## 3.4 CONNECTIONS

There are four basic systems that are commonly used to join sheet steel body members with other members: welding, mechanical fasteners, mechanical fastening by deformation of the parent metal and adhesive bonding. Each system offers advantages and disadvantages, which are discussed in Sections 3.4.1 through Section 3.4.6. In some cases, the systems are combined to optimize the performance of the joint, such as weldbonding, which is discussed in Section 3.4.7.

The following checklist will assist in identifying the factors to be weighed in selecting the optimum system.

1. Assess the Application and Operating Environment.

    1.1 Determine the type and intensity of applied load.

    Type of load is essential; typical types are long-term or continuous, occasional, fatigue, and impact (high strain rate).

    Sources may be anticipated or accidental. Anticipated loads, which are usually quantified, include those computed from suspension G loads; low-speed bumper impacts; rough road tests; and door, hood and deck lid slam.

    Accidental loads, which are not readily quantified, include high speed collision, rollover, and low speed (parking lot) collisions.

    1.2 Assess potential problem areas

    Potential problem areas may include long term structural degradation, potential for rattles, galvanic and atmospheric corrosion, differential rates of thermal expansion, and repairability.

    1.3 Design the joint

    Select the system or combination that will meet the design requirements.

    Evaluate the shape and size of the contact areas of the members to be joined and the relationship between the type and direction of load and the joint configuration.

    1.4 Determine the side benefits

    Examine the side benefits that may be derived, such as galvanic separation, moisture barrier, and effects of load distribution that may eliminate or require local reinforcements and stiffeners on all affected members.

    1.5 Review the cosmetic requirements

    Witness marks are generally prohibitive on appearance surfaces, but may be tolerable on other surfaces.

1.6 Review the effect on materials

Check the compatibility of joining method on materials, such as possible degradation of properties developed by heat treatment, capability for joining dissimilar materials, and toleration of pre-coated stock.

2. Determine which system(s), alone or in combination, is acceptable.

The joint must be accessible to equipment such as weld guns, adhesive application systems and screw drivers.

3. Confer with manufacturing engineers (and suppliers of adhesives or fasteners where they are alternatives).

3.1 Determine the cost of modifying plant operations to accommodate the fastening system.

3.2 Assess the effects on the components to be joined in terms of mass, material cost and process cost. Material cost includes the cost differential for the material required to fabricate the components and the cost of any fasteners or adhesives that are employed.

3.3 Estimate operating costs for each process, such as the cost of energy, maintenance, floor space required and amortization.

4. Determine all of the implications for each option on the entire vehicle and select the optimum system for the joint.

## 3.4.1 WELDED CONNECTIONS

The great majority of welded connections in automotive structures are spot welds. This section will therefore be confined to a discussion of the strength of spot welded joints. Weldability, a description of the spot welding process, and quality control of spot welding are described in Section 4.3-1 and in Reference 1. The spacing of spot welds to achieve structural criteria is presented in Section 3.4.4.

The strength that a spot weld, or a pattern of welds, will exhibit in a particular application can be predicted to some extent from tabulated data compiled from test specimens. Welded joints in real world applications will rarely behave like the test specimens. Therefore, proper application of the data requires a basic understanding of spot weld testing methods.

Spot welds are tested in a variety of ways in order to evaluate their mechanical properties in various modes of loading. These tests include the peel, chisel and a variety of tension, torsion and fatigue tests.

### 3.4.1.1 Peel Test

The peel test is the most commonly used mechanical test for spot welds. It is widely accepted because of the following factors:

1. Ease of performance
2. Low cost
3. The ability to use it on the shop floor for quality control

This test is performed by making two welds on a sample. The first weld serves as a shunt weld to carry part of the current, so the second weld is a more realistic example of what is found in the manufactured automobile. For the purposes of the test, the second spot weld is peeled to destruction.

Research in evaluating the factors that affect the peel test (Figure 3.4.1.1-1) has demonstrated that the smaller the distance between the spot weld and the pulling tab, the greater the maximum load.

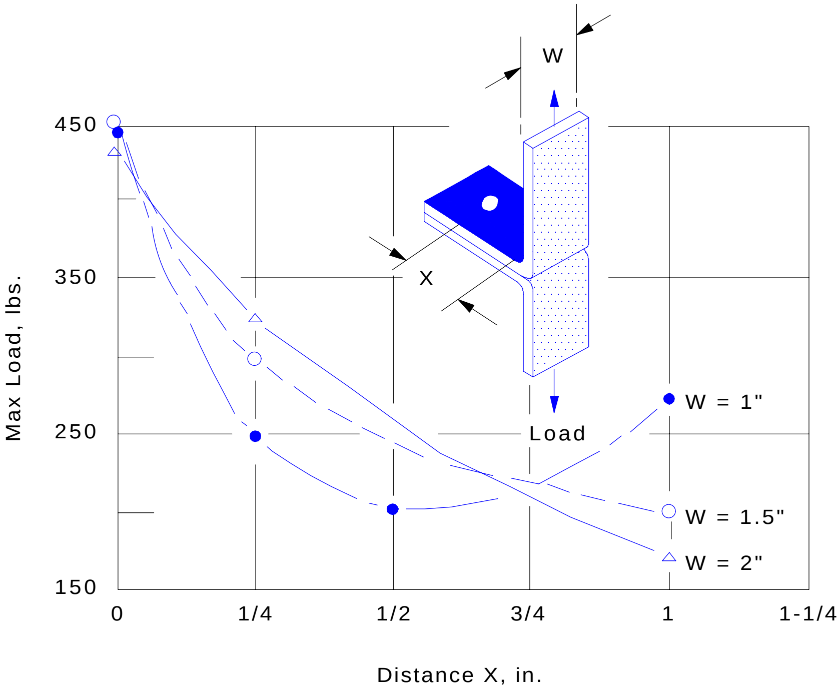


**Figure 3.4.1.1-1** Effect of geometry on quantified peel test

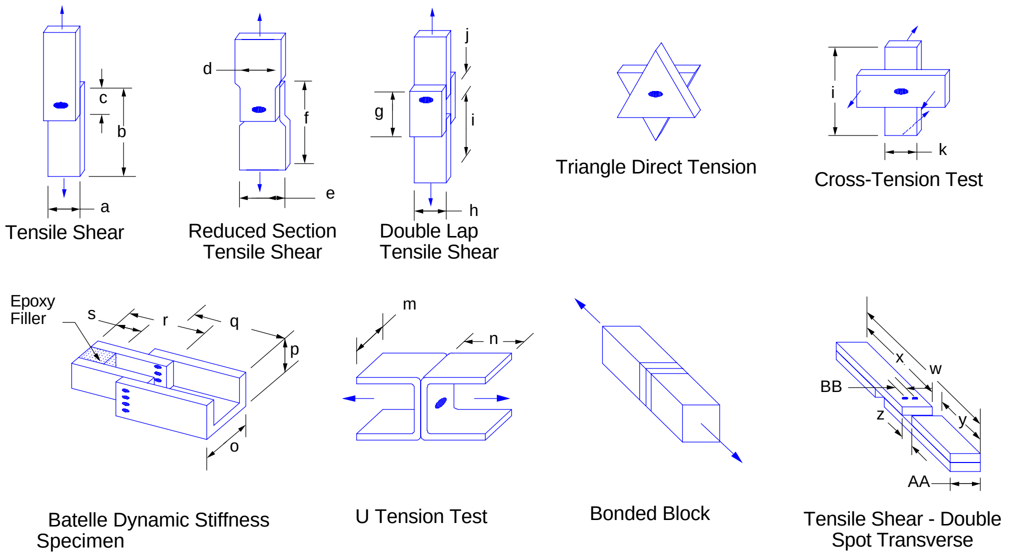
Thus, the results generated may be somewhat operator dependent. It has also been observed on marginally acceptable HSLA materials, that conducting the peel test while the specimen is still warm from welding promotes nugget pullout. In contrast, testing identical specimens after cooling results in a partial fracture of the weld metal. Even with these findings, little work has been done to characterize and qualify the peel test data.

### 3.4.1.2 Chisel Test

Chisel testing is similar to the peel test. The difference is that forcing a chisel, or wedge, between the sheets near the spot weld location may more easily produce an interfacial failure. The stress distribution at the weld for chisel or peel loading is illustrated in Figure 3.4.1.3-2(a).

### 3.4.1.3 Tensile Tests

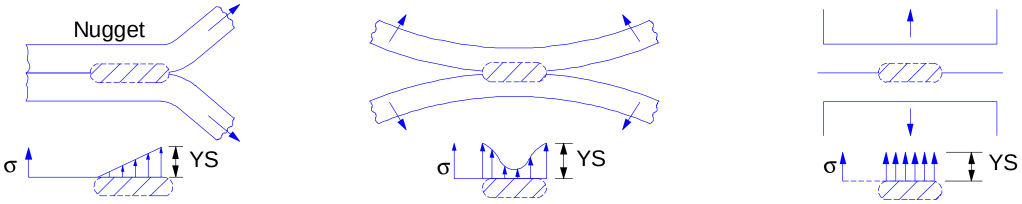
Recently, tensile tests have become more popular in the automotive industry. Numerous tensile test specimen geometries (Figure 3.4.1.3-1) have been utilized to determine tensile properties of spot welds.



| | DIMENSION RANGE (INCH) |
|---|---|
| a | 0.625-2 |
| b | 3-6 |
| c | 0.625-2 |
| d | 0.3625-4 |
| e | 0.72-12 |
| f | 0.81-13.5 |
| g | 1-2 |
| h | 1 |
| i | 4 |
| j | 0.25-0.75 |
| k | 2-3 |
| l | 4-8 |
| m | 1-2 |
| n | 1-2 |
| o | 1 |
| p | 1 |
| q | 30 |
| r | 8 |
| s | 6 |
| w | 180 mm |
| x | 110 mm |
| y | 60 mm |
| z | 10 mm |
| AA | 72 mm |
| BB | 32 mm |

**Figure 3.4.1.3-1** Various tensile test specimens

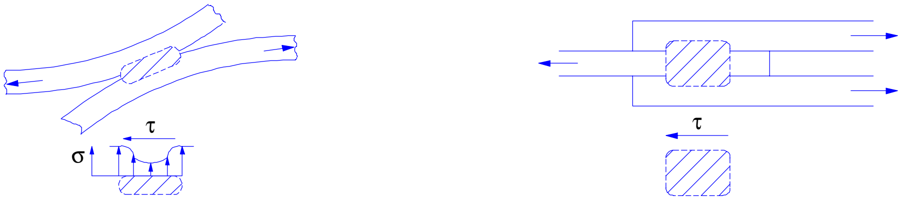
Stress distribution in spot welds of various test geometries is illustrated in Figure 3.4.1.3-2.



a) Stress distribution on nugget under peel/ chisel loading (flange loading)

b) Stress distribution on nugget under cross tension/U tension loading

c) Stress distribution on nugget under bonded block loading



d) Stress distribution on nugget under standard tensile shear mode

e) Stress distribution in the double lap shear specimen

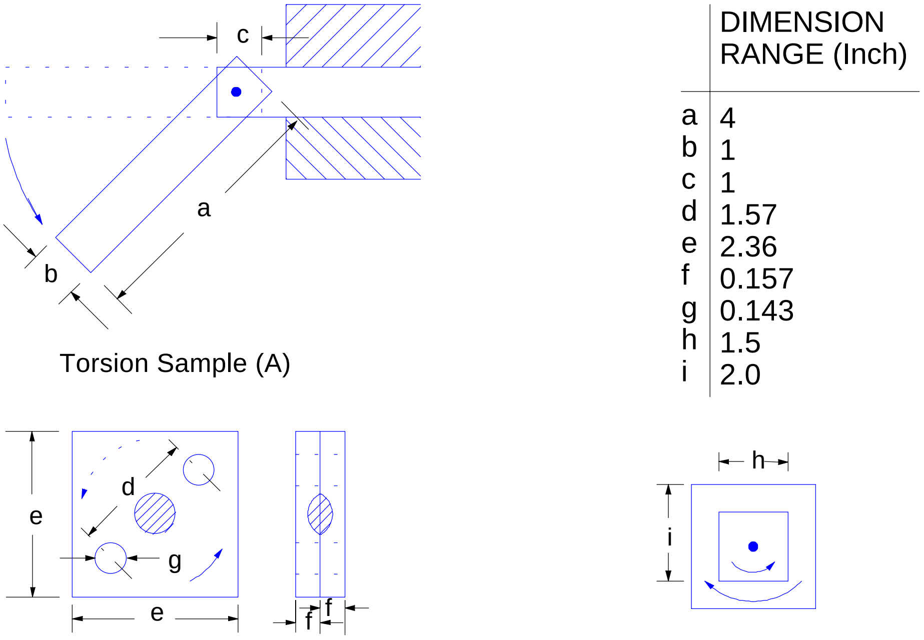
**Figure 3.4.1.3-2** Stress distribution around spot welds of various test geometries

Specimens consist either of those designed to measure normal stress or those that measure shear stress. The lap shear or tensile shear specimen is made by lapping one piece of material over another piece, making a spot weld, then pulling the pieces in opposite directions until failure. Other types of specimen designs include a normal bonded block type specimen, although this is not used very often. Normal tensile tests include the cross weld test and the U-tensile test.

The important finding of test data is that welds made in different geometries have different strengths. For example, the lap shear geometry carries the highest load. Besides specimen geometry, other factors that affect spot weld strength include sheet size, nugget size, weld time, hold time, spot array, coatings, and testing procedures.

### 3.4.1.4 Torsion Testing

The torsion test (Figure 3.4.1.4-1) is also used to evaluate spot weld properties.



| | DIMENSION RANGE (Inch) |
|---|---|
| a | 4 |
| b | 1 |
| c | 1 |
| d | 1.57 |
| e | 2.36 |
| f | 0.157 |
| g | 0.143 |
| h | 1.5 |
| i | 2.0 |

Torsion Sample (A)

Torsion Sample (B)

Torsion Sample (C)

**Figure 3.4.1.4-1** Torsion test specimens

The angle of twist before nugget failure has been used as a measure of spot weld ductility, while the maximum torque has been used to evaluate spot weld strength. Generally, most torsion tests are similar. Although popular in the past, these tests are used less today. They are, however, useful in characterizing various types of steels. Usually, good forming steels that have high ductility have low torques, with large twist angles. In contrast, high strength steels tend to have higher torques with lower twist angles. When a weld is made with some expulsion, it tends to reduce both the torque and the twist angle. All the factors that affect tensile tests also have been observed to affect torsion testing.

### 3.4.1.5 Fatigue Testing

Fatigue testing of spot welds has gained considerable interest in recent years. However, there are unanswered questions about the factors that affect fatigue testing of spot welds such as the effect of specimen geometry, sheet size, nugget size, weld time, hold time, spot array and composition.

One major point concerning fatigue testing is illustrated in Figure 3.4.1.5-1.

