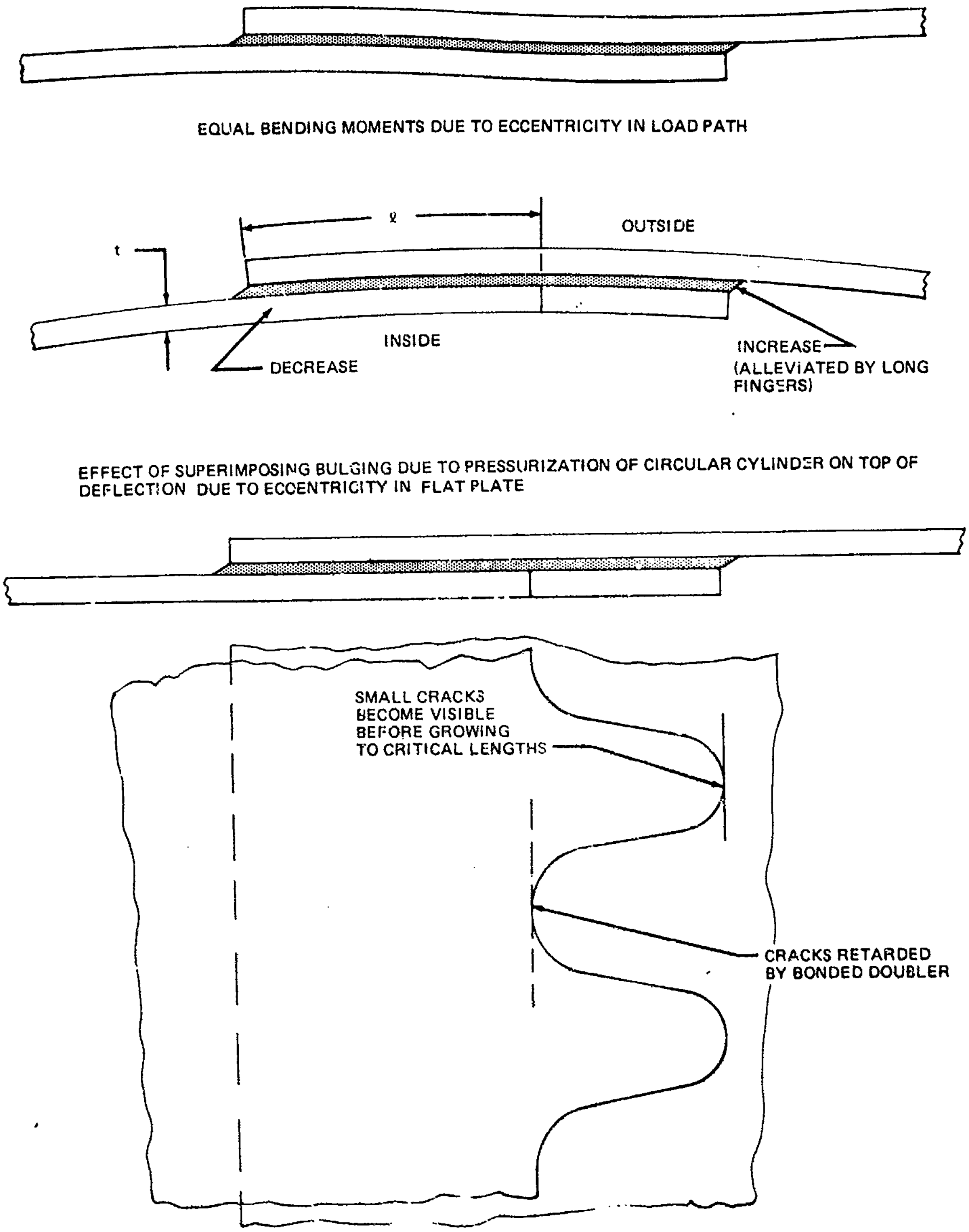


FIGURE 75 . FINGER DOUBLERS FOR LONGITUDINAL SINGLE-LAP BONDED FUSELAGE SPLICES

SECTION XI
ANALYSIS

Design/analysis guidelines are included in this section for: (1) Double-strap, (2) Double-Lap, (3) Single-Lap and (4) Single-Strap (Flush) bonded joints and bonded doublers. Precise elastic/plastic analysis methods are referenced since they consist of lengthy complex analyses which are beyond the scope of a handbook.

11.1 Double-Strap and Double-Lap Bonded Joints

Double-strap and double-lap joints can be designed in two forms. One has a uniform thickness strap and the other has tapered straps as shown in Figure 76. A double-lap joint is one half of a double-strap joint. The tapered-lap splice is applicable for those thicknesses for which the limited adhesive peel strength with uniform outer adherends would preclude their use. Design/analysis, and verification methods for these joints are described below.

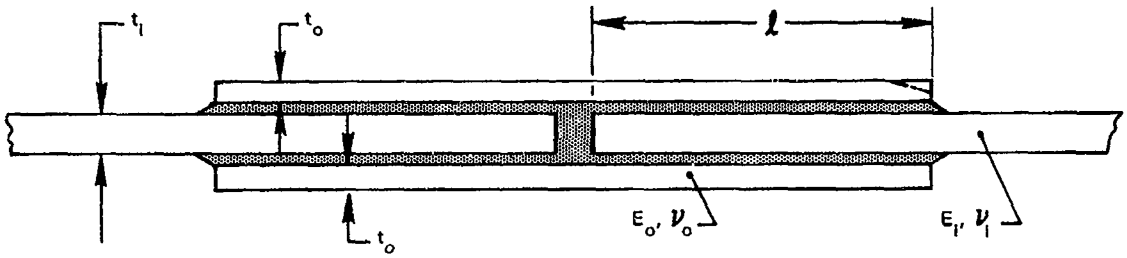
Much of the information in this section is applicable also to bonded doublers, either when used alone or in conjunction with mechanical fasteners.

11.1.1 Simplified Design/Analysis Procedure

The design and analysis methods presented for double-strap bonded joints are simplified versions of those presented in Reference 1. The assumptions on which the analyses are based are:

(1) elastic-plastic adhesive in shear,
(2) elastic adhesive in peel,
(3) adherends elastic until yield is reached,
(4) adhesive not to be loaded beyond proportional limit by recurrent loads, (prefer to stay under 50% of yield strength), leaving considerable margin for once-in-a-lifetime load, manufacturing imperfections, and damage tolerance, and
(5) sufficient total overlap to restrict adhesive stress in the middle of the overlap to prevent cumulative creep damage of the adhesive.

The simplified method is explained in Figure 77.

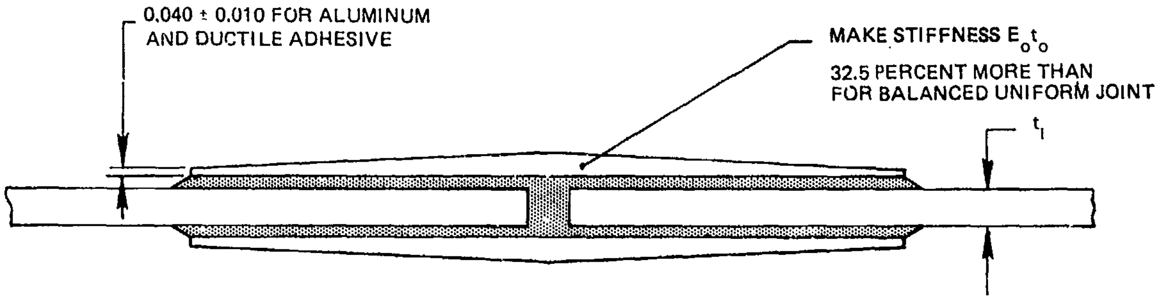


STIFFNESS BALANCE: MAKE $\Sigma E_o t_o$ EQUAL TO OR SLIGHTLY EXCEED $E_i t_i$

OVERLAP REQUIREMENT: MAKE ℓ LARGE ENOUGH TO TRANSFER LOAD AND RESIST CREEP IN MOST ADVERSE ENVIRONMENT

PEEL STRESS RELIEF: TAPER EDGES DOWN TO 0.03 INCH OVER 0.25 INCH IF t_o EXCEEDS 0.05 INCH

DOUBLE-STRAP BONDED JOINT



SHEAR STRENGTH INCREASE OF 24 PERCENT WITH RESPECT TO UNIFORM BALANCED JOINT.

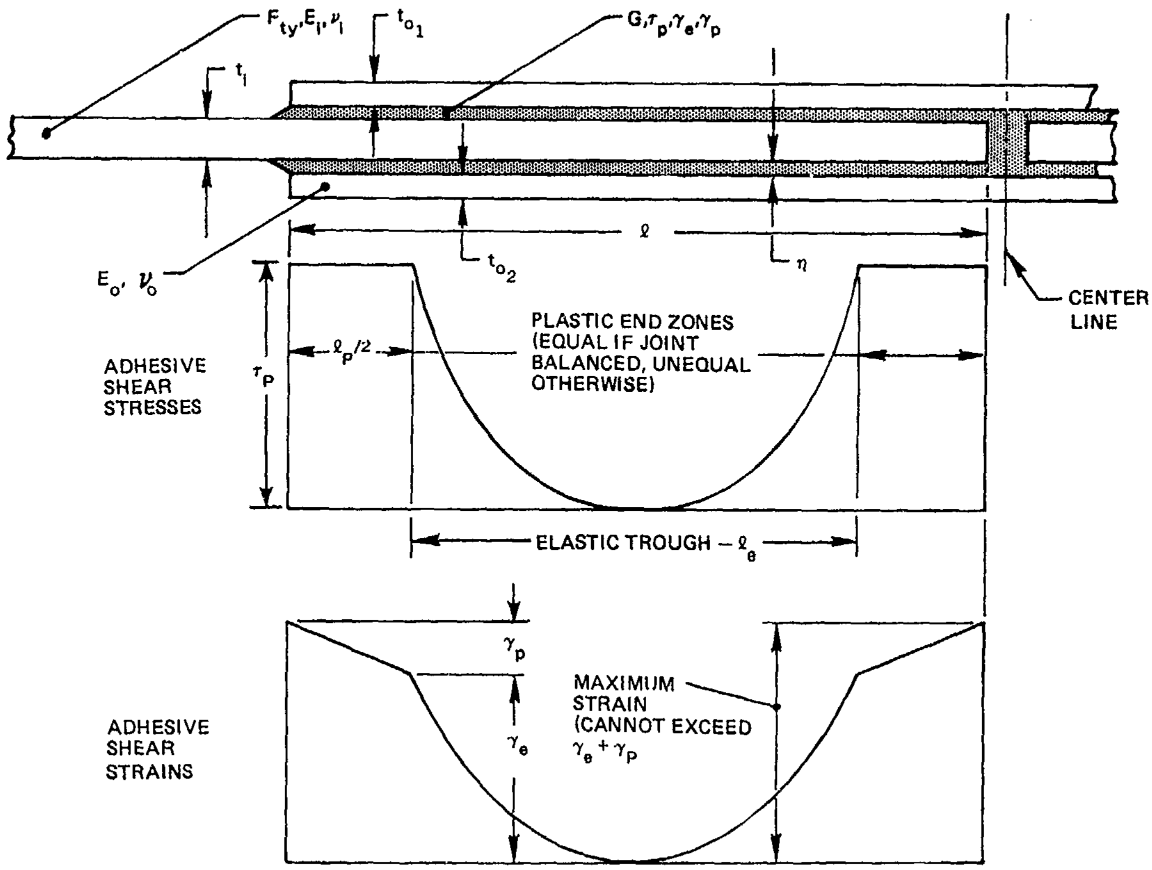
PEEL STRESS RELIEF AT EDGES OF SPLICES.