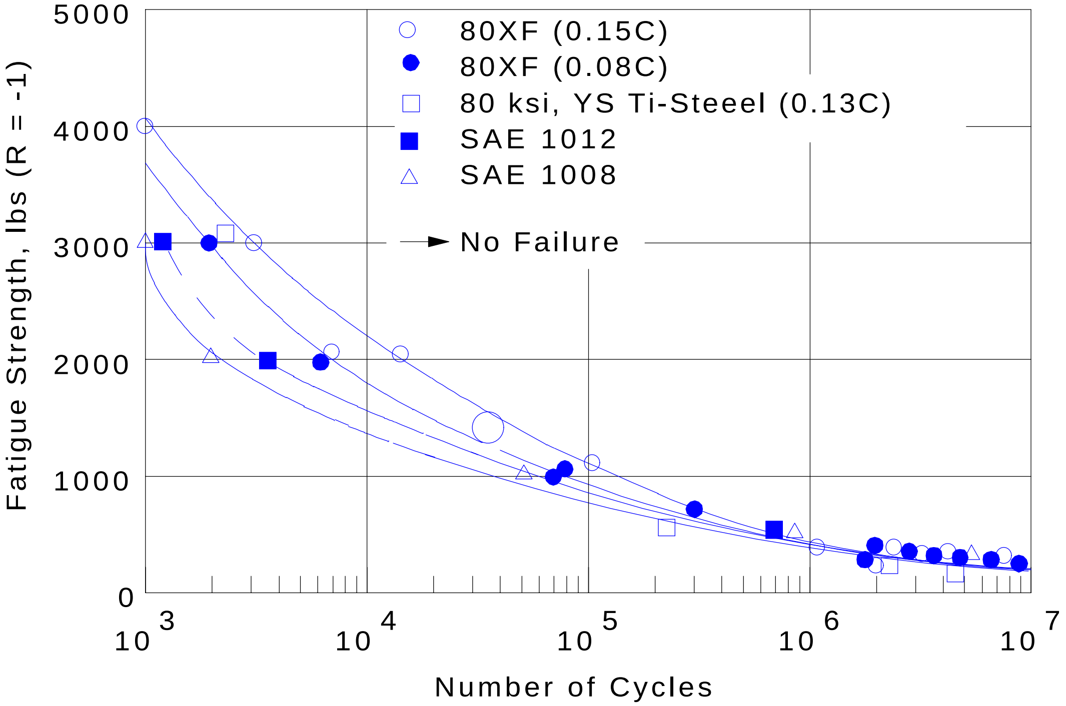


**Figure 3.4.1.5-1** Fatigue endurance curves; high strength and low-carbon steels

The line with triangles indicates SAE 1008 standard carbon steel in comparison with high strength low alloy. At a low number of fatigue cycles, the high strength low alloy material has a definite advantage in fatigue strength. At high cycles (between $10^6$ and $10^7$), this advantage appears to be reduced. Thus, a concern in the auto industry is the small difference in the fatigue strength at high cycles between high strength steels and plain carbon type steels.

Another observation concerns the critical diameter of a spot weld. Once a critical diameter is reached, the failures behave in a normal crack mode, with the crack growing through the sheet; that is, the nugget is pulled out. If the diameter of the nugget is smaller than this critical size, rather than pulling out a nugget during fatigue testing, the result is an interfacial failure.

### 3.4.1.6 Shear Strength Values

Conservative shear strength values have been used to provide allowable shear values in the 1980 AISI specification[2] for spot welds on various sheet thicknesses. The allowable values employ a factor of safety of 2.5. Somewhat less conservative shear strength values, which are shown in Table 3.4.1.6-1, form the basis of the allowable shear values in the 1986 AISI specification[3].

**Table 3.4.1.6-1** Resistance weld capacity.
Ultimate shear strength per spot $V_{su}$

| Thickness of thinnest outside sheet (inches) | $V_{su}$ (kips) | Thickness of thinnest outside sheet (mm) | $V_{su}$ (kN) |
|---|---|---|---|
| 0.010 | 0.130* | 0.25 | 0.58 |
| 0.020 | 0.437 | 0.51 | 1.94 |
| 0.030 | 1.000 | 0.76 | 4.45 |
| 0.040 | 1.423 | 1.02 | 6.33 |
| 0.050 | 1.654 | 1.27 | 7.36 |
| 0.060 | 2.282 | 1.52 | 10.15 |
| 0.070 | 2.827 | 1.78 | 12.57 |
| 0.080 | 3.333 | 2.03 | 14.82 |
| 0.094 | 4.313 | 2.39 | 19.18 |
| 0.109 | 5.987 | 2.77 | 26.63 |
| 0.125 | 7.200 | 3.18 | 32.03 |
| 0.188 | 10.000* | 4.77 | 44.48 |
| 0.250 | 15.000* | 6.35 | 66.72 |

* From AWS C1.1-66 Tables 3.1-1 and 3.1-3. Remainder of values either taken from or interpolated from AWS C1.3-70 Table 2.1.

These values are applicable to all structural grades of low carbon steel, but also give conservative values for medium carbon and low alloy steels as well. These values do not consider capacity reductions due to fatigue loading.

### 3.4.1.7 Tensile Strength of Arc Spot Welds

The tensile strength of arc spot welds was recently studied at the University of Missouri-Rolla[4, 5]. Results of more than 260 welded connection tests indicate that the primary parameters that influence the tensile strength of an arc welded connection are the thickness of the sheet, the diameter of the weld, and the tensile strengths of the sheet and weld. Design recommendations were developed for predicting the tensile strength of arc spot welds.

## 3.4.2 MECHANICAL FASTENERS

Where access to both sides of a joint is available, resistance spot welding is the primary method of joining sheet steel. However, mechanical fasteners have been successfully used when:

- Anti-corrosion coatings or adhesives are incompatible with spot welding
- The steel is too thin to be reliably spot welded
- A neat, flash free appearance is desired
- Spot welds are too close together, causing shunting

Where after-paint repairs of spot welds are required, or where only single side access is available, blind or pop rivets or self-piercing and threading screws are used.

Mechanical fasteners are conveniently divided into two sided and one sided. Two sided fasteners, typically a bolt and nut, require two parts at each location. One sided fasteners, such as self tapping screws, require one part per location.

Fasteners are available in a wide variety of grades, head styles, threads, lengths, locking devices, and other design variations. The number of different fasteners, which tends to proliferate within a company, can become enormous. Many companies are recognizing the cost incurred for each fastener type (and part number) that is on their books, and are taking significant steps toward reducing the number of different fasteners within their system. Designers are being encouraged to consult company standards and specify fasteners within those standards whenever possible.

### 3.4.2.1 Two sided Fasteners

Two sided fasteners, typically bolts, are commonly employed to connect thicker sheet gauges. The nuts may be transported in bulk to the point of assembly or attached to one of the members by means such as projection welding. In some cases bolts are threaded into tapping plates, which are thick plates with tapped threads welded to one of the members. Although the SAE standard for hex bolts[6] includes coarse, fine and 8-thread series, the automotive industry now uses coarse threads almost exclusively.

The strength classification system used by SAE for bolts and machine screws utilizes grade numbers, with the higher numbers indicating higher strengths[7]. In the systems used for some machine screws[8] and for metric[9], the numbers have quantitative relationships to the strength of the fastener. The system used for nuts, in sizes 1/4 to 1-1/2 in. inclusive, is similar to that used for bolts, but the system includes only three grades[10].

The tensile strengths and proof loads for bolts,[7,9] machine screws[8] and nuts[10], as a function of their size, grade, and thread, have been standardized and published for use in design. Where shear strength is the governing factor, it may be taken at 60 percent of tensile strength.

#### 3.4.2.1.1 Clamping Load and Nut Torque

Some attachments that utilize a nut on a bolt (or stud) require that a specified clamping load be developed and sustained. In those cases, criteria governing clamping loads must be met in addition to the design considerations for mechanical fasteners, which are discussed in Section 3.4.2.3.

The clamping force developed when a nut is torqued on a bolt is determined by the applied torque and the surface condition of the threads and bearing areas. The maximum clamping load that can be developed is a function of the strength of the bolt. Normally the proof load (the maximum load that the bolt can sustain without permanent deformation) is specified. When selecting a nut and bolt to develop a specified clamping load, the nut should be of the same or higher grade than the bolt. For example, either a grade 5 or a grade 8 nut can be used with a grade 5 bolt, but a grade 2 nut should not be used.

The clamping load is related to the tightening torque by the formula

$$T = kPd$$

**Equation 3.4.2.1.1-1**

where T = torque applied to the nut
k = a constant that depends on the surface coating and type of lubrication (if any) on the threads and bearing areas. Typical values are listed in Table 3.4.2.1.1-1.
P = axial load on the bolt or stud
d = nominal bolt or stud diameter

The value of **kPd** must be divided by 12 when **T** is in lb ft; it must be divided by 1000 when **T** is in **Nm** and **d** is in mm.

**Table 3.4.2.1.1-1** Typical k values for nut torquing

| Type of Surface | | | |
|---|---|---|---|
| Nut | Bolt and Bearing | Dry or Lubricated | k value |
| zinc | steel | dry | 0.22 |
| steel | steel | dry | 0.20 |
| cadmium | cadmium | dry | 0.17 |
| zinc | steel | machine oil | 0.16 |
| cadmium | steel | wax or oil | 0.15 |
| --- | --- | phosphate and oil | 0.12 |
| --- | --- | moly disulphide | 0.09 |

Some manufacturing facilities employ torque-to-yield power wrenches, which detect the point at which the bolt begins to yield and automatically stop. Where this type of wrench is available, a torque-to-yield specification may be used instead of specifying a torque value.

When bolts or machine screws are inserted into tapping plates, the plate can be made thick enough so that the length of thread engagement is sufficient to permit development of the full strength of the bolt. Nuts generally provide sufficient threads to permit development of the full strength of the bolt, provided that the nut is of equal or higher grade than the bolt, as recommended above.

### 3.4.2.2 One Sided Fasteners

One sided fasteners are widely used because they offer a quick, economical means of fastening that does not require backup operations, access to the opposite side of the joint, or opposite side components, such as loose nuts, weld nuts or tapping plates. The types of one sided fasteners most commonly used in automotive bodies are self tapping screws, self-drilling tapping screws, and thread rolling screws. (This discussion will follow SAE J478a and consider the fasteners commonly called sheet metal screws as type A self tapping screws.) One sided fasteners are hardened to levels that enable them to deform or cut the sheets into which they are installed. The required hardness determines the strength of the fastener. For example, thread rolling screws typically exhibit approximately twice the strength of machine screws, and typically 25 percent higher strength than self tapping screws.

### 3.4.2.2.1 Self Tapping and Self-Drilling Tapping Screws

Self tapping screws are installed through holes in the top sheet into holes in the bottom sheet. The holes are of such sizes that the screw thread draws the sheets tight while cutting a thread or forming threads in the bottom sheet. With a self-drilling tapping screw, the hole is cut into the bottom sheet or both sheets by the screw, which then cuts a thread in the sheet or sheets. The clamping force developed by these types of screws is limited by the thickness and mechanical properties of the sheet. The threads form an interference fit in the bottom sheet, which makes them resistant to loosening, depending on the length of thread engagement.

The penetrating capability of a self-drilling tapping screw may limit its usage, particularly in smaller screw sizes. For example, a #6 screw, which is 3.51 mm (0.138 in.) diameter, can drill through sheet up to 2.3 mm (0.09 in.). This penetration amounts to approximately 0.65 times the diameter, or three threads. A #12 screw, which is 5.49 mm (0.216 in.) diameter, can drill through steel up to 13 mm (0.5 in.). This penetration is 2.3 D or 6.5 threads.

### 3.4.2.2.2 Thread Rolling Screws

Thread rolling screws are becoming more useful as sheet metal gauges become thinner to meet the demand for lighter mass vehicles. Thread rolling screws pass through a clearance hole in the upper sheet into an extruded hole in the bottom sheet (Figure 3.4.2.2.2-1).

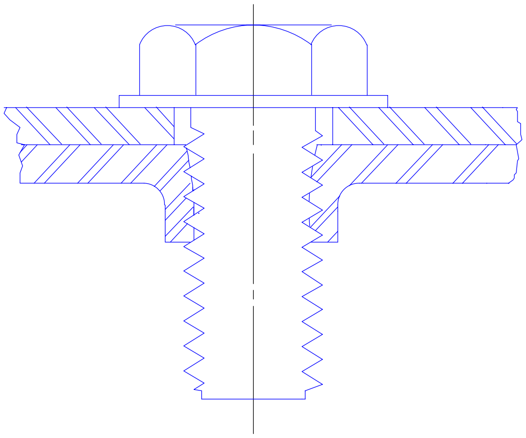
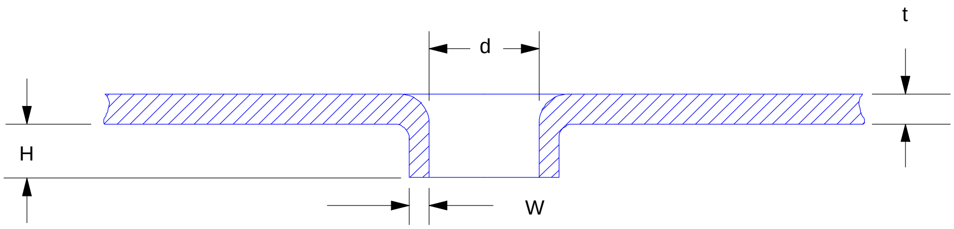


**Figure 3.4.2.2.2-1** Typical thread rolling screw installation

The profile and design dimensions of the thread, combined with the dimensions of the extruded hole, enable the screw to roll a thread into the hole as it is driven. The sheet steels for which thread rolling screws are being specified generally have adequate forming properties to allow the extruded hole to be formed to the required dimensions within required tolerances.

Typical dimensions for extruded holes are shown in Figure 3.4.2.2.2-2, and typical locating dimensions are shown in Figure 3.4.2.2.2-3.



d per manufacturers specifications, typically 0.85 to 0.90 $D_n$

where $D_n$ = Nominal thread size (maximum thread diameter)

$0.20\ D_n \leq t \leq 0.75\ D_n$

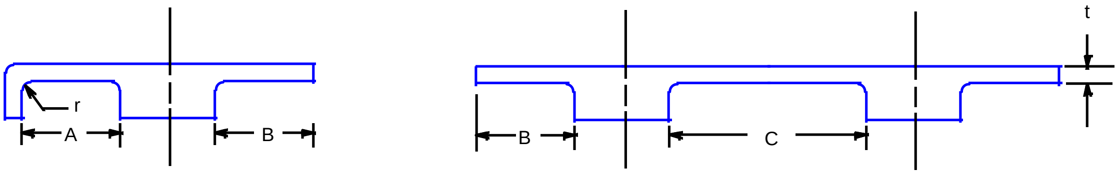
$0.5t \leq W \leq 0.6t$

H minimum ≈ $0.25\ D_n$ (for W = 0.6 t)

maximum ≈ $0.30\ D_n$ (for W = 0.5 t)

Length of thread engagement ≈ 0.95 (t + H)

**Figure 3.4.2.2.2-2** Recommended dimensions for extruded holes



| DIMENSION | MIN DISTANCE | BUT NOT LESS THAN |
|---|---|---|
| A. Edge of extruded hole to inside face of bend | 3t + r | 0.8 mm (0.3 in.) |
| B. Edge of extruded hole to edge of metal | 3t | 1.6 mm (0.6 in.) |
| C. Between edges of extruded holes | 6t | |

**Figure 3.4.2.2.2-3** Recommended dimensions for spacing extruded based holes, on manufacturing criteria

The dimensions shown are fairly conservative, and they reflect the restrictions imposed by material properties and manufacturing processes. Variations can sometimes be made in consultation with the appropriate manufacturing and materials personnel. It should be noted that the dimensions for hole location are also subject to the discussion on spacing and edge distance in Section 3.4.2.3, relative to the strength of the joint. Those relationships do not, however, reflect the benefit of the additional material at the extruded hole.

Thread rolling screws also form an interference fit in the bottom sheet, which makes them highly resistant to loosening under vibration. In addition, there is some latitude to increase the torque required to drive the screw by reducing the diameter of the extruded hole, consequently increasing the interference between the hole and screw. This latitude may be advantageous in the assembly plant, where a single driving tool is employed to install several different fasteners that require drive torques which vary somewhat. Suppliers of thread rolling screws frequently have their own patented thread profile, so that the drive torque data may vary from one manufacturer to another.

It is sometimes possible to provide sufficient length of thread engagement and wall thickness in the extruded hole to develop the full tensile strength of the thread rolling screw. In those cases, the minimum tensile strength of the screw governs the design, and published data on tensile strength are available[11]. If the extruded hole does not develop the full strength of the screw, the threads in the extruded hole are more likely to strip than the threads on the screw.

The strength of the threads in the extruded hole may, therefore, govern the strength of the joint. However, several factors make accurate prediction difficult. The extruding and thread rolling operations cold work the metal and increase the thread strength. Conversely, part or all of the extrusion may have a relatively thin wall, particularly with thin gauge sheet steel, which tends to diminish thread strength. If the strength of the joint is critical, tests should be run to determine the best available thread profile and optimize the dimensions of the extruded hole.

### 3.4.2.2.3 Clamp-Up Fasteners

Clamp-up fasteners (Figure 3.4.2.2.3-1) are also installed from one side.

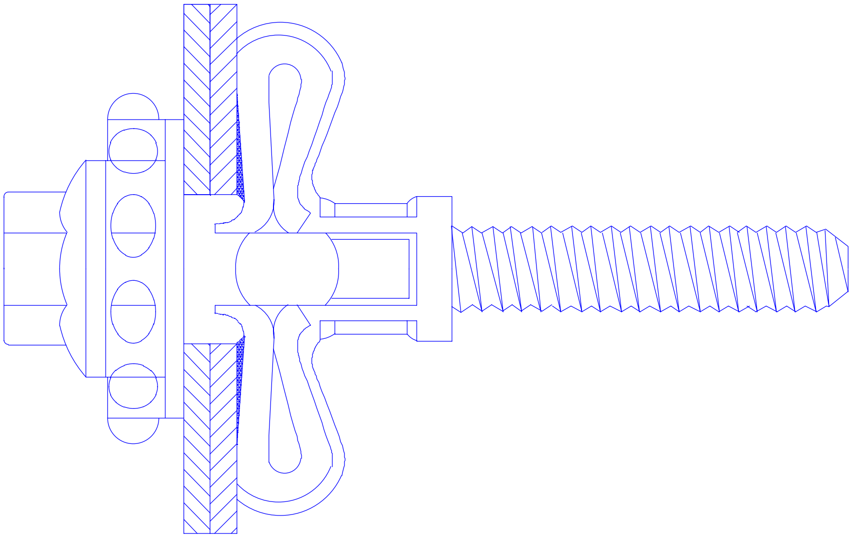


**Figure 3.4.2.2.3-1** Clamp-up type fastener

They are detailed so that, in effect, they have two washers and are thus more resistant to pullover. (See discussion of pullover in Section 3.4.2.3). They also tend to hold better under vibrating loads than the self tapping screw, when joining thin sheets.

### 3.4.2.2.4 Pop Rivets

Pop rivets are another type of one sided fastener that can be used to connect sheet steel. During installation, the fastener pin is drawn through the rivet (which acts as a sleeve) causing it to expand and fill the hole. The tensile capacity of this connection is often less than the shear capacity; however, the shear capacity is good for its size.

### 3.4.2.2.5 One-Side Fasteners Requiring Two-Side Access

Self-piercing, hardened steel fasteners are forced through the sheet metal from one side of the joint. However, access is required for a backup die on the opposite side. Depending on the process, either the die side or both sides are swaged or cold-formed over to lock the sheets together. This type of fastener is not as repeatable as conventional "aircraft" structural rivets, and joint strengths are not as high. However they develop higher static strengths than do mechanically fastened joints.

Self-piercing fasteners offer two principal design advantages:

1. Joints may exceed the normal 2.5-3:1 maximum metal thickness ratio for welding[12].
2. The tolerance stackups resulting from pre-punched holes, and the need to drill at assembly are eliminated.

Two design/processing factors affect the joint strength.

1. The joints are generally stronger when the fastener is inserted through the thinner sheet first with the clinch in the heavier piece of metal.
2. Lubricants should be removed from the joint because they may reduce the long term joint strength.

The phases of joint formation for one style of fastener are shown in Figure 3.4.2.2.5-1. A cross section of the joint is shown in Figure 3.4.2.2.5-2. The head of the fastener may be convex, as illustrated in Figure 3.4.2.2.5-1 or flush as illustrated in Figure 3.4.2.2.5-2.

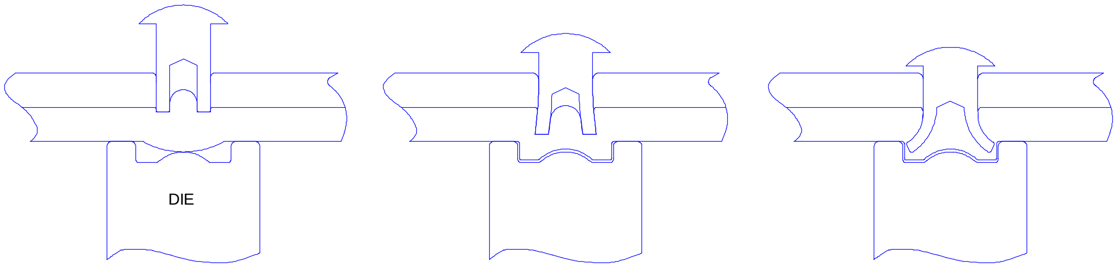


**Figure 3.4.2.2.5-1** Phases of joint formation with a self-piercing fastener

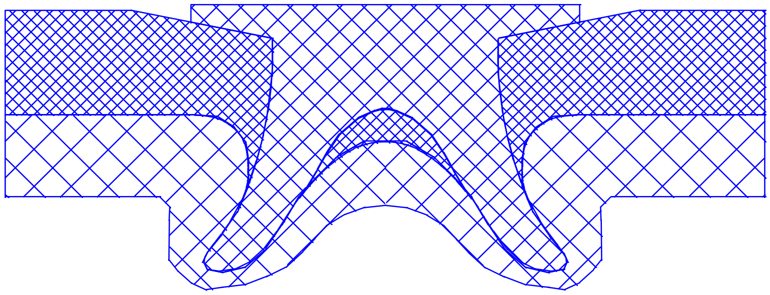


**Figure 3.4.2.2.5-2** Cross section of a joint formed with a self-piercing fastener

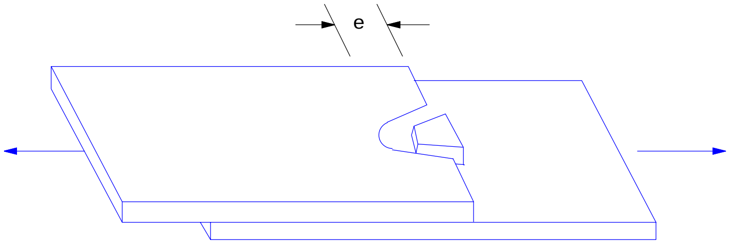
Automotive applications of this type of fastener include:

- Two piece control arm stampings, which are being partially fastened at key locations.
- Zinc coated steel seat structure panels 0.7 mm (0.028 in.) thick, which are being fastened to pre-painted steel seat brackets 3.0 mm (0.125 in.) thick

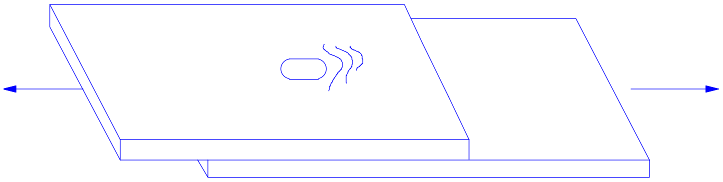
## 3.4.2.3 Design Considerations for Mechanical Fasteners

### 3.4.2.3.1 Pullover

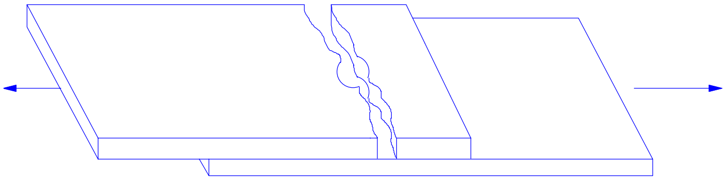
Sheet connections in single shear with thickness of 5 mm (3/16 in.) and smaller (i.e. with limited flexural stiffness) will tend to pull over the fastener. The fastener connecting two sheets in single shear will rotate to accommodate the pullover. Manufacturers' test results for #12 self tapping screws are indicated as pullover values instead of shear values in sheets of 1.27 mm (0.05 in.) thickness or less. This implies that considerations other than fastener shear integrity can control the shear capacity of a connection as illustrated in Figure 3.4.2.3.1-1.



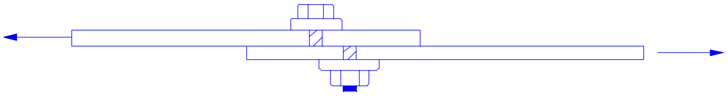
(a) Longitudinal shear failure of sheet



(b) Bearing failure of sheet



(c) Tensile failure of sheet



(d) Shear failure of bolt

**Figure 3.4.2.3.1-1** Types of failure of bolted connections

### 3.4.2.3.2 Spacing and Edge Distance for Bolts

The spacing and edge distance for bolts must be controlled in order to achieve the desired strength. The minimum bolt spacing should be **3d** (where **d** is the bolt diameter) between center of bolt holes. The distance from the center of any standard hole to the end or edge of member or plate should be at least **1.5d**. In addition, the distance $\mathbf{e_{min}}$ from the center of a standard hole to the end of the part or to the nearest edge of an adjacent hole in the direction of the line of force should be limited, to prevent the type of failure shown in Figure 3.4.2.3.1-1(a), to

$$e_{min} = \frac{V_u}{F_u t}$$

**Equation 3.4.2.3.2-1**

where $V_u$ = ultimate shear force to be transmitted
$t$ = thickness of thinnest part
$F_u$ = tensile strength of thinnest part.

Simply use consistent units for stress and dimension in the evaluation of $\mathbf{e_{min}}$.

### 3.4.2.3.3 Bearing Stress Limits

The ultimate bearing stress, **$F_{bu}$**, of the fastener against the sheet can also limit the shear capacity. The bearing stress is calculated as **$V_u/dt$**, where d is the nominal (gross) diameter of the fastener and **$V_u$** and **t** are as defined in Equation 3.4.2.3.2-1. Bearing failure is illustrated in Figure 3.4.2.3.1-1(b).

Although the bearing limit is better for steels with larger **$F_u/F_y$** ratios, the primary variable is the restraint provided to the sheet adjacent to the bolt. If washers are used under both nut and bolt head or the plate is confined by other plates on each side, the ultimate bearing stress is as follows when the deformation around the bolt holes is not a design consideration.

$$F_{bu} = 3.0F_u$$

**Equation 3.4.2.3.3-1**

If an outside sheet is attached with bolts having no washers, then the ultimate bearing stress on that outside sheet should be limited to

$$F_{bu} = 2.2F_u$$

**Equation 3.4.2.3.3-2**

To achieve these bearing limits, the end distance **$e_{min}$** would have to be greater than **1.5d** and based on evaluation of Equation 3.4.2.3.2-1.

### 3.4.2.3.4 Sheet Tensile Capacity

The tensile capacity of the sheet is the last thing to check (Figure 3.4.2.3.1-1(c)). For sheet with thickness **t** < 5 mm (3/16 in.), the capacity is a function of the presence of washers and conditions of single or double shear. The conservative expression that is applicable to cases with no washer under the bolt head or nut is

$$F_t = \left(10 - r + \frac{2.5\,r\,d}{s}\right)F_u \le F_u$$

**Equation 3.4.2.3.4-1**

where r = the force transmitted by the bolt or bolts at the section considered, divided by the tension force in the member at that section. If **r** is less than 0.2, it may be taken equal to zero.
d = nominal (gross) diameter of the fastener.
s = spacing of bolts perpendicular to line of stress. In the case of a single bolt, **s** is the width of sheet.
$F_t$ = ultimate tensile stress on net section

Use of this expression can be facilitated by the chart in Figure 3.4.2.3.4-1.

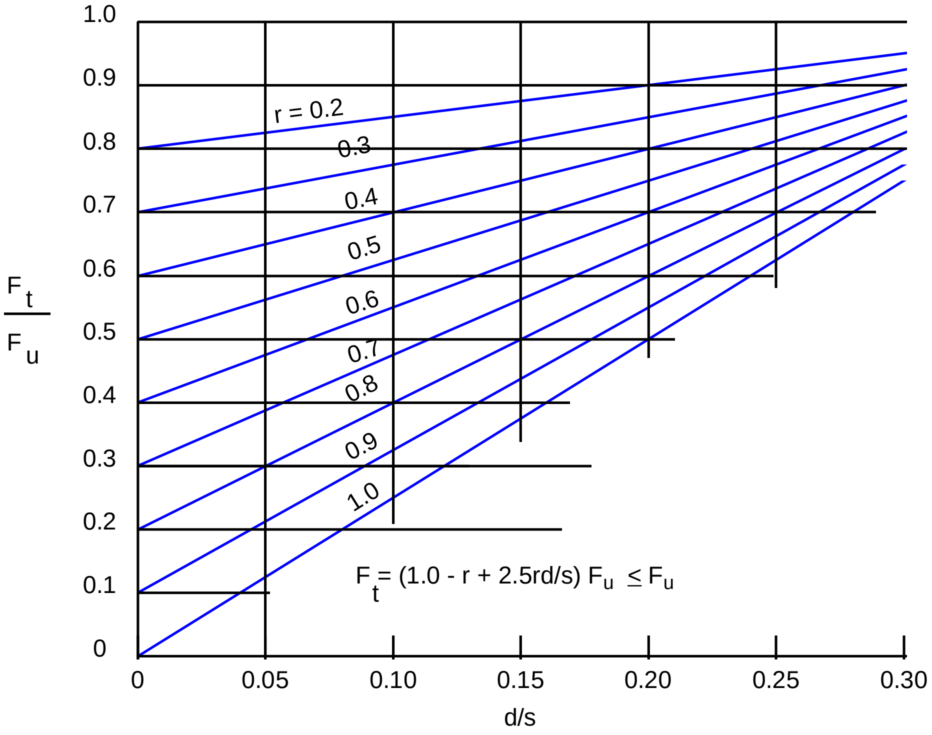


**Figure 3.4.2.3.4-1** Ultimate tension stress on net section for bolted connections

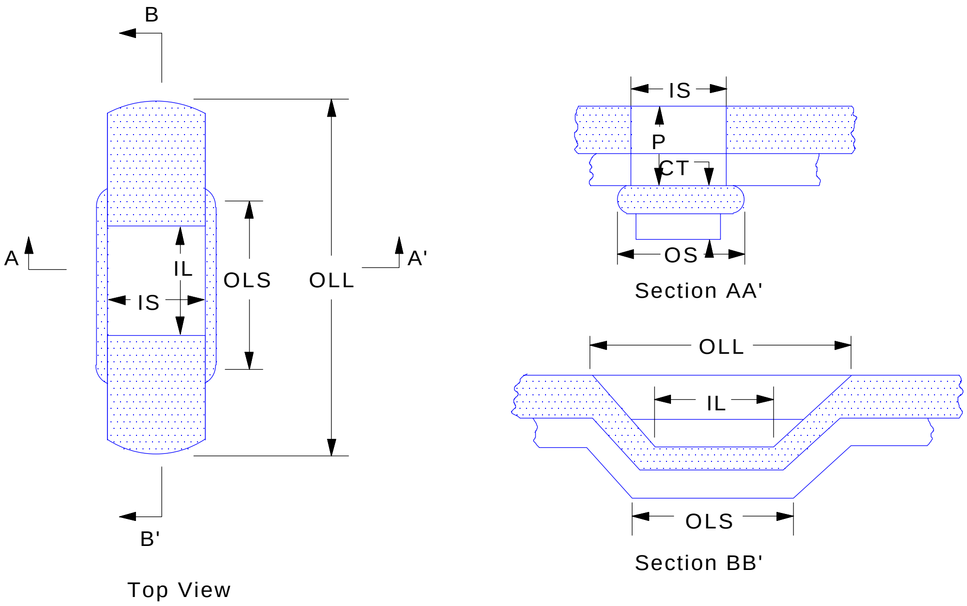
For information on the exact criteria for connections with washers, see the 1986 AISI Specification[3].