USEFUL FOR t_i GREATER THAN 0.1 INCH.

POSES INSPECTION AND POSSIBLE FABRICATION PROBLEMS.

TAPERED-LAP BONDED JOINT

FIGURE 76 . FEATURES OF DOUBLE-STRAP AND DOUBLE-LAP BONDED JOINTS



LENGTH OF PLASTIC ZONES LIMITED BY ULTIMATE STRENGTH OF INNER ADHEREND OR OUTER ADHERENDS

$$\ell_p = \frac{F_{tu} t_i}{2 \tau_p}$$

WITH τ_p EVALUATED AT TEMPERATURE GIVING LEAST VALUE, i.e., MAXIMUM SERVICE TEMPERATURE

LENGTH OF ELASTIC TROUGH JUST SUFFICIENT TO REDUCE SHEAR STRESS IN MIDDLE TO EFFECTIVELY ZERO

$$\ell_e = 6/\lambda \qquad \lambda^2 = \frac{G}{\eta}\left[\frac{2}{E_i t_i} + \frac{2}{E_o(t_{o_1} + t_{o_2})}\right]$$

EVALUATE λ AT TEMPERATURE GIVING LEAST VALUE – AGAIN, THE HOTTEST

OPTIMUM OVERLAP = $\ell_p + \ell_e$

(NO GREATER OVERLAP CAN INCREASE JOINT STRENGTH)

FIGURE 77. SIMPLIFIED DESIGN METHOD FOR DOUBLE-STRAP BONDED JOINTS

Except that the ends of the splice plates may need tapering to protect against peel stresses, there are no viable options available to improve upon this double-strap bonded joint design without going to a stepped-lap joint. It is necessary to verify that the design does not exceed the adhesive shear strain or peel stress capabilities and that the applied fatigue loads do not load the adhesive into the plastic zone. In addition, the maximum sustained load capability at the joint is governed by the yield (not the ultimate) strength of the adherends to provide for damage tolerance as noted in Section 10.1.2.

11.1.2 Verification Method

The following formulae should be used to verify the adequacy of the joint as designed by the simplified method above. For tensile or compressive lap shear loading, the bond shear strength per unit width (for long overlap joints) is given by

$$P = \text{lesser of } \sqrt{2\tau_p n \left(\frac{\gamma_e}{2} + \gamma_p\right) 2E_i t_i \left[1 + \frac{E_i t_i}{E_o(t_{o1} + t_{o2})}\right]} \quad (1)$$

$$\text{and } \sqrt{2\tau_p n \left(\frac{\gamma_e}{2} + \gamma_p\right) 2E_o (t_{o1} + t_{o2}) \left[1 + \frac{E_o(t_{o1} + t_{o2})}{E_i t_i}\right]}$$

If possible, this should exceed the ultimate strength of the inner adherends by at least 50 percent. Note that these equations are independent of the over-lap length. For very short overlaps,

$$P = 2\tau \ell \quad (2)$$

but such joints should not be used because of poor creep and environmental resistance. If the load in the sheets is in-plane shear, rather than tension or compression, the bond shear strength per unit length is given by

$$S = \text{lesser of } \sqrt{2\tau_p\eta\left(\frac{\gamma_e}{2} + \gamma_p\right)\frac{E_i t_i}{(1+\nu)}\left[1 + \frac{E_i t_i}{E_o(t_{o1} + t_{o2})}\right]}$$

$$\text{and } \sqrt{2\tau_p\eta\left(\frac{\gamma_e}{2} + \gamma_p\right)\frac{E_o(t_{o1} + t_{o2})}{(1+\nu)}\left[1 + \frac{E_o(t_{o1} + t_{o2})}{E_i t_i}\right]} \qquad (3)$$

If the sheets being bonded together are subjected to simultaneous tension (or compression) and in-plane shear loads, replace the $(\frac{\gamma_e}{2} + \gamma_p)$ terms in equations (1) and (3) by γ_{max} and obtain γ_{max} by solving equations (1) and (2) using the yield strength P and S values for the thinner adherends. The respective shear strains induced in the adhesive by these two load components are orthogonal. Therefore, the joint has adequate strength provided that the shear strains satisfy the inequality

$$(\gamma_{max})^2_{\text{tension or compression}} + (\gamma_{max})^2_{\text{in-plane shear}} \ngtr (\gamma_e + \gamma_p)^2 \qquad (4)$$

The fatigue load analysis involves an inversion of equations (1) or (3), as appropriate, into the form

$$\left(\frac{\gamma_e}{2} + \gamma_p\right) = \text{greater of } \frac{P^2}{2\tau_p\eta\, 2E_i t_i\left[1 + \frac{E_i t_i}{E_o(t_{o1} + t_{o2})}\right]}$$

$$\text{and } \frac{P^2}{2\tau_p\eta\, 2E_o(t_{o1} + t_{o2})\left[1 + \frac{E_o(t_{o1} + t_{o2})}{E_i t_i}\right]} \qquad (5)$$

and must not exceed $\frac{\gamma}{2}$ and preferably be less than $\frac{\gamma}{4}$. Equation (5) is usually critical for pressurized transport fuselages; however, the corresponding inversion of equation (3) should also be checked.

It is also necessary to check the peel stress at the ends of the splice plates. The peak peel stress is given by:

$$\sigma_{peel} = \tau_p \left(\frac{3E_c' \, (1-\nu^2) \, t_o}{E_o n} \right)^{\frac{1}{4}} , \qquad (6)$$

in which E_c' is the effective peel modulus of the adhesive film, as constrained by the stiffer thicker metal adherends. This equation can be rearranged to express the maximum permissible thickness of the ends of the outer adherends,

$$t_{o_{max}} = \frac{E_o n}{3(1-\nu^2)} \left[\frac{1}{E_c} + \frac{k_1}{E_i} + \frac{k_2}{E_o} \right] \left(\frac{\sigma_{peel}}{\tau_p} \right)^4 , \qquad (7)$$

in which k_1 and k_2 represent the contribution to the flexibility of the thin adhesive layer by the stiffer thicker outer adherends. To allow for manufacturing and curing variability, the calculated $t_{o_{max}}$ should be reduced by a factor of 5 to give a 50 percent margin on peel. For typical ductile adhesive properties and aluminum adherends, $t_{o_{max}}$ should not exceed 0.05 to 0.063 inch. Thicker splice members should have their ends tapered to relieve peel stresses. A tip thickness of 0.040" ± .010" is reasonable. If necessary, the splice thickness should be increased to a value about 32.5 percent greater than for a balanced joint to compensate for the loss in shear strength, due to the tapering, to relieve peel. A precise elastic-plastic analysis of tapered bonded splices is not currently available, but an approximate plastic analysis is given in Reference 2.

All of the design checks should be evaluated for the most critical combination of loads and environment. For example, the shear strength will typically be more critical at the highest temperature because the aircraft joint still has to withstand proof pressurization on the ground sitting in the desert during some inspections. The fatigue and peel loads are usually worst when the adhesive is coldest and most brittle at cruise altitude.

If materials having dissimilar coefficients of thermal expansion; e.g., aluminum to titanium, are to be bonded together, some of the equations above need additional terms. The interested reader is referred to the basic derivations in Reference 2.

11.1.3 Design Tables for Joint Proportions

The considerations discussed in Sections. 11.1.1 and 11.1.2 led to the derivation of the design tables presented below to assist in the design of double-strap or double-lap bonded joints in aluminum alloys for the PABST program.

Table 2 gives the nominal recommended overlaps for balanced double-lap and double-strap joints used in the PABST designs.

TABLE 2
RECOMMENDED OVERLAPS FOR BALANCED DOUBLE LAP AND DOUBLE STRAP JOINTS

CENTRAL SHEET THICKNESS t_i (INCH)	0.040	0.050	0.063	0.071	0.080	0.090	0.100	0.125
RECOMMENDED OVERLAP ℓ (INCH)†	1.21	1.42	1.68	1.84	2.01	2.20	2.39	2.84
STRENGTH OF 2024-T3 ALUMINUM (LB/IN.)	2600	3250	4095	4615	5200	5850	6500	8125
POTENTIAL ULTIMATE BOND STRENGTH (LB/IN.) * ‡	8115	9073	10184	10812	11477	12173	12831	14346

† Based on 160°F properties giving lowest value of λ

* Based on -50°F properties giving lowest joint strength. (The thicker adherends, say 0.100 and 0.125, would need peel stress relief by feathering the outer edges of the splices.)

‡ For nominal adhesive thickness η = 0.005 in.

The lengths calculated are for balanced joints. Slightly different overlaps would be used if $E_i t_i \neq 2E_o t_o$. These overlaps are sufficient to permit riveted repairs, if necessary. The potential bond shear strengths can be multiplied by the following sequential modification factors as required.

For Thin bonds - reduce strengths by $\sqrt{\eta/.005}:1$

For in-plane shear - reduce by 0.62:1

For fatigue with γ_{max} restricted to 0.05 - reduce by 0.138:1

For stiffness imbalance, see Table 3

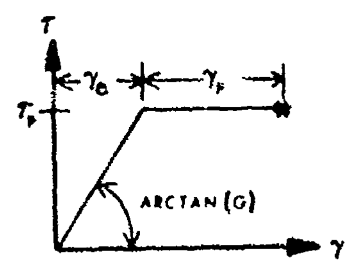
TABLE 3
STIFFNESS IMBALANCE RATIOS

$2E_ot_o/E_it_i$	0.2	0.4	0.6	0.8	1.0	1.2	1.4	1.6	1.8	2.0
RATIO OF STRENGTHS OF UNBALANCED AND BALANCED JOINTS	0.35	0.53	0.69	0.85	1.0	0.96	0.93	0.90	0.88	0.87

The recommended joint dimensions given here have been derived for typical 250°F cure ductile adhesives using the properties in Table 11.3.

TABLE 4
TYPICAL CHARACTERISTICS FOR 250°F CURING DUCTILE ADHESIVES

Temperature	τ_p (psi)	G psi	γ_e	γ_p
R.T. (70°F)	5,000	50,000	0.1	1.0
-50°F	7,000	60,000	0.12	0.5
+140°F	2,500	40,000	0.063	1.5



Most such adhesives have very similar mechanical properties, even though they vary greatly in resistance to adverse environments. The same procedures would lead to very similar results for 350°F cured ductile adhesives except that, because of a lower peel strength at -67°F, the tapering of the extremities of the splice plates should begin at 0.030 inch instead of 0.040 inch. Subject to that proviso, the joint dimensions given here should be applicable to any ductile adhesive (epoxy nitrile, epoxy nylon, vinyl phenolic, nitrile phenolic, etc.) used on subsonic transport aircraft. The unmodified adhesives (epoxy and phenolic) used for high-temperature applications ($> 180°F$) are sufficiently more brittle that separate calculations should be performed.

11.1.4 Worked Examples

Consider the following sample bonded joints in 2024-T3 aluminum structure using a typical ductile 250°F curing adhesive for which the properties are given in Table 4

Case 1: Longitudinal fuselage double-strap splice in 0.05 inch skin. One-P hoop load is 772.20 lb/inch; i.e., load intensity in the fuselage skin which is a product of design cabin pressure (7.15 psi) times fuselage radius (108 inches).

(a) Make the splice sheets 0.032 inch thick to avoid the weakness, in the middle of the strap, found by test. A balanced joint, 0.025 inch splice sheets, would have a slightly greater bond strength but the weak link will be in the metal for an 0.05 inch skin and test results indicate a greater likelihood of fatigue failure right in the middle of the splice, over where the skins butt together. Table 3 indicates that the bond stress loss for $2 E_o t_o/E_i t_i = (2 \times .032)/0.05$ is only about 5 percent.

(b) Read off the overlap from Table 2 as 1.42 inch. To explain in more detail how this length was arrived at, follow Figure 77, step by step.

$$\ell_p = \frac{F_{tu} t_i}{2 \tau_p}; \quad \ell_e = \frac{6}{\lambda}, \quad \lambda^2 = \frac{G}{\eta} \left[\frac{2}{E_i t_i} + \frac{2}{E_o (t_{o_1} + t_{o_2})} \right]$$

Note, use average values of properties not reduced allowables. Testing environment for adhesive was 140°F, 100% R.H. Lower temperatures are less critical and do not require as long an overlap.

$$\ell_p = \frac{65{,}000 \times 0.05}{2 \times 2{,}500} = 0.65 \text{ inch}$$

and $$\lambda^2 = \frac{40{,}000 \times 2}{0.005 \times 10.5 \times 10^6} \left[\frac{1}{0.05} + \frac{1}{0.064} \right]$$

or

$$\lambda = 7.37$$

therefore

$$\ell_e = 0.81 \text{ inch}$$

and

$$\ell = \ell_p + \ell_e = 1.46 \text{ inch},$$

so the 1.42 inch calculated on the basis of ultimate material strength (65 ksi) for short-term loads and balanced adherends ($t_o = 0.025$) is slightly less than for .032 inch outer splice sheets (t_i) but within normal manufacturing tolerances.

(c) Verify that there is no adhesive shear strain problem. From equation (1) the bond shear strength, in lb/inch is the lesser of

$$\sqrt{2 \text{x} 2500 \text{x} .005 \text{x} \left(\frac{.063}{2} + 1.5\right) \text{x} 2 \text{x} 10.5 \text{x} 10^6 \text{x} 0.05 \; 1 + \frac{0.050}{0.064}} = 8462 \text{ lb/inch}$$

and

$$\sqrt{2 \text{x} 2500 \text{x} .005 \text{x} 1.5315 \text{x} 2 \text{x} 10.5 \text{x} 10^6 \text{x} .064 \left(1 + \frac{0.064}{.05}\right)} = 10{,}832 \text{ lb/inch}$$

for the harshest environment (140°F, 100% R.H.). Again, checks at room temperature and -65°F show more strength. In general, one should check for each end and the middle of the operating temperature range. However, for ductile adhesives, in the absence of adherend thermal mismatch, the highest temperature is usually most critical. For brittle adhesives, and for any adhesive bonding thermally dissimilar materials, it is the lowest temperature that is most frequently the worst. Since the maximum possible metal strength outside the joint is $F_{tu} \times t_i$ or 65000 x 0.05 = 3250 lb/in, there is obviously a considerable margin at ultimate load. Consider the effect of the high adhesive flow associated with the new 250°F ductile adhesives. Instead of the ideal 0.005 to 0.010 inch bond thicknesses, pinch off at the edges leads to bonds which are no more than 0.002 inch thick. Therefore the 8,462 lb/inch stated above should be reduced in accordance with the factors in Section 11.1.3 to

$$8462\sqrt{\frac{.002}{.005}} = 5352 \text{ lb/inch}$$

There is still an adequate margin at ultimate load.

$$\text{M.S.} = \frac{5352}{3250} - 1 = .65$$

(d) For the fatigue load condition, it is more appropriate to use room temperature or cold environment to check that the adhesive is not loaded into the plastic state. The -70°F condition is the more severe. From equations 5, the strain given by

$$\left(\frac{\gamma_e}{2} + \gamma_p\right)$$

is the greater of

$$\frac{772.2^2}{2x7000x\ .005x10.5x10^6x0.05\ \left(1 + \frac{0.05}{0.004}\right)} = 0.0080$$

and

$$\frac{772.2^2}{2x7000x.005x10.5x10^6x.064\ \left(1 + \frac{.064}{.050}\right)} = 0.0056$$

that is γ_e = 0.016 which is only 13 percent of the cold elastic strain of 0.12 inch/inch. Therefore, the fatigue loads will not be a problem for this thin skin unless the manufacturing imperfections are gross. Even if the bond were squeezed down to 0.002 inches at the highly stressed edges of the overlap, the strain would be raised only to 0.046 or 38 percent of the elastic capability.

As a rough guide for the fatigue joint strength, Section 11.1.3 would permit 0.138 x 8462 = 1168 lb/inch for using no more than half of the elastic adhesive strain. This exceeds the applied 772.2 lb/inch by an adequate margin.

(e) the final check is on peel stress which will be most severe at the lowest temperature. Equation (6) predicts that

$$\sigma_{peel} = 5000 \left[\frac{3x500,000x0.91x0.032}{10.5x10^6x0.005}\right]^{1/4} = 4775 \text{ psi}$$

which is less than the typical peel strength of 10,000 psi for ductile adhesives at room temperature. Since the peel strength of the PABST adhesive (FM-73) is better at -70°F than at room temperature, there will be no problem here. However, some ductile adhesives have poor low temperature peel strengths, so the appropriate elastic properties would need to be obtained. Note that the adhesive peel modulus of 500,000 is an estimate based on the bi-axial constraint of the aluminum which makes the adhesive nearly infinitely stiff because it is close to incompressible. There is very little test data available in this area and the rule based on observations of tapering the tips of any outer adherends down to 0.04 ± .010 inch for highly ductile adhesives with high peel strengths and 0.02 ± .010 inch for adhesives having poor peel strengths is probably more useful with the present state of the art.

Case 2: Consider now the same longitudinal fuselage splice, but with a skin thickness of 0.125 inch.

(a) The splice sheets can now be 0.063 inch thick since they will not deform at the middle during curing.

(b) From Table 2, the recommended overlap is 2.84 inch. Checking with the method per Figure 3,

$$\ell_p = \frac{65,000 \times 0.125}{2 \times 2,500} = 1.63 \text{ inch}$$

and

$$\lambda^2 = \frac{40,000 \times 4}{0.005 \times 10.5 \times 10^6} \times \frac{1}{0.125} = 24.38$$

or

$$\lambda = 4.94$$

therefore

$$\ell_e = 1.21$$

and

$$\ell = 2.84 \text{ inch}$$

The 2.84 inch applies for rapidly applied loads up to a skin stress of 65 ksi, to prevent local damage from spreading.

(c) Verify that there is no adhesive shear strain problem. From equation (1), the bond shear strength is

$$[2x2500x0.005x(\frac{.063}{2} + 1.5)x2x10.5x10^6x.125x2]^{1/2} = 14,178 \text{ lb/inch},$$

for the harshest environment (140°F, 100% R.H.)

The maximum possible metal load which could be developed by redistribution of load around a damaged area is 65,000 x 0.125 = 8125 lb/inch, so the bond is still stronger than the metal.

(d) For the fatigue load condition, due to the 772.2 lb/inch pressure load, the maximum adhesive strain in the cold environment will be reduced as follows

$$(\frac{\gamma_e}{2} + \gamma_p) = \frac{772.2^2}{2x7000x0.005x10.5x10^6x0.125x2} = 0.00324$$

γ_e = 0.006 which is even less than for the thinner 0.05 inch skin.

(e) The peel stress, predicted by equation (6) would now be 5678 psi; i.e., the 4775 psi of case 1 multiplied by $2^{1/4}$. This value would not indicate concern about a possible peel failure but that is contrary to test and service experience. It would appear that the stiffer adherends increase the effective peel modulus E_c' above the 500,000 psi estimate used here and in case 1. Insufficient test data exists to determine properly the value of that modulus under varying degrees of constraint. Therefore, it is recommended here that the empirically deduced procedure of tapering the ends if the thickness exceeds 0.04 inch for a ductile adhesive or 0.02 inch for a brittle adhesive be adhered to.

11.2 Single-Lap Joints

The precise analysis of single-lap joints is very complicated because of the interactions between each of the three potential failure modes: adherend yielding, adhesive peel, and adhesive shear. See Reference 3 for further information.

For design purposes, the following simplified procedures are recommended for the ductile adhesives used on subsonic transport aircraft.

(1) Set the length-to-eccentricity ratio (length-to-thickness for uniform adherends) in the range 50 to 100, depending on the criticality of the application. At maximum possible loads for 2024-T3 and 7075-T6 aluminum alloys, the ratio $\ell/t = 100$ corresponds to bending stresses that are only 10 percent of the membrane stresses, while $\ell/t = 50$ gives a corresponding joint efficiency of 80 percent. In contrast, $\ell/t = 10$ for aluminum produces an efficiency of only 35 percent; i.e., the bending stresses are 65/35 times as high as the membrane stresses that induced them. Figure 11.3 presents a chart with which to calculate the joint efficiency factor. The joint efficiency is a function of the applied stress level, with lower efficiencies for lower stresses.

It should be noted that the PABST program set $\ell/t = 80$ for rivet/bonded splices.

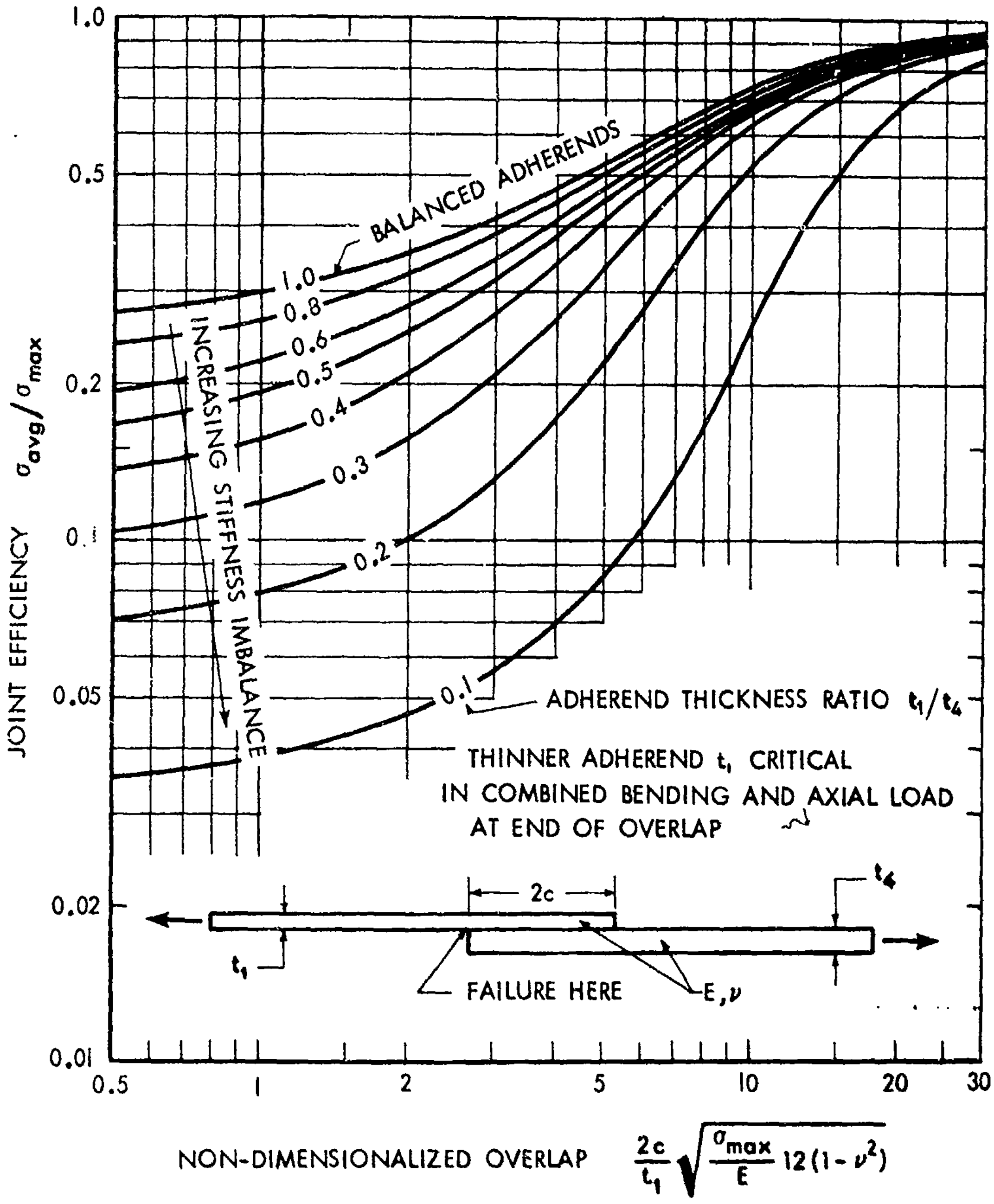


FIGURE 78. EFFECT OF ℓ/t RATIO AND ADHERENT STIFFNESS IMBALANCE ON STRENGTH OF SINGLE-LAP BONDED JOINTS

(2) Taper the edges of the adherends down to 0.030 ± 0.010 inch, if the basic thickness exceeds 0.050 inch, to relieve peel stresses. The taper angle should be about 0.020 inch per quarter of an inch.