## 3.4.3 MECHANICAL JOINING

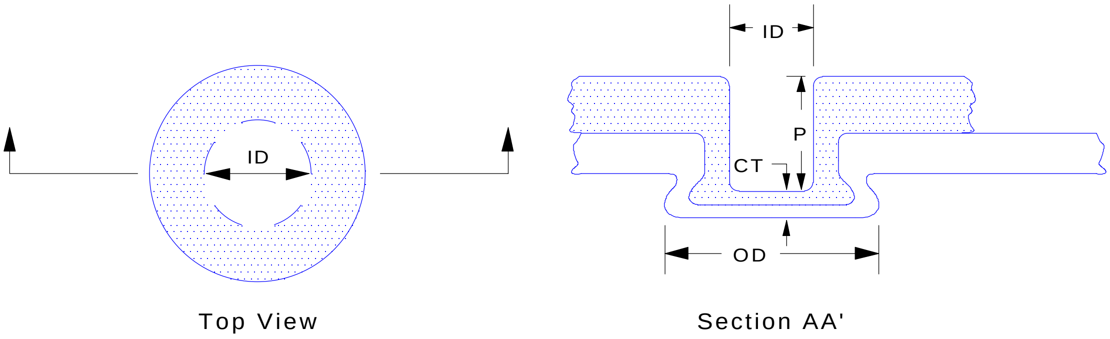
Where an alternative to spot or arc welding is required, and access for a backup die is available, mechanical clinch fastening may be used. The process is accomplished by mechanically joining two sheets directly, rather than using bolts or screws. It requires established minimum elongations of the clinched materials. Two mechanical joining methods are shown in Figure 3.4.3-1. The general guideline for clinching is to place the most formable or thickest piece on the punch side so that it can support the loads imposed by the tools.

The conventional mechanical joint is formed by first piercing or slitting the sheets; then the tabs are pressed to lock the two sheets together. The technique shown in Figure 3.4.3-1(a) is one employed to join two lapped sheets. The strength under static and fatigue loading is not as good as that of a spot weld[13, 14, 15].

The button type mechanical joint has better behavior than the conventional mechanical joint[13]. It exhibits better static strength, better energy absorption on peeling and since it has no cracks, it has better fatigue properties. The button type mechanical joint has high cycle fatigue properties comparable to those of a spot weld even though its static properties are lower.



(a) Schematic of conventional mechanical joint



(b) Schematic of button-type (crack free) mechanical joint

**Figure 3.4.3-1** Mechanical joint types

Examples of automotive applications of mechanical joining include:

- Assembly of very thin gauge, aluminized mild steel or ferritic steel heat shield assemblies.
- Assembly of adhesively bonded deck lid or hood inner and outer panels to fixture them before ELPO heat or to improve shear and peel strength. Also avoids "crows feet" caused by welding heat.
- Assembly of steel brackets sandwiched to plastic fender panels.
- Assembly of steel fan blades
- Assembly of steel chassis stampings prior to welding to minimize distortion.

## 3.4.4 SPACING OF MECHANICAL CONNECTIONS ..........................................

The spacing of mechanical connections between components, to develop a composite section, is typically based on several strength considerations. Little is presently known regarding the composite stiffness that is achieved. Flexural and axial stiffnesses are usually close to that expected. However, when open sections are attached intermittently to produce a closed section, the expected torsional stiffness is not achieved. Apparently, such sections behave as open sections between connection points. Connection integrity must be more than that required by strength in order to approach the stiffness obtained from continuous connection between components.

An example of a strength consideration is the connection between a pair of channel shapes to form either an **I** or tubular section in compression. AISI requires a connection spacing that limits the slenderness ratio of the channel between connectors to half the critical slenderness ratio of the composite section.

Where a flat sheet in compression is attached to another element, AISI recommends a spacing that prevents Euler buckling at the stress level desired. The effective length factor is considered at 0.6 and the radius of gyration is $t/\sqrt{12}$ for the cover sheet of thickness **t**. This leads to an expression in the AISI Specification with parameters $t/\sqrt{f}$ where **f** is actual stress in the plate. This expression does not have a constant factor of safety since it is based on elastic buckling only. It will tend to be unconservative as the stress **f** approaches the yield stress.

An alternate to the above can be employed for compression members. The alternate procedure is to employ the relative slenderness approach used for connecting channels.

Letting **K** = 0.6 for the plate of thickness **t**

$$\frac{0.6s}{\left(\frac{t}{\sqrt{12}}\right)} \le C\left(\frac{KL}{r}\right) \text{ from which } s \le 0.48\,Ct\left(\frac{KL}{r}\right)$$

**Equation 3.4.4-1**

where **KL/r** is the critical slenderness ratio of column.

With factor **C** = 1, the slenderness of the plate just matches that of the section; with **C** = 0.5, the slenderness ratio of the plate is half that of the section. Equation 3.4.4-1 considers inelastic buckling since the limiting stress is governed by Equation 3.1.2.5-1 using the **KL/r** of the section.

If the flat sheet or cover plate is used in a flexural member where **KL/r** is not calculated, establish a pseudo **KL/r** from the maximum stress required in bending. Determine **KL/r** from Equation 3.1.2.5-1 by letting $F_u$ equal the maximum bending stress in compression. This procedure is illustrated in Example 3.4-1 shown in Section 6.2.4.1.

Another design consideration for connection spacing of a flat plate (or any plate that laps with another leaving an unstiffened edge) is to prevent premature buckling of the unstiffened edge. This can be achieved if the spacing is kept to three times the projecting flat width **w**, (see Figure 3.4.4-1. This **3w** value need not be less than $500t/\sqrt{F_y}$ (millimeters) since the unstiffened element can yield at this spacing. The $500\,t\sqrt{F_y}$ is again likely to govern in most cases.

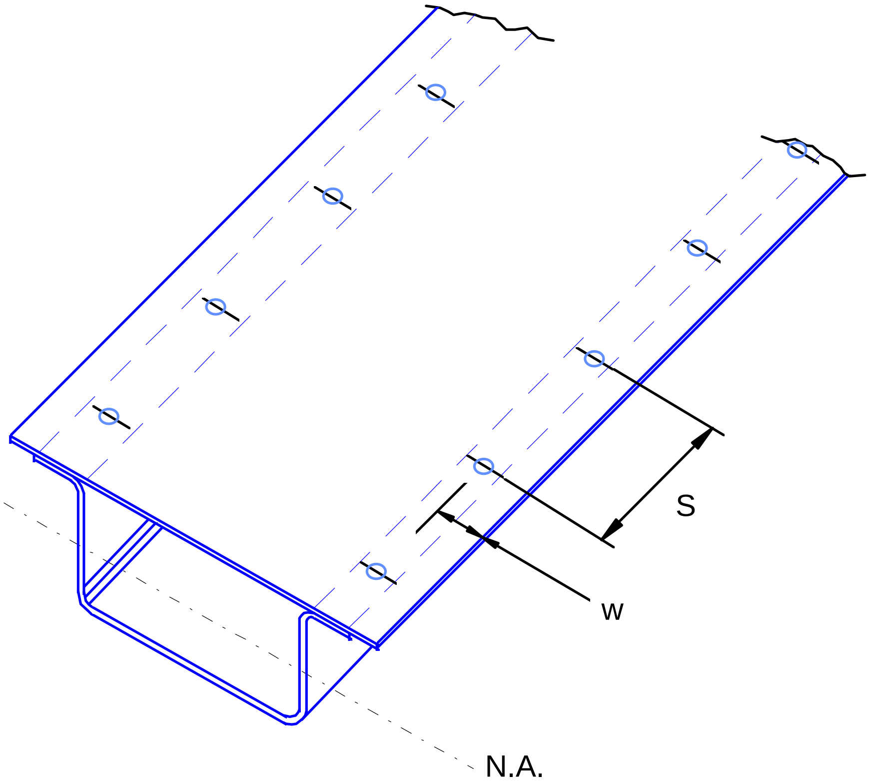


**Figure 3.4.4-1** Spaced connection of flat element

The strength of the connection must also be checked, of course. For a flexural member, a longitudinal shear stress calculation is required. If the flat plate acts like a flange, the shear force per attachment point, **$V_s$**, is calculated as

$$V_s = \left(\frac{s}{n}\right) V_m \frac{Q}{I}$$

**Equation 3.4.4-2**

where $V_m$ = maximum shear force in the beam
$Q$ = moment of the plate area about the section neutral axis
$I$ = section moment of inertia
$n$ = number of attachment points per location along section
$s$ = spacing of attachment points

Under axial load, there is no equation that dictates the shear requirement. Yu[16] recommends using 2 percent of the axial load as a shear force that can be employed in Equation 3.4.4-2 to compute the required **$V_s$**. Another approach would be to check the ability to transfer the axial load on the cover plate from the rest of the section into the plate over a certain length of the compression member.

$$V_s = \frac{\left(\frac{s}{n}\right) P_p}{L'}$$

**Equation 3.4.4-3**

where $P_p$ = axial force in cover plate
$L'$ = length over which it is desired to transfer load

The value of $L'$ could be as much as a quarter of the unbraced length of the member or may be only twice the depth of the section.

Example 3.4-1 illustrates the evaluation of the spacing criteria for connection of a flat plate.

### 3.4.5 ADHESIVES

This section gives an overview of the types of adhesives available to the body designer and their characteristics, especially as they relate to the performance of the adhesive bonded joint. A glossary of terms relevant to adhesives is included in Section 3.4.8.

Adhesive materials are basically plastics. Their characteristics and their performance in a given application will therefore be distinctly different from that of the steel members that they join. For example, the modulus of elasticity and design strength of steel are unaffected by temperature changes that vehicle bodies experience. However, for some adhesives they vary considerably over the same temperature range. The performance of the joint will also be distinctly different from joints made be welding or mechanical fastening.

Adhesives, like all plastics, are classified as either thermoplastic or thermosetting. Thermoplastics can be repeatedly heated to the point where they become liquid, and resolidified by cooling. Thermosets solidify by a chemical reaction called crosslinking, in which the molecules form bonds with adjacent molecules to form a solid state. The thermoset reaction does not result from the addition or removal of heat (although in many cases it is accelerated at elevated temperatures) and it is not reversible.

Structural and nonstructural are the two major categories for the adhesives that are used to bond sheet steel. In some literature, the distinction is made on the basis of application; that is, adhesives are called structural when used in applications where joints are integral parts of load bearing structures, such as B pillar to roof rail attachment. Adhesives are called nonstructural when used in applications where loads are negligible, such as the attachments of trim. In other literature the distinction is based on capability; that is, adhesives are called structural if they can be used in structural applications such as the one described above, and they are called nonstructural if they are not suitable for such applications. In this manual the latter distinction, capability, is applied.

The properties of all adhesives allow them to serve additional functions as sealants, insulators, and corrosion barriers. Table 3.4.5-1 summarizes the properties and characteristics of five types of structural adhesive.

**Table 3.4.5-1** Summary of properties and characteristics of structural adhesives[17]

| | Epoxy | Polyurethane | Modified Acrylic (Reactive) | Cyanoacrylate | Anaerobic |
|---|---|---|---|---|---|
| Typical Cure Time: hr @ °F | 2-24 @ 70<br>0.1-4 @ 300 | 0.2-0.5 @ 70 | 0.05-1 @ 70 for handling | 0.5-5 @70 | 1-12 @ 70<br>0.05-2 @ 250 |
| Number of Components | One or Two | One or Two | One or Two | One | One |
| Mixing | Some | Some | Some | No | No |
| Materials Bonded | Most | Most smooth, non porous | Most smooth, non porous | Most non porous | Metals, Glass thermosets |
| Gap Filling Characteristics | Tends to become brittle when thick | Excellent | Excellent | Poor | Poor |
| Substrate Preparation Required | Can bond oily Surfaces | Surface treatment and cleaning important | Some can bond oily surfaces | Yes | Yes |
| Service Temperature Range °F | -67 to 250 | -250 to 175 | -100 to 250 | -67 to 175 | -67 to 300 |
| Impact Resistance | Good | Excellent | Good | Poor | Fair |
| Maximum Lap Shear Strength: psi @ 70° F | 2200 | 2200 | 3700 | 2700 | 2500 |
| Typical T-Peel Strength: lb/in. | < 3 - 20 | 80 | 30 | < 3 | 10 |
| Remarks | Good for dissimilar materials. Low Shrinkage. Variable pot life. | Excellent for low temperatures and dissimilar materials. | Good for bonding dissimilar materials. | Brittle, low shrinkage. Thin films are best. | May cause crazing of some thermoplastics. Thin films best. |

## 3.4.5.1 Structural Adhesives

Structural adhesives are classified as either one or two part. This classification is important because the systems have different processing characteristics. One part adhesives are easy to dispense, but cure slowly, often requiring heat. Two part adhesives cure rapidly, but usually require mixing and dispensing equipment.

### 3.4.5.1.1 Epoxies

Epoxies are available as one or two part adhesives. One part epoxies require heat to cure; two part may utilize mild heating to speed curing. Epoxies have exceptional shear strength and can be chemically modified to increase peel strength or durability. One part epoxies generally have superior environmental durability and oil resistance compared with two part. This makes one part epoxies useful for bonding coated steel after stamping.

### 3.4.5.1.2 Urethanes

Urethanes are also available as one or two part adhesives. They require less heating to develop handling strength than epoxies. Cure can sometime be accelerated by adding water. Urethanes have shear strength that is equal to epoxies, and better peel strength. However, they may require a clean or primed surface for bonding. Urethanes are used for bonding glass to painted metal.

### 3.4.5.1.3 Acrylics

Acrylics are also available as one or two part adhesives and they cure rapidly without the need for fixturing. They are well suited to bonding metal because of their oil resistance, but the best acrylics currently have an objectionable odor. Since zinc is a catalyst for curing acrylics, galvanized steel could be potentially used as the catalyst. The actual surface area of zinc that is available to the acrylic is important. For example, electrogalvanized is rougher than hot dip galvanized, and would therefore have a higher potential for success. Special formulations are required to bond both sides of one side galvanized steel.