(3) Make an approximate check on the shear strength of the adhesive by assuming that the joint is one half of a double-lap joint which has twice as thick mid skin, Section 11.1. Effectively this means that single-lap joints should not be used with aluminum gages in excess of about 0.063 for ductile adhesives and subsonic aircraft.

The effect of using brittle, high temperature adhesives in single-lap joints is to restrict still further the applicable range of adherend thicknesses which could be bonded effectively into a single-lap joint.

The simple rules stated above can be used for design because the correct answers are so strongly dependent on the ℓ/t ratio. It is important to stress that the design condition is one of ℓ/t for the bending of the adherends and not one of bond area for the shear strength of the joint.

The key difference between double-lap and single-lap joints is that bending moments are set up in the metal due to the single-lap load path eccentricity. A simple linear analysis considerably over-estimates the correct bending stresses, typically by a factor of two to three. A non-linear analysis must be used to account for structural distortions under load to relieve the eccentricity in load path. Such refined analyses indicate that the combination of direct and bending stresses can be only about twice as high as the nominal stress instead of the four-times-as-much that the linear analysis would have predicted. Refer to Reference 3 for a more complete discussion. The key to this bending-moment relief is a long overlap to absorb the eccentricity smoothly. Figure 44 shows that, for aluminum adherends, an average stress of 20 ksi will induce a maximum sheet stress of 55 ksi if the overlap/thickness ratio is 10, but only 30 ksi if the ratio is 50. There is usually no point in increasing the ℓ/t ratio beyond 100, but an ℓ/t ratio of 50 should be regarded as an absolute minimum for unsupported single-lap joints. Testing on the PABST program of riveted single-lap joints with sealant, having an ℓ/t ratio of 80, showed that the joints usually attained an adequate fatigue life unless there was countersinking into sheet 0.050 inch thick or less. This is where hot bonding in combination with the fasteners would be valuable.

11.3 Single-Strap (Flush) Bonded Joints

No published analysis exists for this class of joints so considerable reliance must be placed upon testing. The simple design rules of thumb which are apparent are that the total splice width should be about 150 to 200 times the thickness of the sheet being spliced and that the thickness of the tip of the splice should not exceed about 0.04 inch. It has also been confirmed that a linear analysis for the bending stresses is far too conservative. No analysis exists for the shear stresses in the bond, so a simulation of the joint as part of a double-strap splice is all that can be used at present. Likewise, the peel stresses can be calculated only at the critical locations, i.e., the ends of the splice or sheet.

The experimental evidence on this subject for the tapered single-strap splice can be summarized as follows. Fatigue tests were run on a panel of 0.090 inch 7075-76 aluminum alloy 24 inches wide and 48 inches long. The splice plate was 0.125 inch thick, tapered down to a 0.040 inch at the edges and a total of 7 inches wide. Two Z-longerons were bonded longitudinally to restrict the bending deflections of the panel. The test results are recorded in Reference 4. While the panel survived a considerable number of cycles prior to failure, the final failure was a crack running straight across the middle of the splice, from one side to the other, with both stiffeners completely disbonding. The initial fatigue damage prior to fast fracture had not been detected and consisted of a crack in the splice only 1.4 inches long on the outer surface and 0.5 inches long on the inside. The average skin stress at the time of failure was only 24 ksi but analysis predicted a stress of 50,400 psi at the origin of the crack. In other words, this kind of bonded joint has poor fail-safety and is difficult to inspect. The analysis also predicted disbonding throughout a band across the middle of the splice. Testing confirmed the presence of such a band, of width 0.3 to 0.4 inch on each side of the skin joint. The peel stresses decrease as that disbond grows because the splice plate is able to deflect more easily, just as in an equivalent riveted joint (which has a finite distance between the inner rows of rivets instead of just a narrow gap over which to deflect).

11.4 Bonded Doublers

Reference 2 contains some specific non-linear analyses for the bending stresses induced by bonded doublers. A good rule of thumb is that the doubler installation should either terminate at or close to a stiffener to provide a kick-load reaction or should be sufficiently flexible; i.e., ℓ/t in the range 50 to 100 beyond the nearest stiffener, to permit the structure to deflect freely alleviating the bending moments. Figures 79 and 80 show the skin efficiency $[\sigma_{avg}/(\sigma_{avg}+\sigma_{bndg})]$ for a variety of bonded doubler geometries, based on the analysis in Reference 5. The powerful negative effect of short ℓ/t ratios is quite apparent. The adhesive shear stresses can be analyzed by adding a mirror image structure to create a hypothetical double-lap joint, as shown in Figure 81. Alternatively, the precise analysis method in Reference 2 could be employed.

11.5 Cracking and Damage Tolerance

The effect of bonding on the damage tolerance capability of stiffened structure is described in the following subsections. Crack growth and residual strength analysis, as well as durability and damage tolerance tests, are used to demonstrate that the structure designed will meet the criteria, presented in Section IV, for the chemical, thermal, and load spectra environment associated with the service life of the structure.

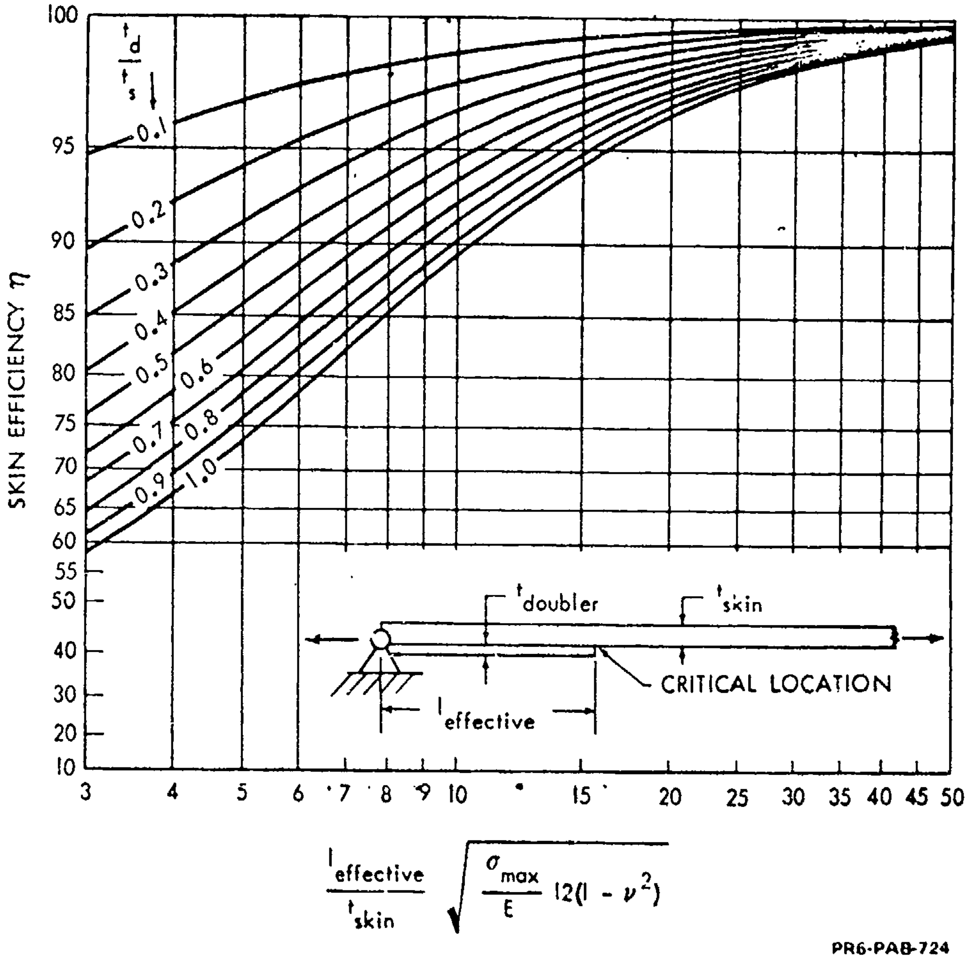
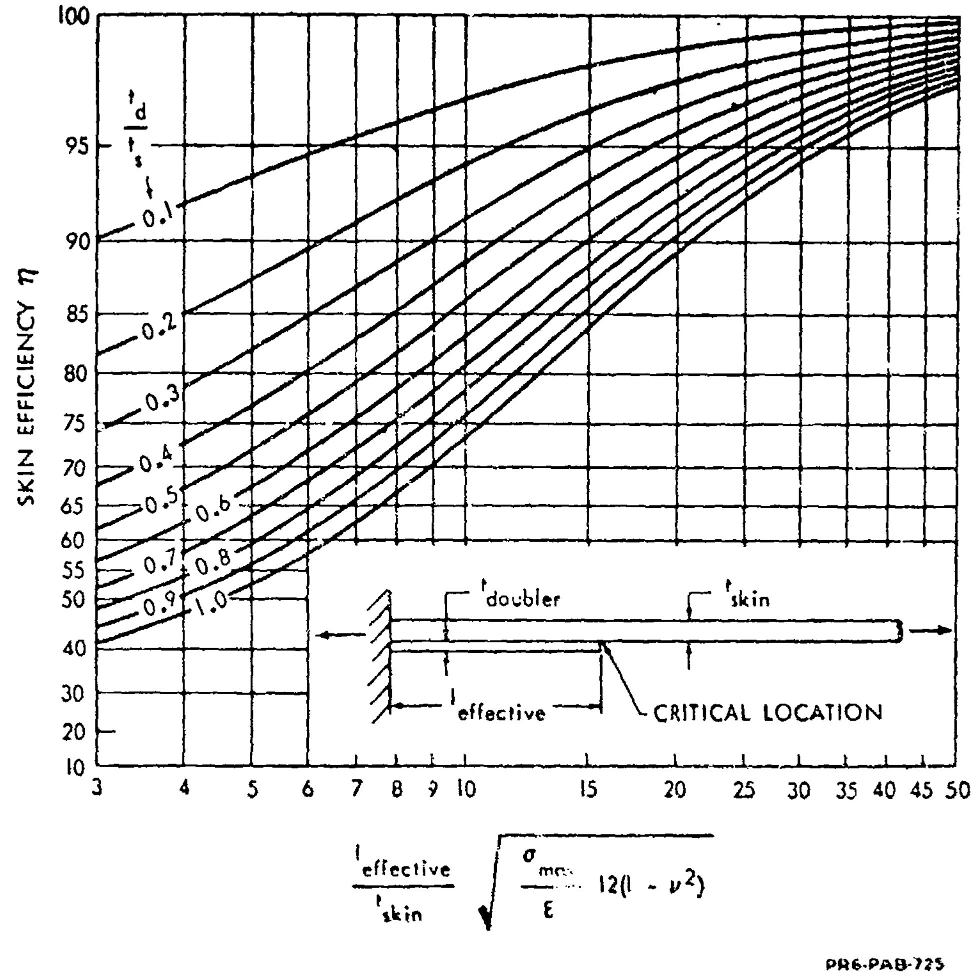
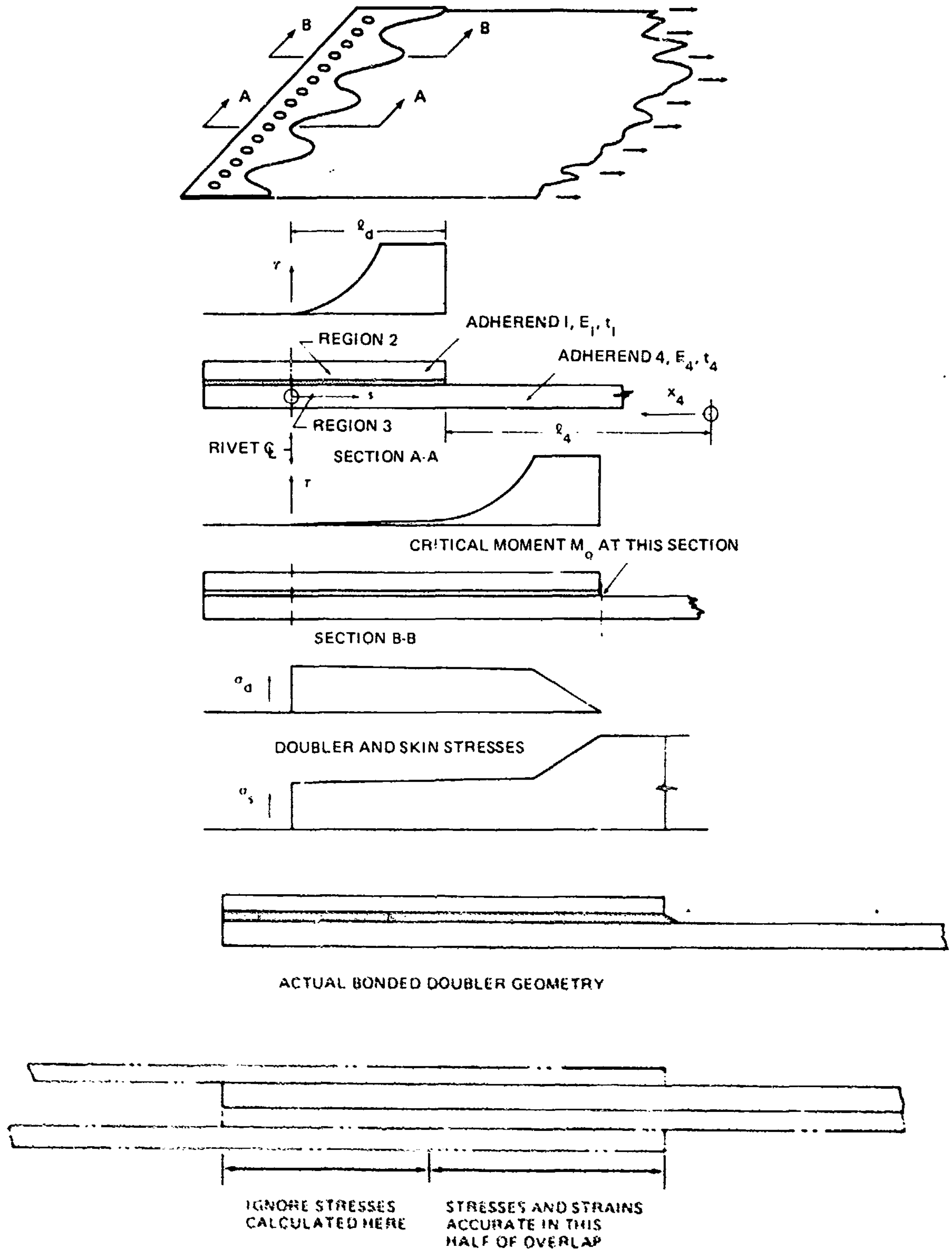