### 3.4.5.1.4 Cyanoacrylates

Cyanoacrylates cure rapidly and react with the moisture on the surface of the steel to induce cure. However, they cure too quickly to bond large areas. They require well mated surfaces because they cannot be used to fill gaps, and they will not tolerate oil on the adherends.

### 3.4.5.1.5 Anaerobic Adhesives

Anaerobic adhesives cure when oxygen is removed from the joint; this is especially useful when the joint is in a confined space. They are best suited for thread locking applications.

### 3.4.5.2 Nonstructural Thermoplastic Adhesives

Two categories of nonstructural thermoplastic adhesives are currently used in automotive body applications: vinyl plastisols and hot melt adhesives.

### 3.4.5.2.1 Vinyl Plastisols

Vinyl plastisols heat cure and require fixturing. They are oil resistant and, although they are not structural adhesives, have high peel strength. Vinyl plastisols are used to assemble the hood inner and outer panels.

### 3.4.5.2.2 Hot Melt Adhesives

Hot melts are thermoplastic adhesives that bond very quickly. They are applied as a hot liquid that wets the surface and gains adhesive strength as the liquid solidifies. Hot melts have difficulty in bonding to nonpainted metals because metals, depending on the surface area, may dissipate the heat before the adhesive wets the surface.

### 3.4.5.3 Physical Properties of Adhesives in Vehicle Design

The physical properties of an adhesive determine where it can be effectively used in automotive applications. For example, impact resistance is important in the front or "crush zones"; joints in the passenger compartment are designed for stiffness and strength. An understanding of the properties of adhesives is therefore important for selection of the optimum adhesive.

There are two principal modes of failure in a bonded joint: adhesive and cohesive. Adhesive failure occurs at the interface of the adhesive and adherend, whereas cohesive failure occurs within the adhesive. The two modes are shown schematically in Figure 3.4.5.3-1.

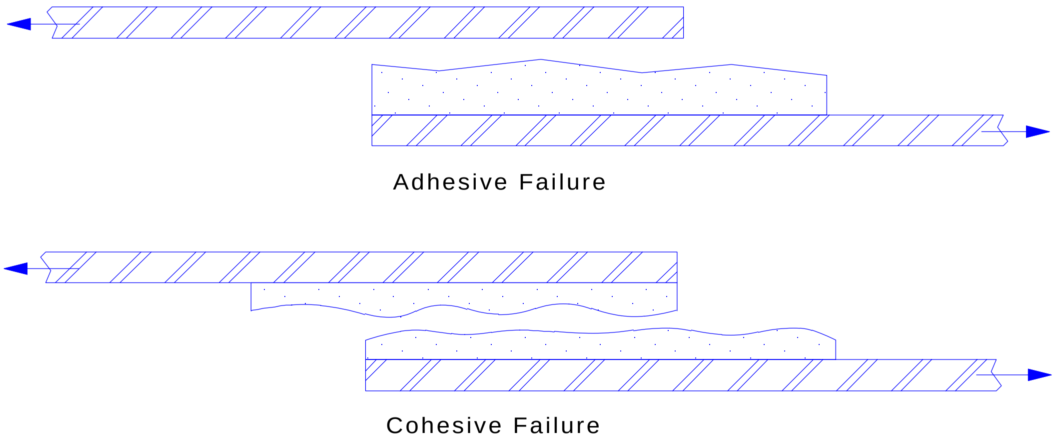


**Figure 3.4.5.3-1** Adhesive and cohesive failures

Cohesive failure is the preferred mode because the life of the bonded joint can be predicted to some degree of accuracy, provided that applied loads and environmental conditions are as anticipated.

### 3.4.5.3.1 Impact Strength

Impact strength is essential for bonding components that are prone to large deflections, and subject to impact loading, as in a collision. In order for the adhesive to absorb impact energy without disbonding, it must exhibit high elongation and high peel strength. These properties allow the adhesive to stretch and flex before failure. Therefore, an elastic adhesive is generally selected when impact strength is the major consideration. Urethanes and toughened epoxies (modified to increase elongation and peel) are good choices for impact absorbing applications since they have more peel strength than rigid epoxies. Figure 3.4.5.3.1-1 shows the relationship between load and deflection for rigid (brittle) and flexible (elastic) adhesives.

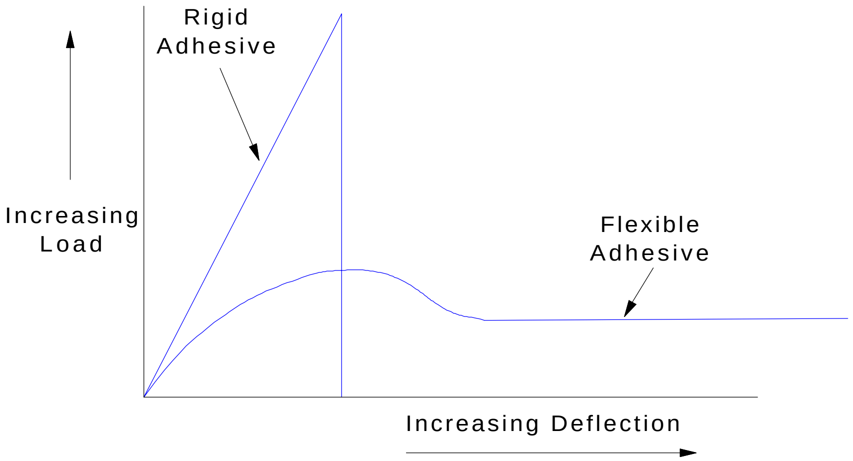


**Figure 3.4.5.3.1-1** Adhesive load deflection properties.
Typical characteristics for static loading[18]

Figure 3.4.5.3.1-2 shows the peel stress distribution for rigid and flexible adhesives.

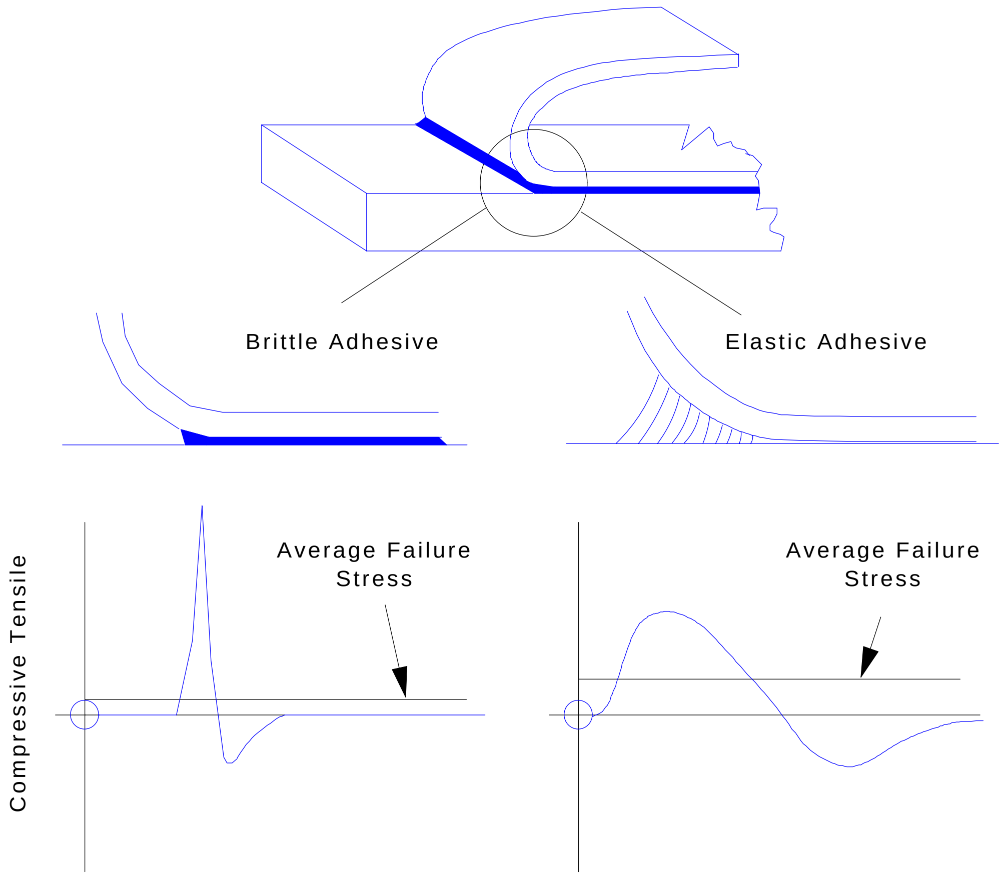


**Figure 3.4.5.3.1-2** Peel stress distribution[19]

Adhesives become more rigid when suddenly loaded, as in the sudden impact of a collision. Thus, an elastic adhesive has more impact strength than static load tests indicate. Although the properties of adhesives may be reasonably well characterized, testing is always needed to determine if the adhesive retains both the minimum load bearing and impact characteristics necessary for a given application.

### 3.4.5.3.2 Peel Versus Shear Strength

Peel and shear are two of the four types of loading discussed in Section 3.4.6.1. Since bonded joints often experience load combination, adhesive selection often involves a trade-off. Applications demanding high shear strength require a rigid adhesive; applications demanding high peel strength require a flexible adhesive.

Most rigid adhesives are too brittle for joints with simple geometries, such as lap joints, that require high shear strength. A rigid adhesive can be blended with a more flexible component, a process called toughening the adhesive. The addition of the flexible component allows the adhesive more elongation before bond failure, which increases peel strength and reduces shear strength. The trade-off is illustrated in Figure 3.4.5.3.2-1.

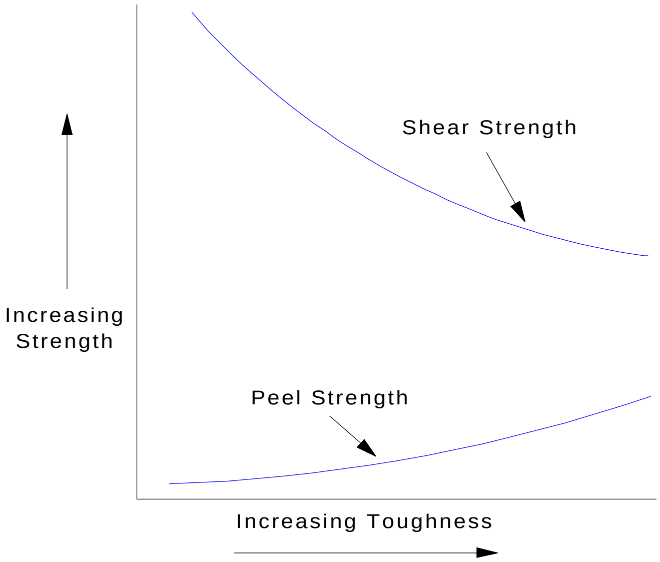


**Figure 3.4.5.3.2-1** Strength-toughness trade-off in an adhesive[18]

Epoxies are usually toughened (modified to increase elasticity and peel strength) for automotive applications. Urethane adhesives have high peel strength and are usually stiffened (modified to reduce elasticity) to obtain high shear strength. In vehicle design, the high peel strength of urethanes makes them an excellent choice for bonding painted metal to glass. However, unlike epoxies, urethanes have poor oil resistance, and so toughened epoxies are used for bonding of oily steels in high strength applications.

### 3.4.5.3.3 Bond Life Versus Load

The load that a bonded joint can withstand, under conditions of long term continuous loading, is a function of the expected life. Typical of plastics, the design strength must be reduced when long term continuous loading is anticipated. For a life of five to ten years, design strength should be 10 to 20% of the ultimate strength of the joint. When the anticipated life is very short, in the order of one month or less, the design strength may be as high as 50 to 75% of ultimate. The relationship is shown schematically in Figure 3.4.5.3.3-1.

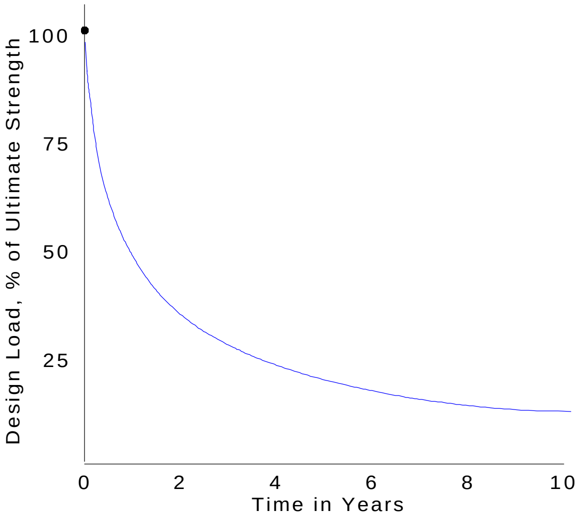


**Figure 3.4.5.3.3-1** Schematic design load vs. time relationship for adhesive bonded joints

Actual values for design strength at expected life depend on the adhesive formulation and the anticipated loading.

### 3.4.5.4 Other Factors Affecting Performance Operating Temperature

Adhesives exhibit thermal properties that must be considered when selecting an adhesive. Some adhesives lose their elasticity at lower temperatures. Maximum shear strength generally occurs at a temperature within the useful range, and decreases at lower and higher temperatures. Figure 3.4.5.4-1 shows typical relationships between shear strength and temperature for several structural adhesives.

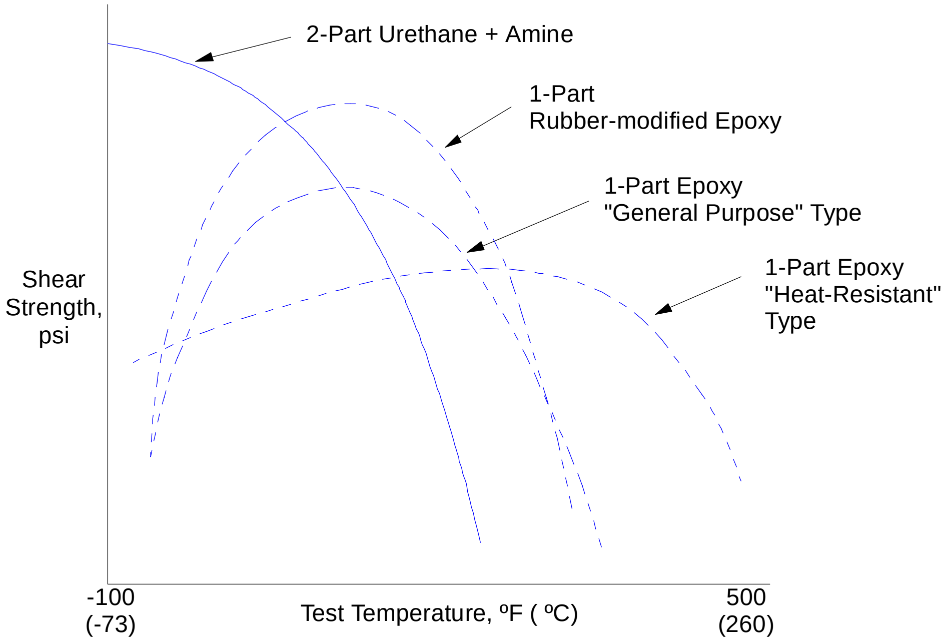


**Figure 3.4.5.4-1** Effect of temperature on shear strength[19]

Therefore, for automotive applications the elongation and peel strengths of an adhesive must be tested over the entire anticipated temperature range. Although the properties of rigid adhesives are less temperature dependent than flexible adhesives, their peel strength is too low for normal conditions.

High temperature resistance must be considered for adhesive joints neat the engine or exhaust system of the car. As temperature increases, rigid adhesives soften and become more flexible. Although the rigid adhesive loses its shear strength upon softening, its peel and impact strength increase.

### 3.4.5.4.1 Bondline Thickness

Peel and shear strength are both affected by bondline thickness. Figure 3.4.5.4.1-1 shows that bondline thickness should be kept to a minimum to fully exploit the shear and peel strengths of the adhesive.

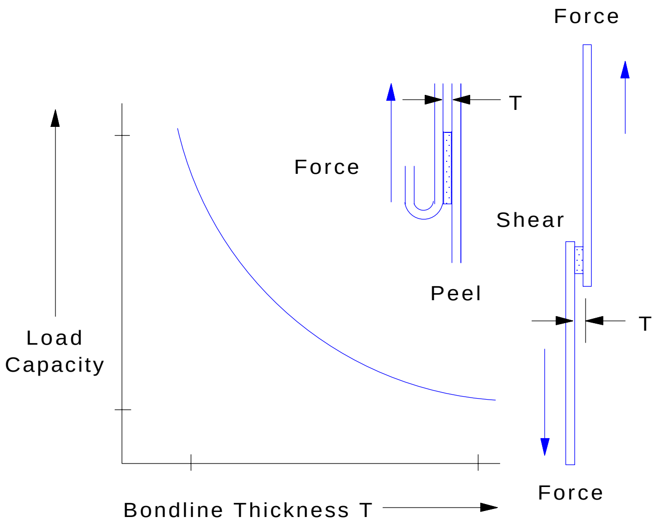


**Figure 3.4.5.4.1-1** Load capacity vs. bondline thickness

### 3.4.5.4.2 Curing

Many adhesives require a curing cycle, in which the temperature of the adhesive is elevated, causing it to develop its full strength. It is current practice to tailor the adhesive so that it will cure in the paint ovens. The economic advantages are obvious; however two areas require special precautions:

1. Anticipated changes in paint oven temperatures must be known in advance so that adhesives can be modified and tested to meet production schedules. Adhesives suppliers usually assume this responsibility.
2. Adhesives that will operate at high temperatures (above 230°F or 110°C) require higher curing temperatures. The designer must ensure that paint oven temperatures will be high enough, or that a special cure cycle will be employed.

## 3.4.6 DESIGN FOR ADHESIVE BONDING

Adhesive bonding can be an economical and effective means for joining sheet steels. Bonding, may, however, be overlooked if the designer is not familiar with the process. Structural bonding using adhesives has been employed in the aircraft industry since World War II, and it is finding numerous applications in automotive sheet steel body components. Design guidelines for bonding sheet steels have been developed and published[19]. This section focuses on automotive body applications.

Adhesive bonding offers a number of advantages over welding and mechanical fastening:

1. Sheet steel can be joined to materials that are dissimilar, including those with widely varying thicknesses and thermal properties.
2. Adhesive bonding produces a relatively uniform distribution of loads, and it does not impose discontinuities (such as holes) in the sheet steel. These characteristics lead to relatively good fatigue resistance.
3. Mechanical properties of the steel, such as those developed by heat treatment, can be preserved.

4. With proper design, the process does not leave a witness mark to mar cosmetic surfaces, such as the indentation that is left by spot welding.
5. A bonded joint develops a continuous contact area between surfaces that is attractive. It also provides a tight seal, which can act as a nonconductive layer (insulator ) across the joint and as a barrier to corrosive environments.
6. Adhesives can be used in combination with spot welding or mechanical fastening to improve the performance of the joint.

The use of adhesive bonding imposes special conditions and requires the input of materials and manufacturing as well as design personnel. Among the problems that the team must address are:

1. build variations
2. surface conditions of the parts to be bonded
3. processing steps
4. substrates (coated or uncoated steel)
5. substrate lubricants
6. preparation and storage of adhesive
7. health of workers (toxicology)

### 3.4.6.1 Types of Static Load

There are four types of static load to which bonded joints may be subjected: tension, cleavage, shear and peel. These are illustrated in Figure 3.4.6.1-1.

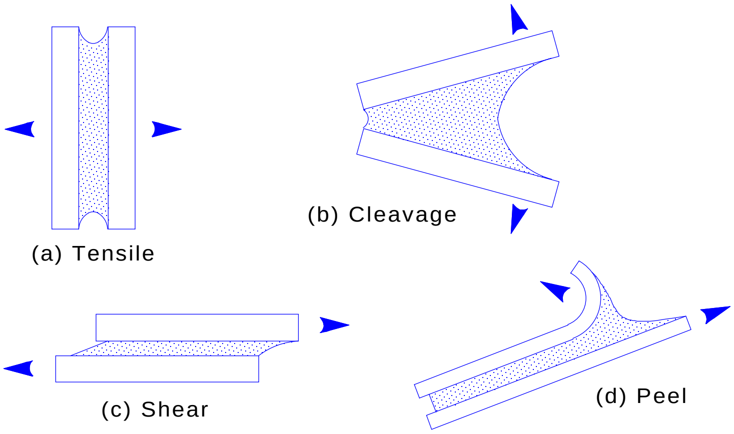


**Figure 3.4.6.1-1** Types of static load

The designer should bear in mind that bonded joints usually experience a combination of these loads. Load types are discussed individually in this section to identify the characteristics of each and clarify their behavior.

In pure tensile loading, Figure 3.4.6.1-1(a), the forces are applied perpendicular to the plane of the joint, and the stress is distributed uniformly over the bond. This type of loading occurs when the applied loads are uniformly distributed, or when the members are rigid enough to effect uniform distribution of concentrated loads. Pure tensile loading is rare and impractical for joining sheet materials, as Figure 3.4.6.1-2 would suggest.

Cleavage occurs when tensile forces are applied nonuniformly over the bond as shown in Figure 3.4.6.1-1(b). Adhesives will fail at lower cleavage loads than tensile loads, because localized stresses are higher than the stress associated with uniform distribution. Pure cleavage occurs only when the members are sufficiently rigid to avoid deflections that induce peel loading.

Shear, shown in Figure 3.4.6.1-1(c), occurs when the applied loads are parallel to the plane of the joint, and distributed uniformly. Pure shear is desirable, but it rarely occurs in bonded sheet metal structures because the members must be sufficiently rigid to remain parallel to the applied load, and thus avoid deflection that induces cleavage loading (Figure 3.4.6.1-2).

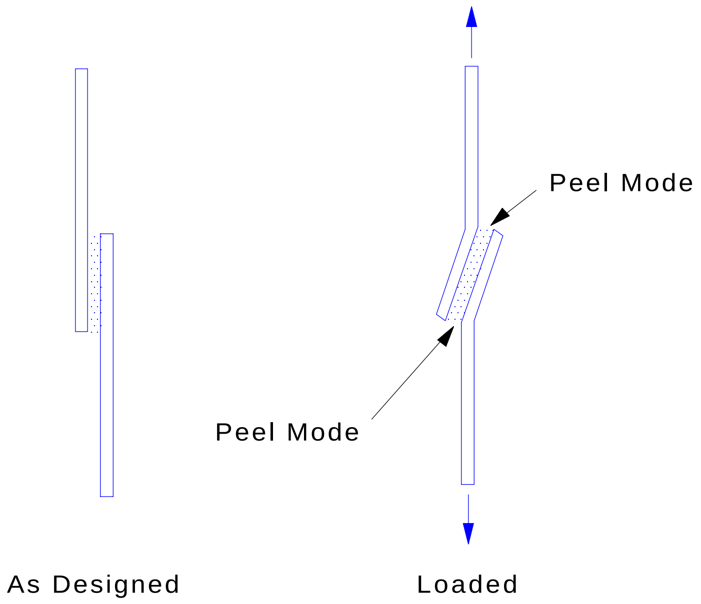


**Figure 3.4.6.1-2** Peel forces may be generated at the bond ends when lap joints are subjected to a tensile load

Peel, illustrated in Figure 3.4.6.1-1(d), is undesirable because the stresses in the adhesives are concentrated on a very thin line at the edge of the bond. Since most automotive body components are made from relatively thin gage steel, most of the joints designed for automotive bodies will experience some degree of peel.

## 3.4.6.2 Adhesive Design Data

The designer must rely on test data to design bonded joints. While the performance of the joint will ultimately be verified by testing, the initial design will rely on test data. The data must adequately indicate whether bonding is feasible, and support a reasonably accurate first design approximation. ASTM test methods for developing tensile, lap shear, and peel data are illustrated in Figure 3.4.6.2-1 and Figure 3.4.6.2-2.

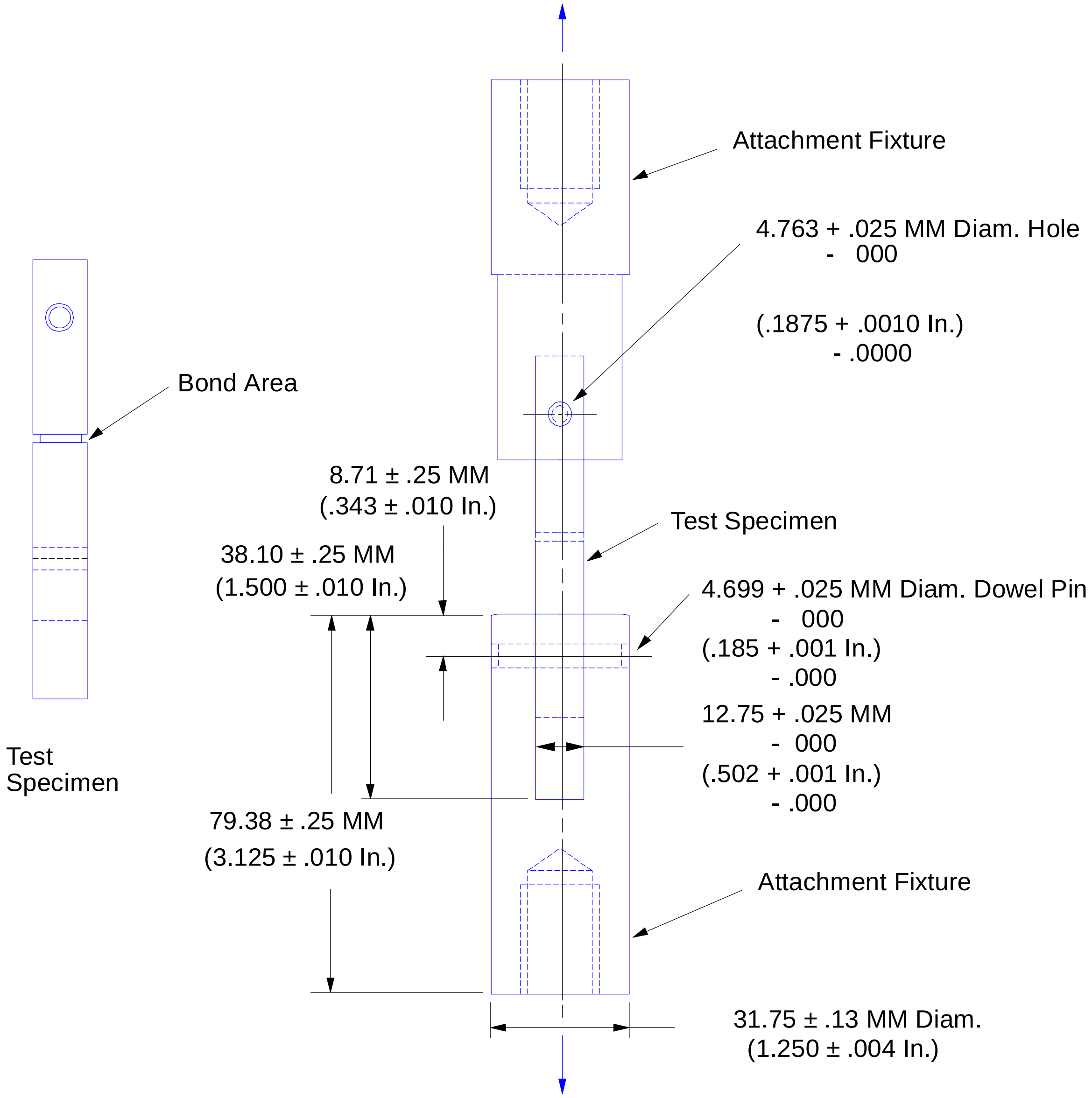
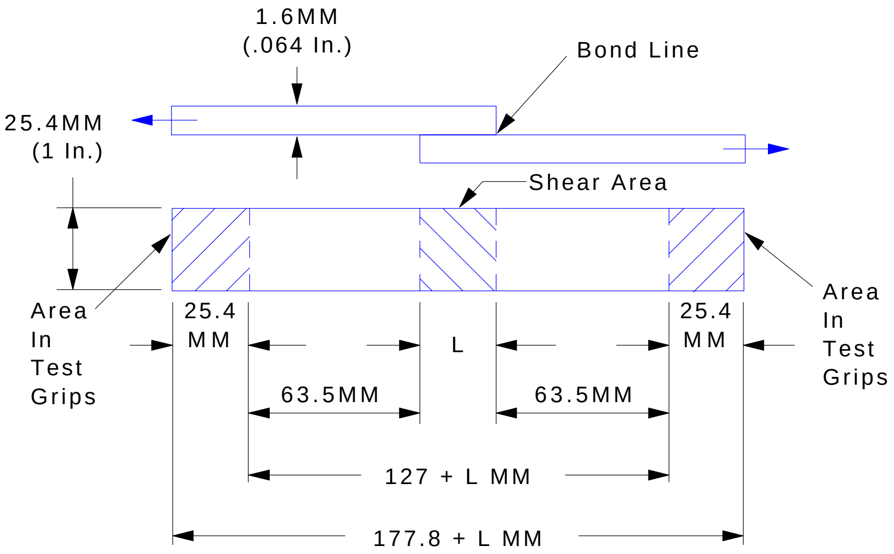
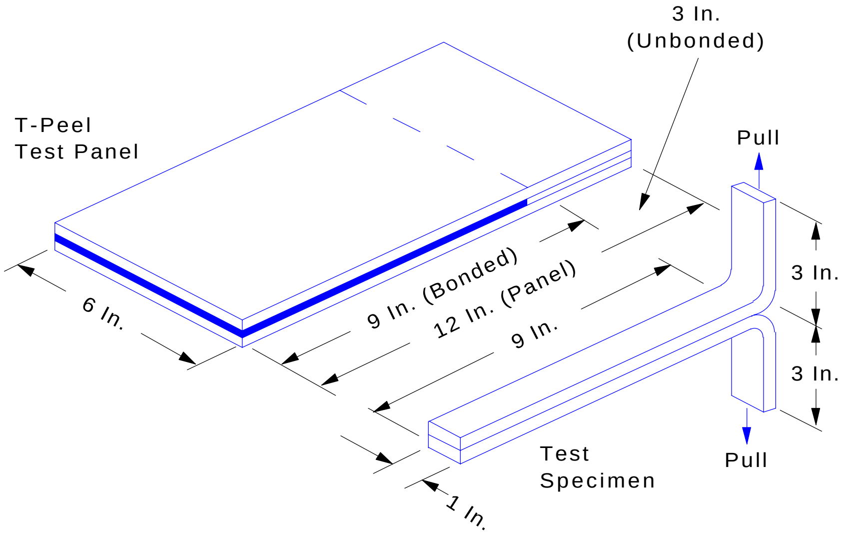


**Figure 3.4.6.2-1** Test specimen and attachment fixtures for tensile test (from ASTM D2095-62T)

The dynamic performance of a bonded joint is also of interest, particularly on joints that are subject to loads imposed by vehicle crash conditions. Efforts are now being made to establish test procedures that can be used to quantify the dynamic performance of bonded joints in sheet steel structures.