FIGURE 79. STRUCTURAL EFFICIENCY OF ADHESIVE-BONDED DOUBLERS (SIMPLY-SUPPORTED EDGES)



$$\frac{l_{effective}}{t_{skin}} \sqrt{\frac{\sigma_{max}}{E} \, 12(1 - \nu^2)}$$

FIGURE 80. STRUCTURAL EFFICIENCY OF ADHESIVE-BONDED DOUBLERS (BUILT-IN EDGES)



APPROXIMATE ANALYSIS MODEL FOR ADHESIVE STRESSES (REF. 2)

FIGURE 81. ADHESIVE STRESSES IN BONDED DOUBLERS

11.5.1 Adhesive Stresses at Discontinuities and Cracks in the Adherends

Analysis methods, reported in Reference 5, have been developed for adhesive bond stresses associated with structural discontinuities such as broken members, stiffener runouts and the notching of one stiffener where two intersect. The scope of these analyses is given in Section 10.6.3. The solutions can be used to establish acceptable combinations of stiffener spacing, sheet gage, stiffener area, bond width, and adhesive properties. Such solutions are simple enough to evaluate on pocket electronic calculators, yet show good agreement with the available experimental evidence.

11.5.2 Adhesive Stresses Due to Pressure Pillowing in Stiffened Cylinders

In pressurized stiffened cylinders; e.g., transport aircraft fuselages, the skin pillows between the stiffeners and frames, Figure 61. This pillowing causes peel stresses in the adhesive at the edge of the bonded stiffener. Fatigue resistance in this case may be increased by: (a) decreasing the distance between stiffeners, which decreases the deflection of the skin, or (b) increasing the flexibility of the base of the stiffener by lengthening it and/or tapering it as shown in Figure 26.

Parametric residual strength analyses based on Linear Elastic Mechanics for the cracked sheet with bonded stiffeners can be used to determine the maximum allowable stiffener spacing, when assessed in terms of the metal, rather than the bond. The method is described in the following section.

11.5.3 Effect of Bonded Stiffening on the Life and Residual Strength of Cracking Adherends

Linear Elastic Fracture Mechanics methods, as described in Reference 6, are currently being used to analyze cracks in the adherends of adhesive bonded structure. The basic model consists of a symmetric (skin) crack growing from a through-the-thickness flaw in an infinite sheet. A fundamental assumption made is that the local stress conditions at the crack tip are defined by the local stress intensity K, where:

$$K = \sigma\sqrt{\pi a}\ \beta_n$$

σ = gross area stress remote from the crack tip, psi

a = half crack length, inches

β_n = modification factors

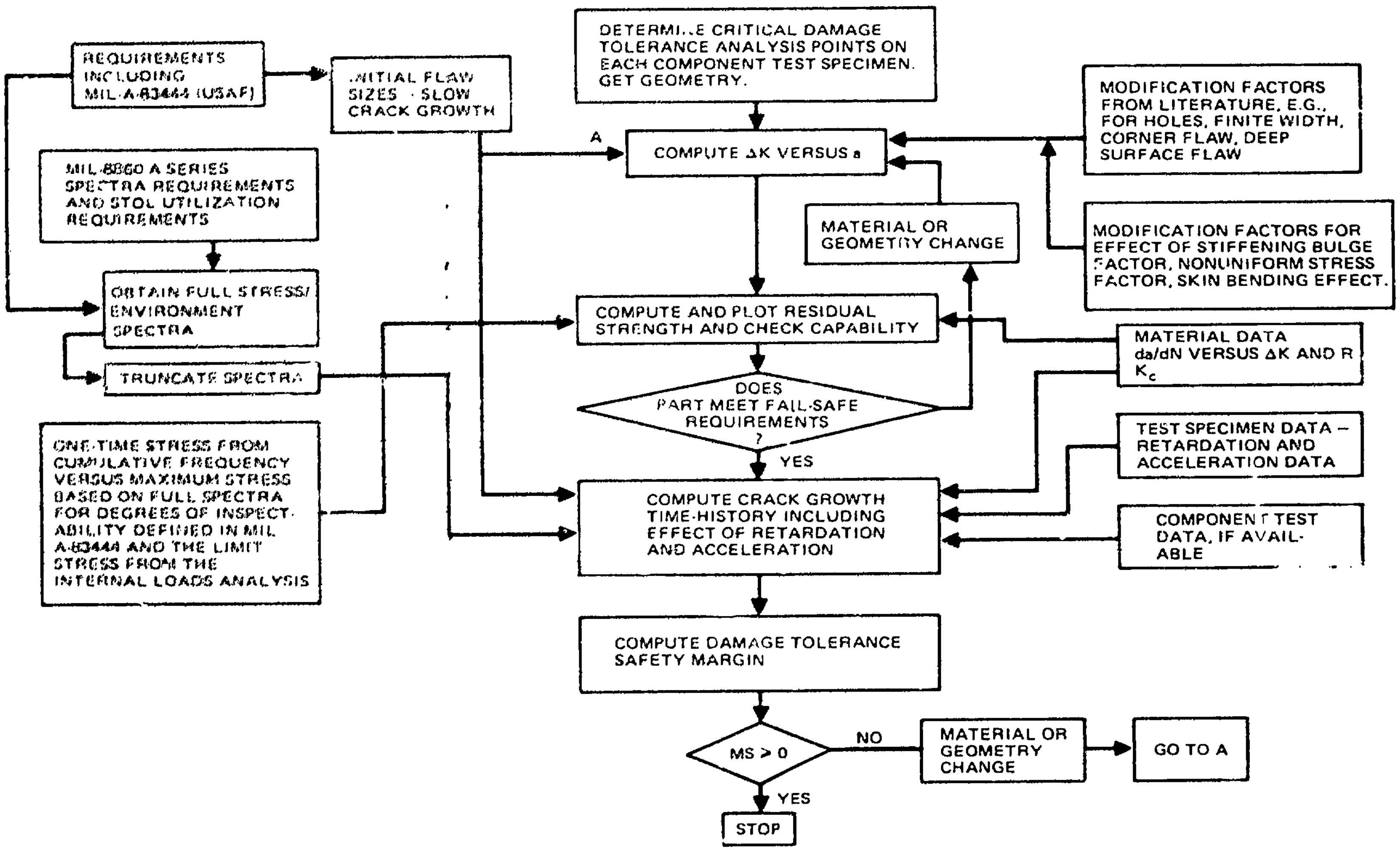


FIGURE 82. DAMAGE TOLERANCE ANALYSIS FLOW CHART FOR METAL STRUCTURE

MODIFICATION FACTOR (β) ACCOUNTS FOR THE EFFECT OF STIFFENING ON A CRACKED SHEET

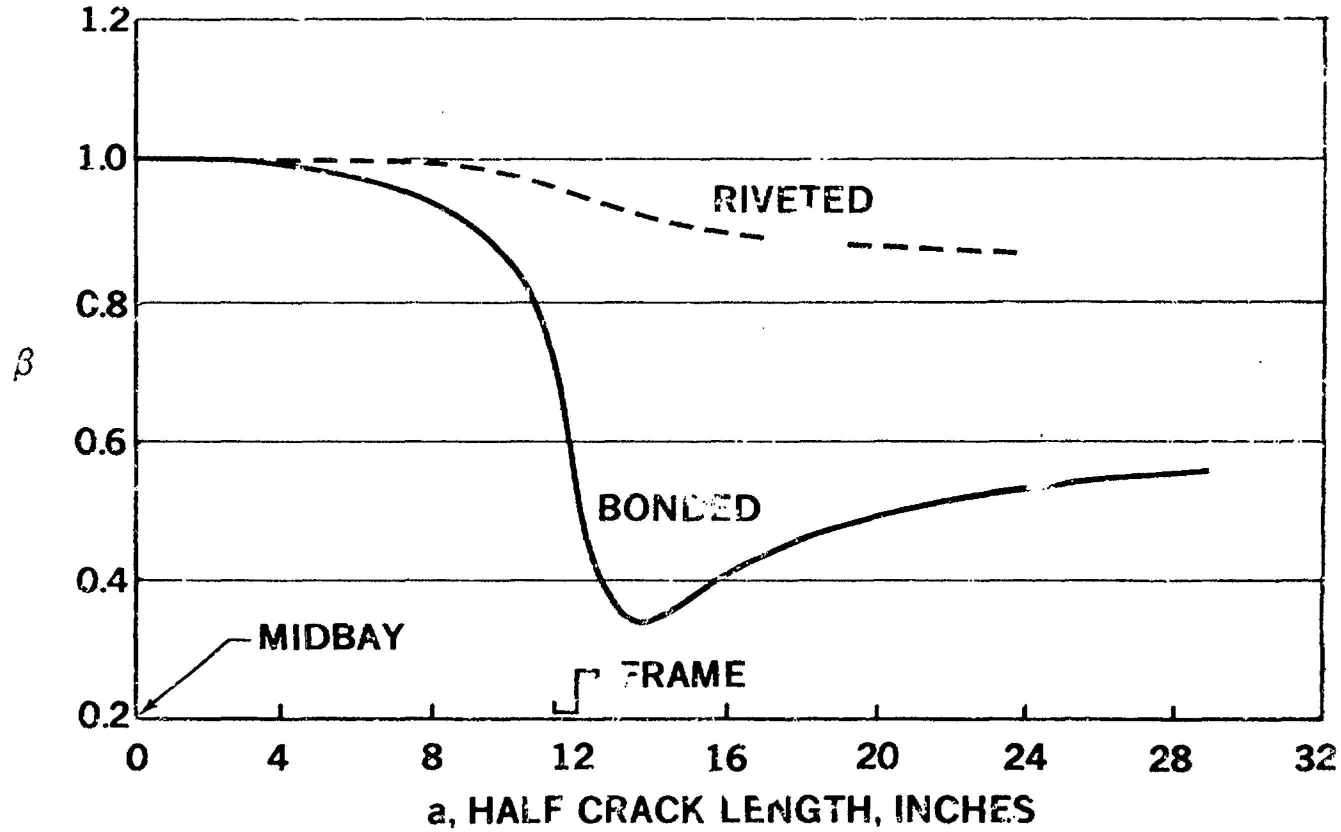


FIGURE 63. EXAMPLE OF IMPROVEMENT PROVIDED BY BONDING

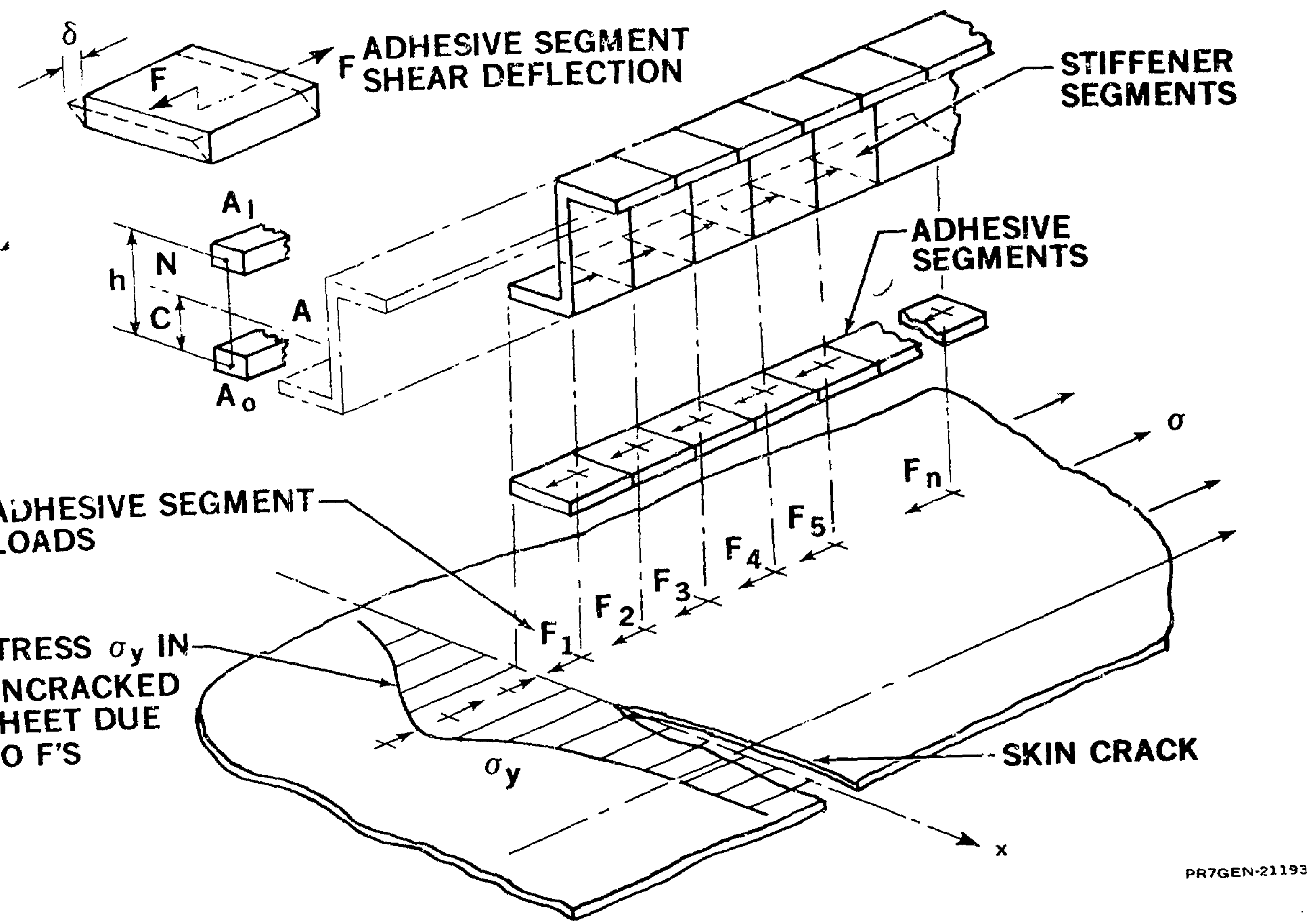
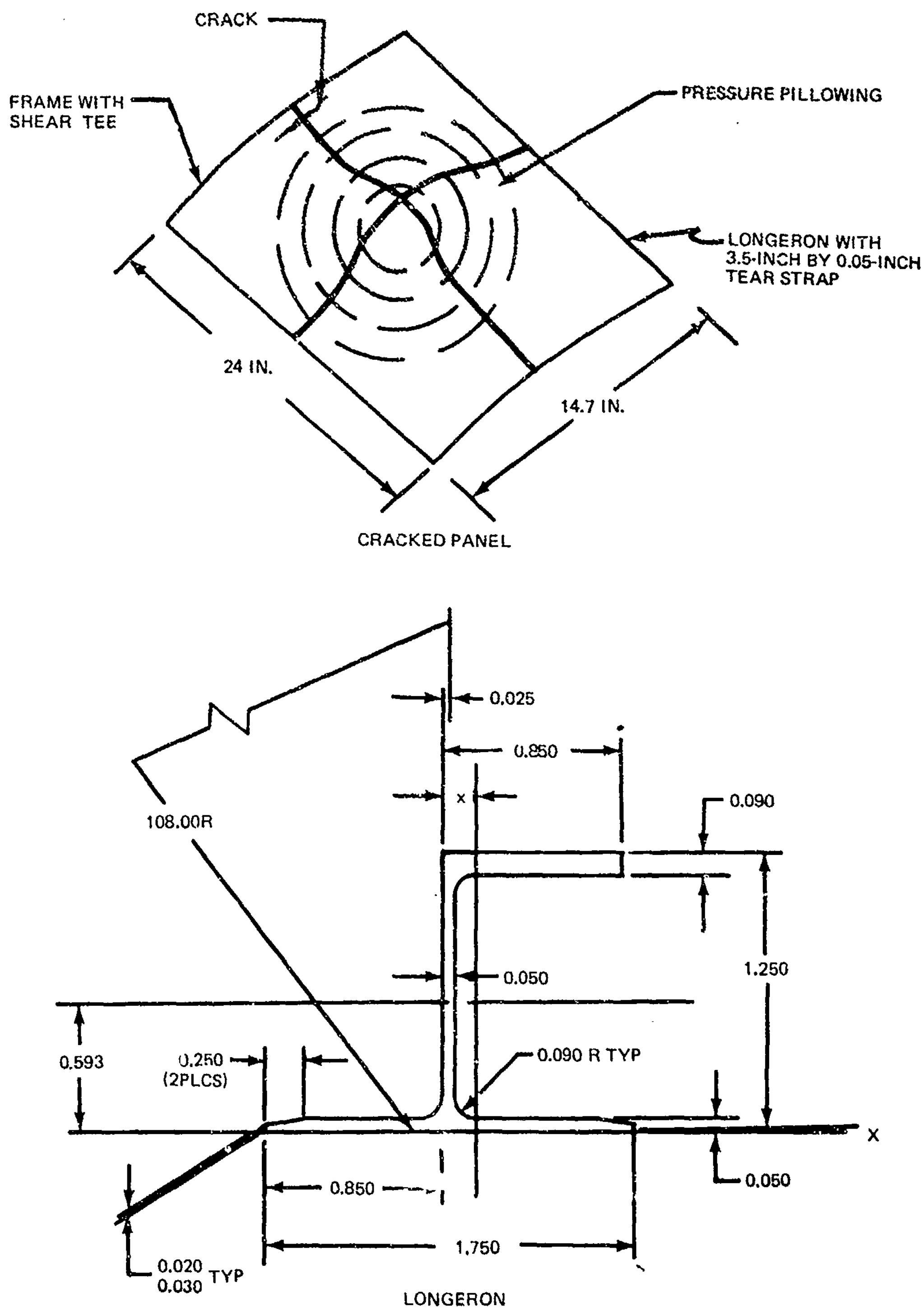


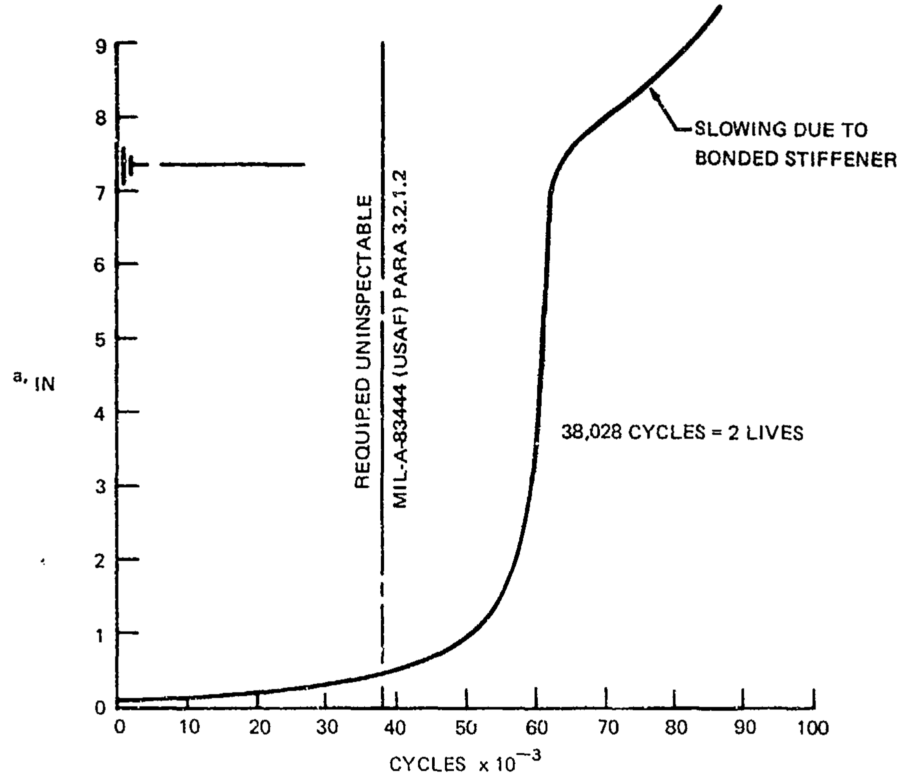
FIGURE 84. ADHESIVE MODELED AS BLOCKS.

Results from a PABST analysis are presented here to illustrate the effect of bonded stiffening on a cracking fuselage skin. The 30,000 hour PABST 108-inch-radius fuselage was pressurized to a maximum Δp of 7.15 psi. The 2024-T3 bare skin at the check point noted in Figure 85 was 0.057 inches thick and was stiffened by internal longerons with crack stoppers, Figure 85 (a) and by frames on shear-tees. The circumferential one bay initial flaw, assumed per MIL-A-83444 (USAF), was a 0.25 inch through-the-thickness surface flaw. The crack growth time history is shown in Figure 85 (b). The longeron, the tear strap, and the adhesive layers remained elastic. Note the considerable slowing of the crack at the bonded stiffening. The residual strength diagram for 15 inches of foreign object damage cutting through stiffening and skin is shown in Figure 85 (c). The applied stress accounts for pressure pillowing. The bonded stiffening takes a large amount of load from the cracking skin; more than for riveted construction.

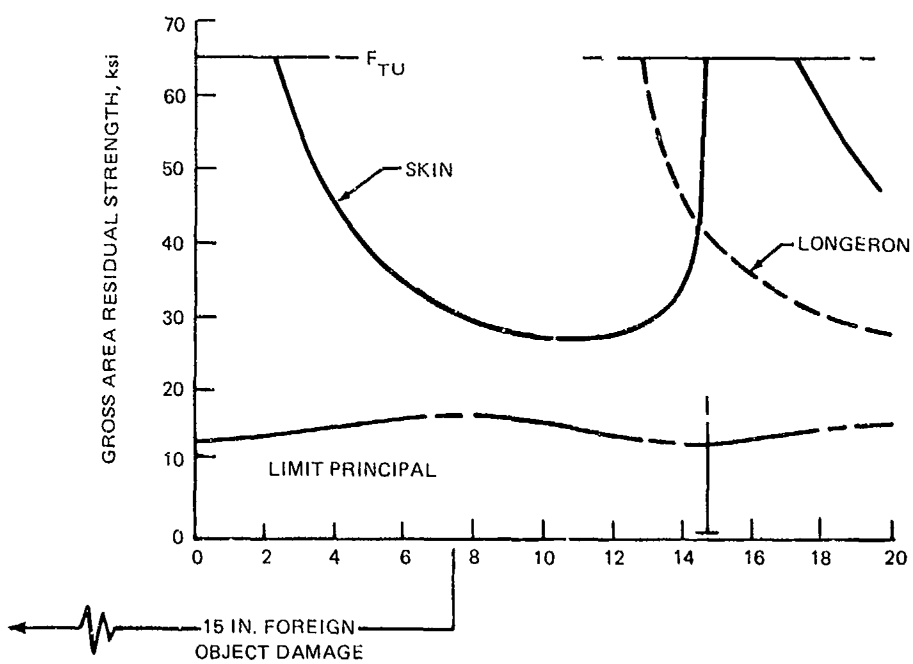


(a) GEOMETRY

FIGURE 85 . PABST DAMAGE TOLERANCE EXAMPLE



(b) CRACK GROWTH TIME HISTORY



(c) RESIDUAL STRENGTH FOR FOREIGN OBJECT DAMAGE

FIGURE 85 . PABST DAMAGE TOLERANCE EXAMPLE – CONCLUDED

SECTION XII
TESTS

A development test program for process and material properties characterization, and verification and substantiation is more critical for bonded structure design than for riveted structure, mainly because of the greater number of design/analysis test variables present. It is strongly recommended therefore, that the test program be initiated early in the design phase; i.e., concurrently with preliminary design. Besides reduced properties in the metal due to cure cycle temperatures, failures can occur between the metal and the surface treatment film (anodize film), between the anodize film and the primer, between the primer and the adhesive, in some cases in the adhesive itself, and lastly, a failure in the protective coating which could cause early failure due to exposure to the environment. Each of these variables contain many variables in themselves. For instance, there are several anodizing processes to choose from and each has processing tolerance variables such as anodizing time and solution content. Also there are many primers, adhesives, and coatings available that are not necessarily interchangeable with respect to performance. In addition to all these variables, the usual qualifying tests of the structure must be performed: static load, durability and damage tolerance. The decision to test or rely on the analysis should be made by considering the critical nature of the structure which is no different for riveted structure.

This section describes the testing that may be involved in an adhesive bonded structure development program. Testing done during the PABST program is documented in References 4 and 8.

12.1 Small Specimen Tests

Early in the design of bonded structure some basic decisions must be made concerning adherend material, adhesives, surface treatments, primers, coatings, non-destructive testing (NDT) methods, and environmental and cyclic testing methods. Appropriate data for making these decisions may be obtained from small specimen tests some of which are illustrated in Figure 86.

12.1.1 Selection of an Adhesive System

All of the small specimen tests illustrated in the figures may be used to select an adhesive system that consists of the adherend material, surface

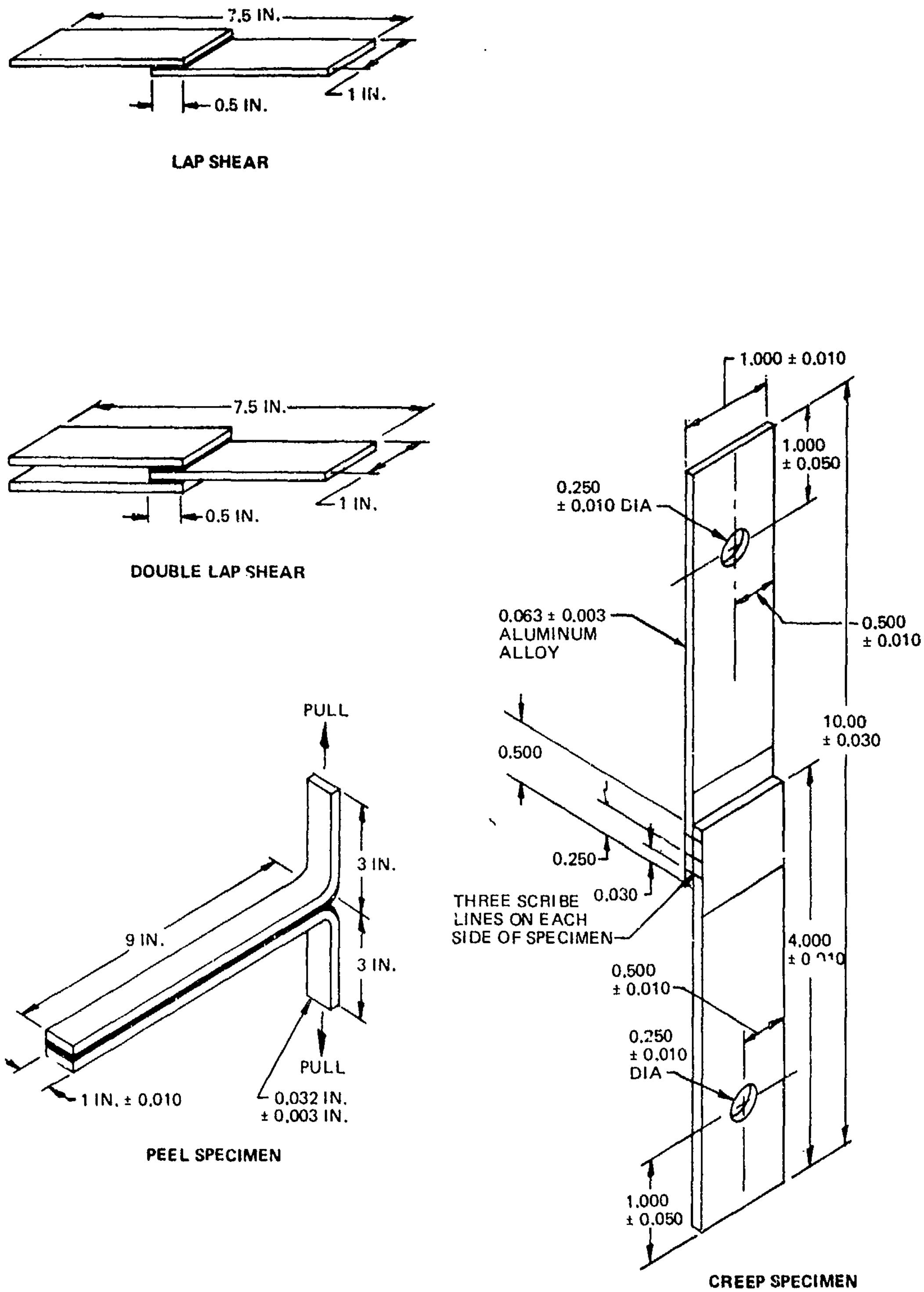


FIGURE 86. TEST SPECIMEN CONFIGURATIONS

treatment, primers, adhesive, and protective coating. A test plan that determines by comparison the best product for each of these variables independently would produce misleading data that would probably result in an inferior adhesive system. The fact is that the variables do not perform independently of the other variables. For instance, some adhesives work well with some primers and not with others, and the performance of the protective coating may be related to the surface treatment used. A complete test plan should include all combinations of these variables including environmental variables; i.e., temperature, humidity, salt air, etc.

12.1.2 Non-Destructive Testing Methods

A test program must be designed to correlate non-destructive test methods with actual flawed or low bond strength specimens. The program may include the specimens used in selecting an adhesive system, but should also include specimens that resemble the structural design being investigated. For this case, large specimens used for qualifying the structure may be monitored for flaw growth during testing and then cut up into shear, peel and wedge crack specimens after testing to correlate bond strength with non-destructive test indications. Reference standards with intentional flaws may also be required to compare NDT methods. Two methods for fabricating these standards are shown in Figure 87.