(a) Lap Shear Specimen (From ASTM D1006-62)



(b) Test Panel and T-Peel Specimen (From ASTM D1876-61T)

**Figure 3.4.6.2-2** Lap shear and T-peel test specimens

Predicting the performance of an adhesive bonded assembly based on data derived from tests on standard test specimens presents certain difficulties. They can be handled if the designer understands the relationship between laboratory results and real world performance. There are four factors that must be considered when correlating test data with the design at hand.

1. ASTM test methods are designed to compare adhesives and minimize substrate effects. The values derived are good for rigid members, but overestimate the strength of joints using drawing quality sheet steels.
2. Test specimens used on tests other than ASTM may not be subjected to a single pure load condition. The specimens, like the bonded joint, may deflect somewhat so that the adhesive is loaded in several directions.

3. The joint will be subjected to load combinations that are less predictable and less controllable than for the test specimen. For example, Figure 3.4.6.1-2 shows a combination of shear and peel in a joint with a simple geometric configuration. The peel is induced when the offset load causes the steel sheet to bend. The mode of loading becomes less predictable as the geometry of the joint becomes more complex.
4. Test data on standard specimens are based on the bonding technique as well as the adhesive. Factors such as substrate surface preparation, adhesive application, and curing, which are integral parts of published test data, may not be achieved or well controlled in practice.

Difficulties in correlating test data with real world performance can be minimized by utilizing good design practice. Ultimately the joint must be tested and the results evaluated against the design performance requirements.

### 3.4.6.3 Design Considerations

The design requirements of a bonded joint may be met by following four principles that are essential to good design practice:

1. Adhesives are strongest when loaded in shear. Design to maximize shear and minimize peel, cleavage and tension forces.
2. Use the maximum feasible bond area.
3. Avoid stress concentrations.
4. Consult with appropriate manufacturing and adhesives personnel to ensure that the adhesive and assembly techniques are optimized.

It is especially important to minimize peel, which is a concern whenever thin gage steel is joined. For example, Figure 3.4.6.3-1(a) shows a joint that may be appropriate for spot or resistance seam welding when the joint is accessible from only one side. However, this configuration is inappropriate for adhesive bonding, because the mode of loading would be mostly peel. Two design alternatives for a bonded joint, which introduce shear as the principal loading mode, are shown in Figure 3.4.6.3-1(b).

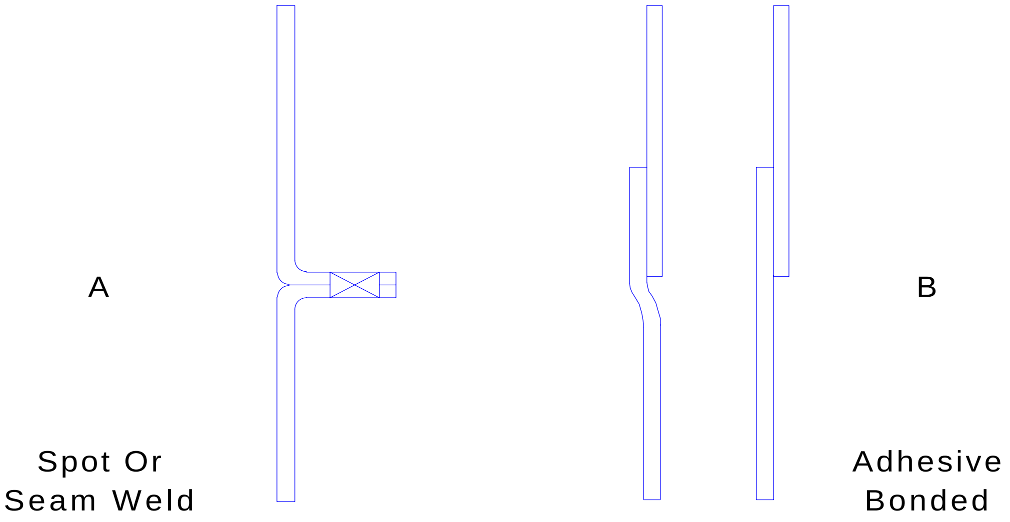


**Figure 3.4.6.3-1** Alternative joint designs

When only one side of a joint is accessible, a spot or seam weld flange may be acceptable. If the members are bonded, a lap joint is preferred.

It is not always possible to redesign a joint to minimize peel. In those cases another type of fastening with adequate resistance to peel, such as spot welding or mechanical fastening, may be combined with bonding to develop the desired joint properties. Spot welding or mechanical fastening may also be required to fixture the joint while the adhesive develops handling strength (see Section 4.4.2). All such factors should be identified early in the design so that supplementary fastening can be designed to meet both requirements.

The adhesive application process must be factored into the design of the joint if the adhesive is to be constrained from wiping onto visible metal surfaces. When the adhesive is spread over 80 percent of the flange width or less (Figure 3.4.6.3-2), the application process can usually be controlled to confine the adhesive within the flange. The design strength of the joint will be based on a bond width that is less than the flange width.

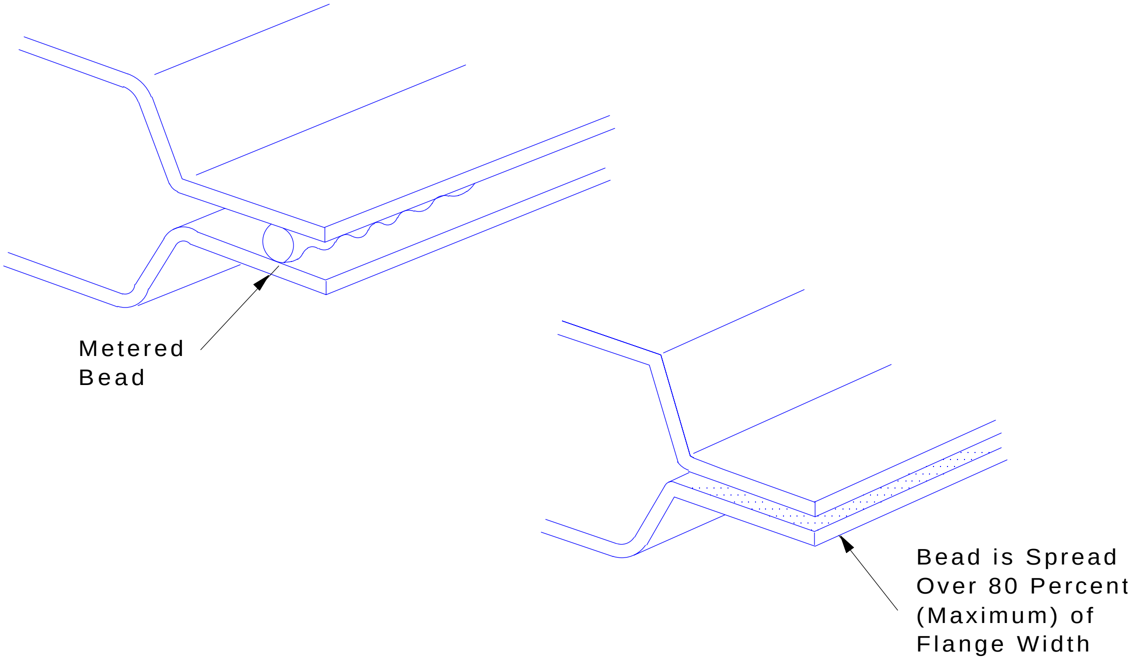


**Figure 3.4.6.3-2** Squeeze out control by providing adequate flange width

Alternatively, Figure 3.4.6.3-3 and Figure 3.4.6.3-4 show joints that have been designed so that the full width of a designed area is bonded, and any excess adhesive is squeezed into nonvisible areas.

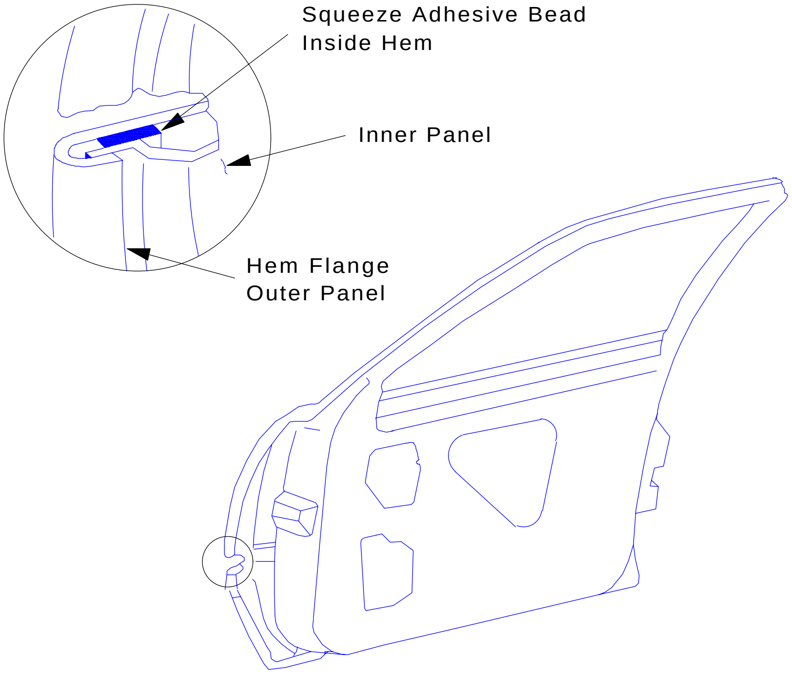


**Figure 3.4.6.3-3** Squeeze out control of a bonded joint for a door hem in a visible area[18]

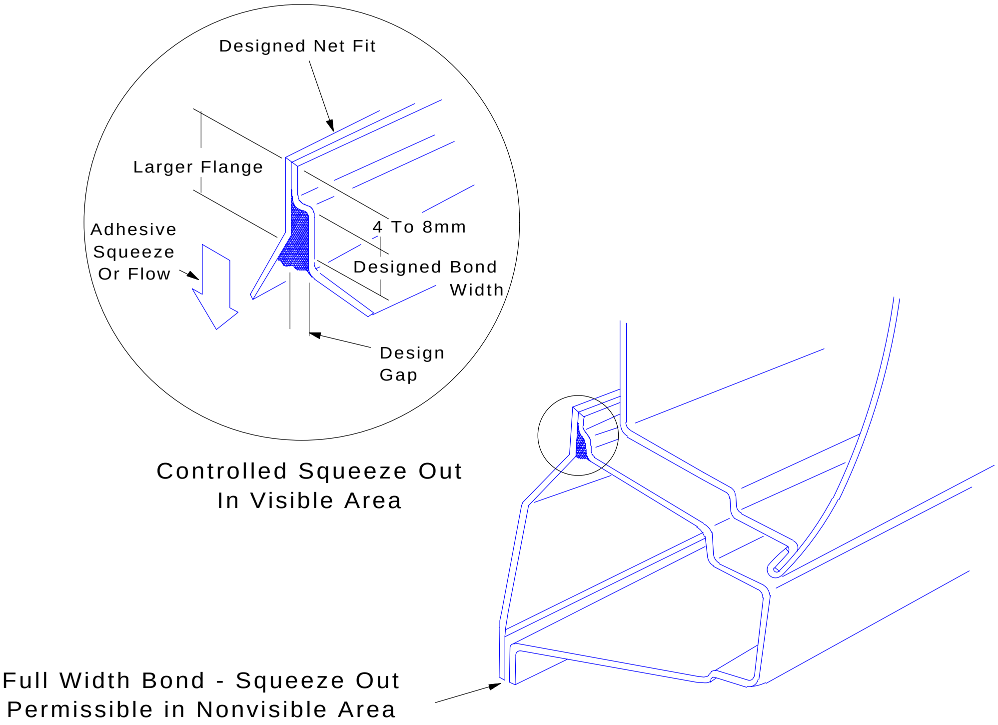


**Figure 3.4.6.3-4** Squeeze out control of bonded joints for visible and nonvisible areas of a sill[18]

Figure 3.4.6.3-5 shows a joint that was designed so that the assembly operation would force any excess adhesive into nonvisible areas. These alternatives allow the joint design to utilize the full flange width.

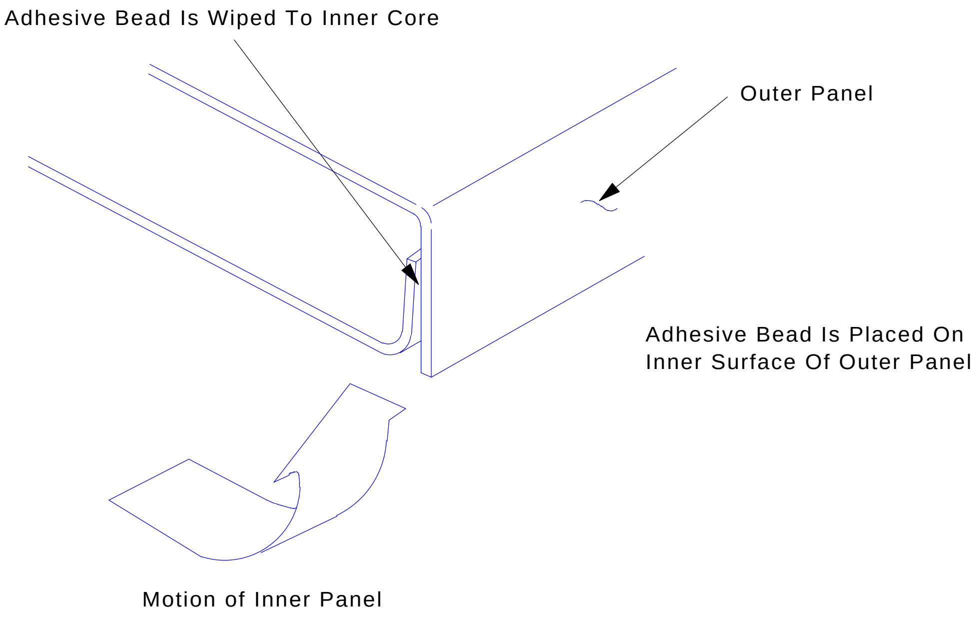


**Figure 3.4.6.3-5** Adhesive squeeze out controlled by assembly operation[18]

The bonded joint may include features such as standoff pads or dimples along the bond surface that provide a gap to control the bondline thickness for consistent bond strength (recall Figure 3.4.5.4.1-1). Standoff pads can additionally provide locations for spot welds or mechanical fastening that will improve the performance of the joint or retain the members while the adhesive cures (Figure 3.4.6.3-6).