12.1.3 Environmental and Cycle Evaluation Tests

Simulating the environment is an important part of the test program. Temperature, humidity, and contact with corrosives such as fuel, bilge fluid, salt air, etc., must all be introduced into the tests to determine the critical conditions. If a structure is designed to last 20 years, it is very difficult to simulate the effect that environment will have had on the bondline; however, some information may be obtained by fabricating test specimens from bonded structure that has been in service already.

The fatigue life of a bonded structure is dependent on the cycle; i.e., load off time, time to load on, load on time, time to load off, etc. It has been noted in RAAB tests (Figure 88 and 89) of some adhesive systems that while the usual 30 Hz produces no failures in a specimen for a certain cycle life, a

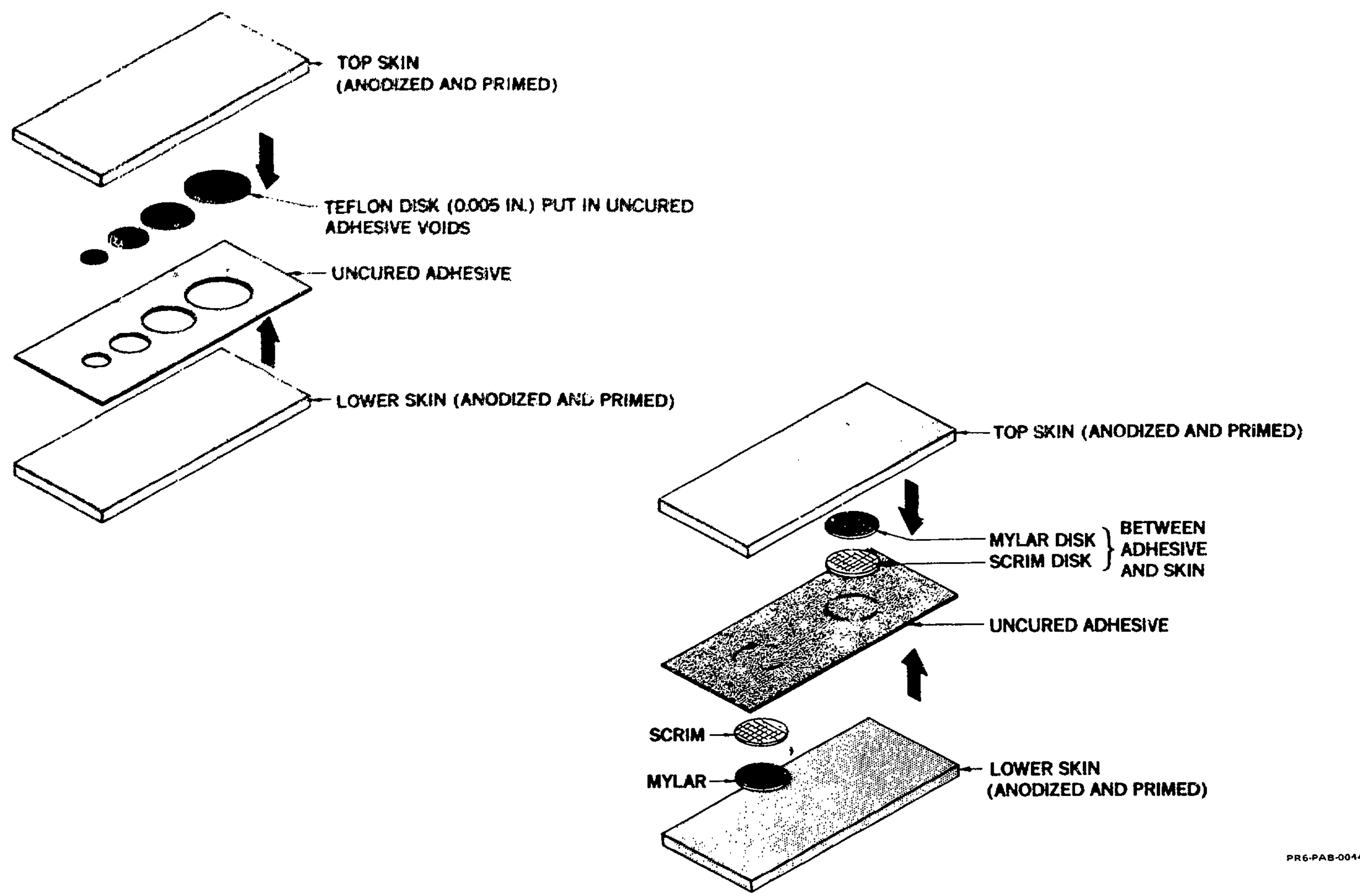


FIGURE 87. METHODS FOR FABRICATING REFERENCE STANDARDS

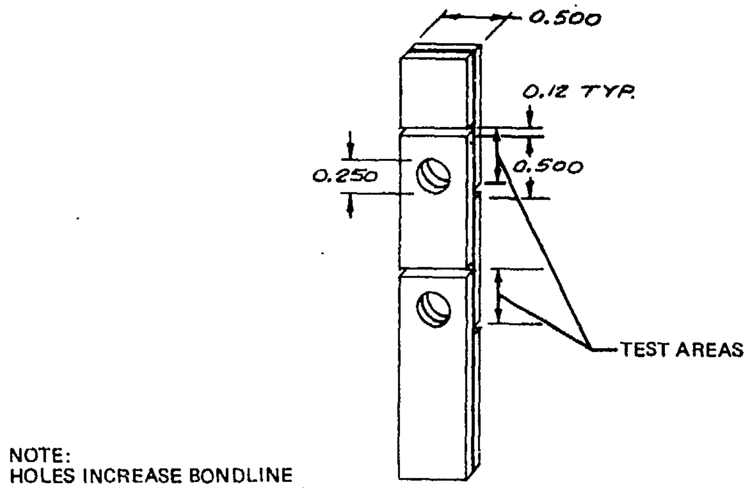
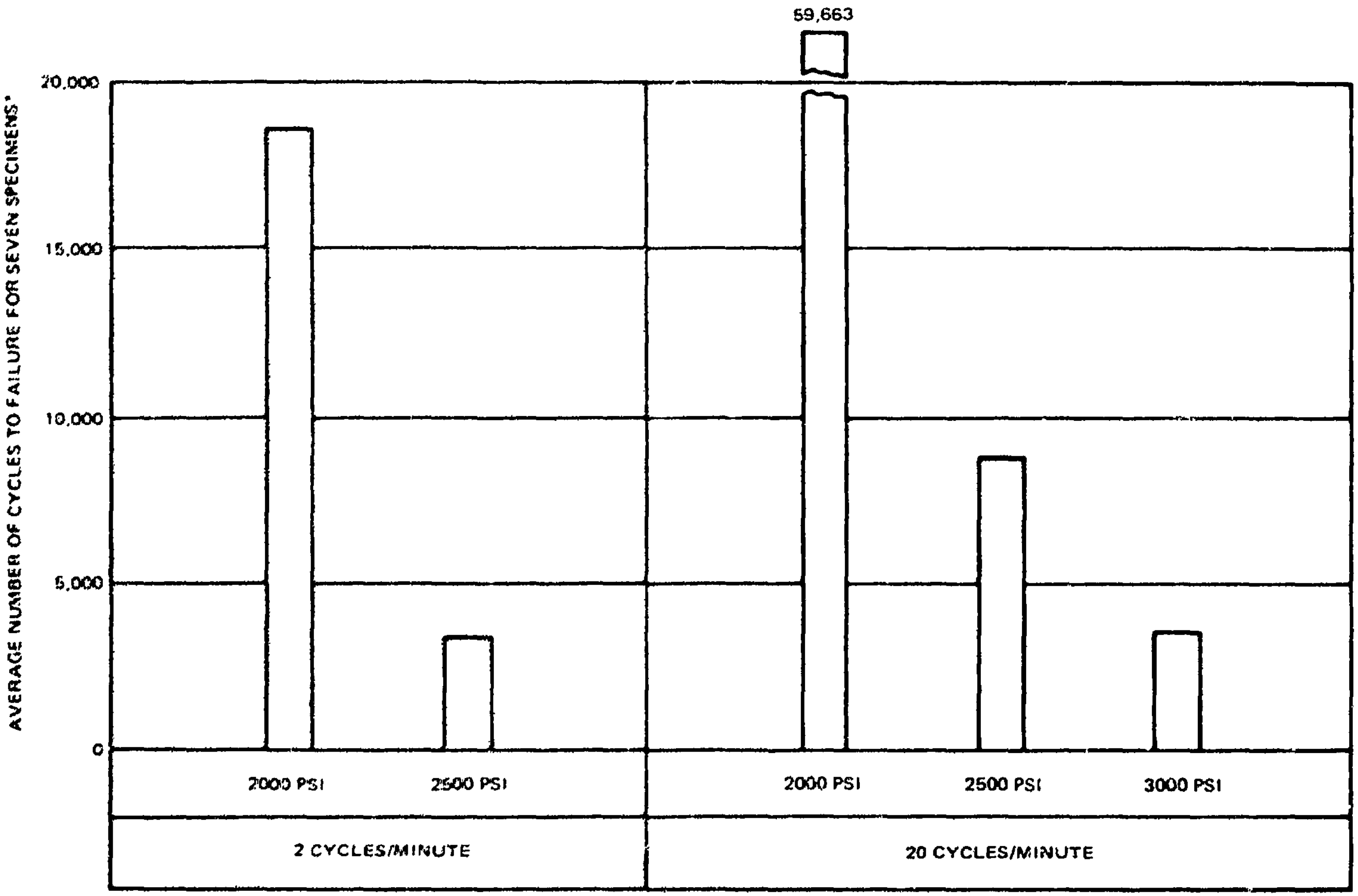


FIGURE 88 . REDUCED AREA ADHESIVE BOND (RAAB) LAP SHEAR, TWO TEST AREAS SHOWN



*CONSTANT AMBIENT ENVIRONMENT, ONE ADHESIVE SYSTEM

FIGURE 89. RAAB SPECIMEN TEST CYCLE DETERMINATION

longer cycle with longer load off, load rate, and load on times will produce earlier fatigue failures in the bond, especially at elevated temperatures. Test cycles should, therefore, be designed to represent the real life cycle of a structure as nearly as possible or practical.

12.1.4 Miscellaneous Small Specimen Tests

The designer must be aware of peculiar areas of the design that may require testing. For example, many areas of a bonded panel, including mechanical splice areas, have fasteners installed through a bondline. It is necessary to determine the effect that countersinking or drill speeds and feeds will have on the bondline.

12.2 Structural Integrity Tests

Structural integrity tests are intended to support analyses, to verify that the structural configuration will satisfy the design criteria, and to optimize the structure with regard to cost and weight. Early in the design formulation period some fundamental test data may be needed. These tests may be used where failure to meet some basic design criteria by the intended structure would have a great impact on the design. For instance, if a bonded skin splice could not meet the fatigue or damage tolerance criteria for a structure, then all the bonded assemblies in a structure could not be larger than a single piece of skin material. Therefore, it would be necessary to test some bonded splices in the expected environment before the design becomes very firm.

Crack growth rates, shear and interaction allowables, or the effects of pressure for different stiffener and skin combinations may be needed to verify or supplement the analyses. The panels for these tests must be large enough to ensure realistic loads in the test area. Ultimately, it would be desirable to perform a full scale article test where all the criteria could be verified.

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