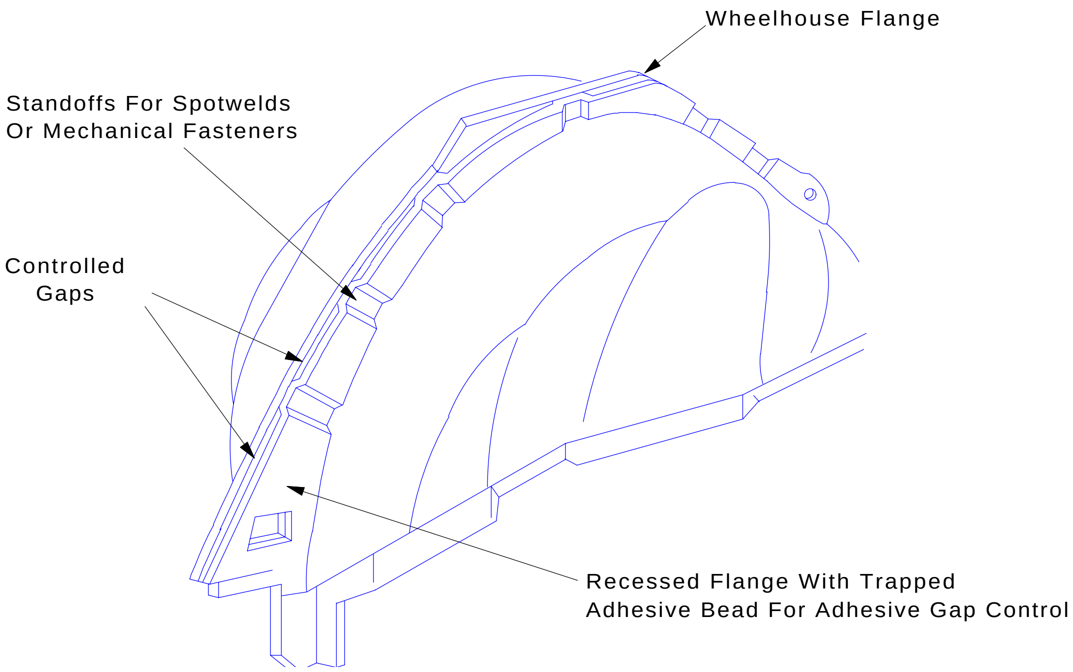


**Figure 3.4.6.3-6** Standoffs utilized to form controlled gaps for adhesive and provide surfaces for spotwelds or mechanical fasteners[18]

### 3.4.6.4 Distribution of Shear Stresses in a Lap Joint

The distribution of shear stresses in the adhesive governs the strength of the joint to a great extent. Shear stresses are distributed uniformly across the width of a lap shear joint (direction perpendicular to the applied load). They are not uniformly distributed along the length (direction parallel to the applied load) because the steel and adhesive layers make up an elastic system in which strain, and consequently stress, is nonuniform. The strength of a lap joint is therefore proportional to its width, but is not proportional to its overlap length (Figure 3.4.6.4-1).

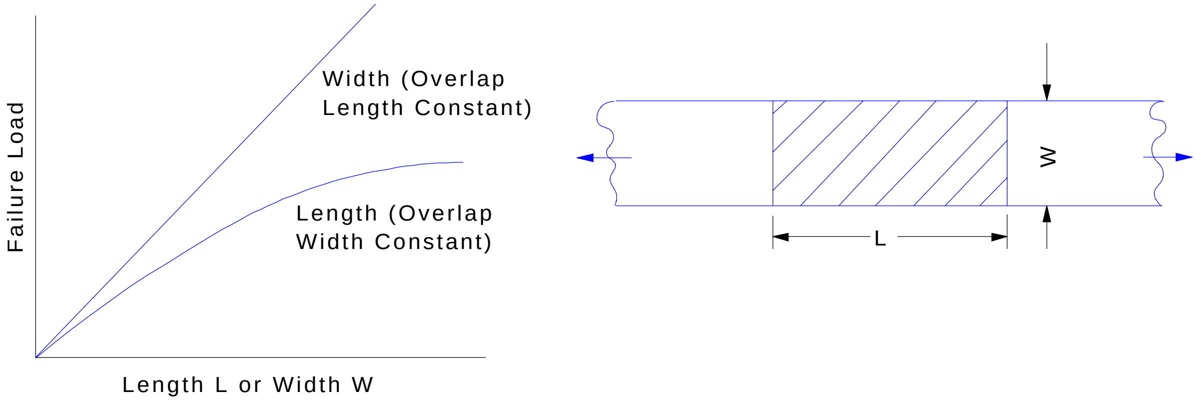


**Figure 3.4.6.4-1** Effect of overlap and width on a typical lap-shear joint

Figure 3.4.6.4-2 shows the typical stress distribution in a simple lap-shear joint.

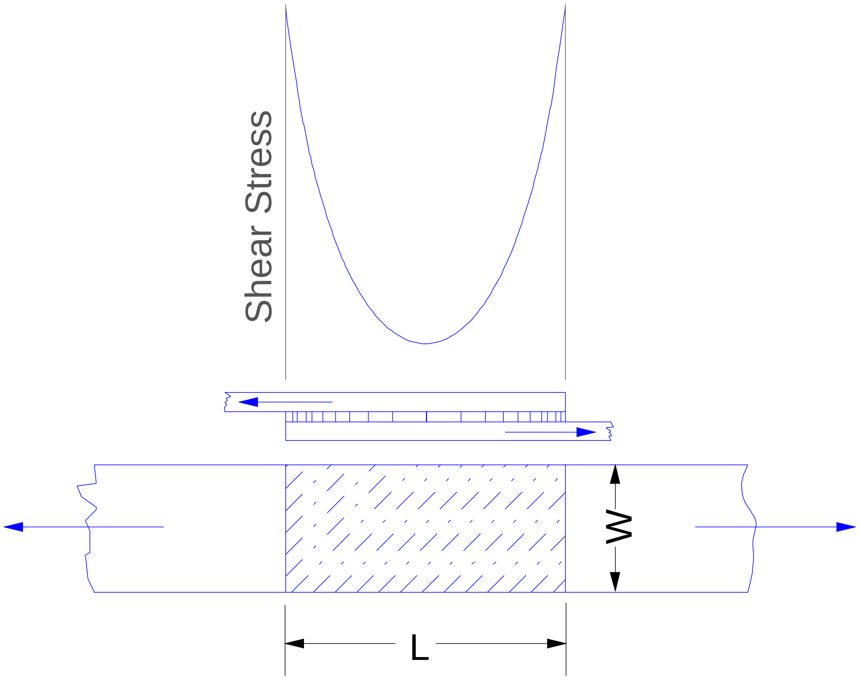


**Figure 3.4.6.4-2** Typical stress distribution in a lap-shear joint parallel to the applied load

The figure indicates that the stress distribution in the direction parallel to the applied load is nonlinear as well as nonuniform. Therefore the strength of the joint cannot be predicted by simple calculations.

The relationship between the mean failure stress of an adhesive, the thickness of the steel, and overlap length has been demonstrated by De Bruyne [20].

Figure 3.4.6.4-3 shows the mean failure stress versus "joint factor", which is the ratio of the square root of metal thickness to overlap length.

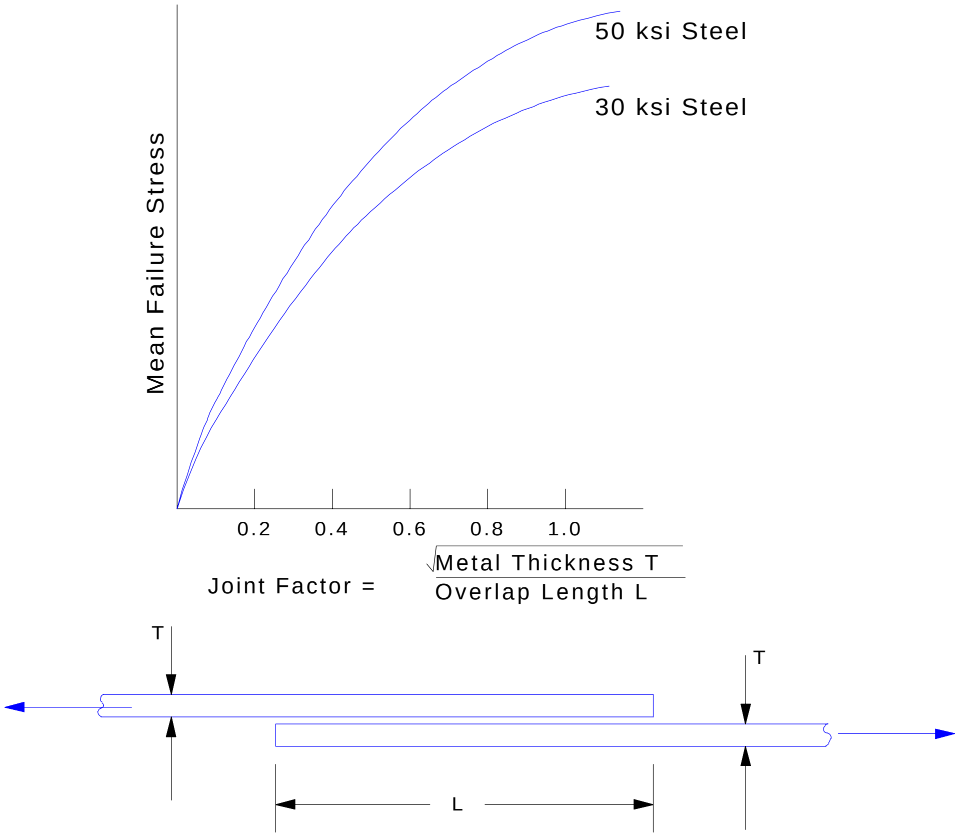


**Figure 3.4.6.4-3** Typical mean failure stress vs. joint factor for a lap-shear joint[19]

At low values of joint factor, the failure stress is proportional to the square root of metal thickness, and essentially independent of the steel yield strength. The figure implies that the designer can vary factors such as metal thickness and length of overlap, within limits, to obtain the desired strength. The joint factor must be determined for each adhesive and for the strength of the steel.

### 3.4.6.5 Selection and Specification of Adhesives

Selecting an adhesive is a very complex process. The designer wants a strong adhesive that will never fail. Management wants the lowest overall cost; that is, material cost, equipment cost, maintenance cost and labor cost must all be minimized. Manufacturing engineering wants an adhesive that is safe and easy to use. The manufacturing manager wants an adhesive that is compatible with existing assembly practices. An applications engineer must select an adhesive that comes close to satisfying the needs of each group.

Fortunately, considerable amounts of information and assistance are available. Adhesive manufacturers and users are developing much needed technical information. The amount of practical experience grows every day as adhesives are used in new applications.

### 3.4.7 WELDBONDING

The term "weldbonding" describes the process of joining steel with a combination of spot welds and a structural adhesive. The process can lead to improved joint stiffness, strength and fatigue life, depending on the adhesive chosen and the spot weld spacing.

Weldbonding processes take advantage of the resistance to peel loading offered by spot welds and the resistance to shear loading offered by the adhesive. In a weld bonded joint, the adhesive distributes the load over a greater area of sheet metal surfaces, which results in a wider stress distribution across the members being joined, and reduced stress levels.

- Under static loading, reduced stresses in some cases increases the spring rate of the joint.
- Under cyclic loading, the more uniform distribution of stresses around the spot welds substantially increases fatigue life.
- Under dynamic loading, the wider uniform stress distribution will enable the joint to absorb more impact energy. Spot welds maintain the integrity of the joint after the adhesive has failed during large plastic deformations. Figure 3.4.7-1 compares the impact energy absorption by spot welded, adhesive bonded and weldbonded joints.

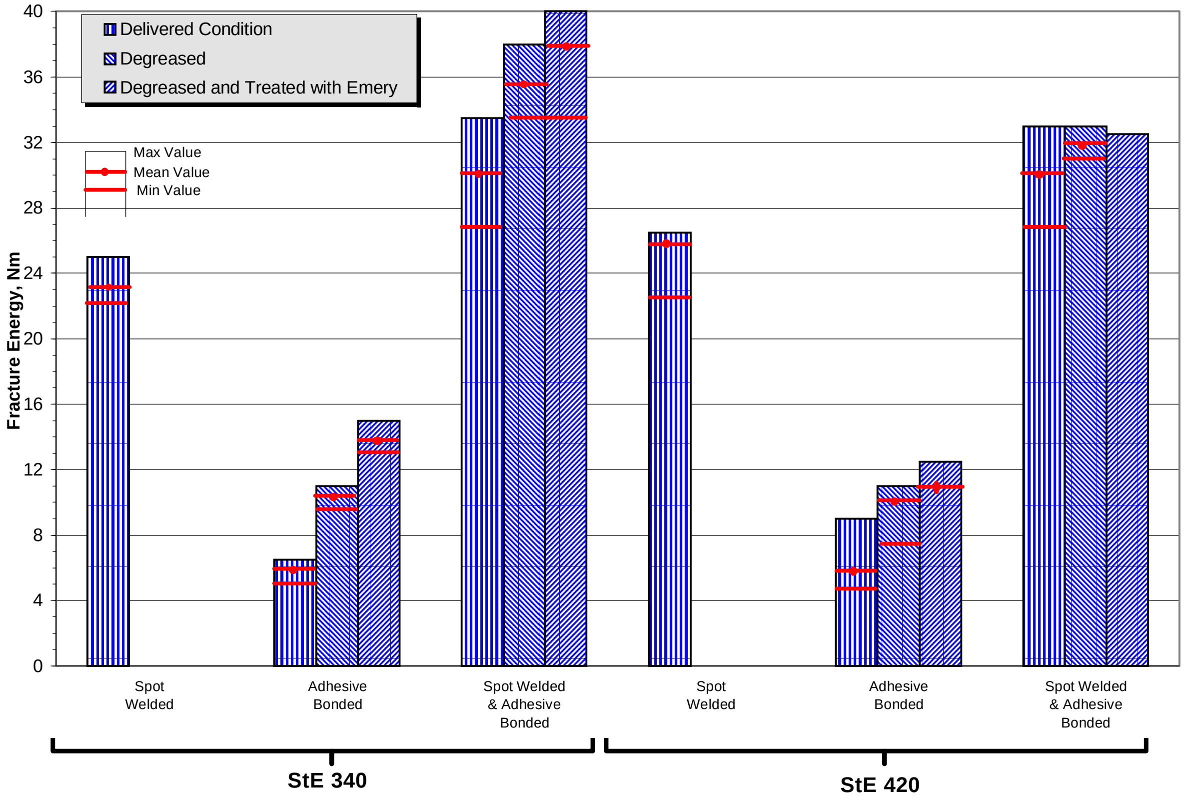


**Figure 3.4.7-1** Impact energy absorption for various joints

Four design parameters affect the performance of weldbonded joints: weld spacing, adhesive type, width of weld flange and the type of loading for which the joint is designed.

Weld spacing will have the greatest effect when an adhesive with a low modulus of elasticity is used. The flexible adhesive allows the spot welds to be loaded, and they contribute more to the overall joint characteristics (refer to Section 3.4.1 Welded Connections, and Section 3.4.4 Spacing of Mechanical Connections for more information on spot welds and their spacing).

Adhesive type will influence weldbonded joint stiffness, strength, and fatigue life since different adhesives have varying moduli and tensile strengths (refer to Section 3.4.5 Adhesives for more information).

Weld flange width contributes directly to the area bonded, and thus the area over which the load is applied. A larger bonded area will decrease the stress level in that location, increasing the fatigue life and increasing the amount of energy that can be absorbed from impact loading.

The type of applied load will affect how a weldbonded joint can perform. A joint designed so that the load will be in shear will best take advantage of the stiffness and strength of adhesives in shear. The spot welds in a joint designed so that the load will be tension or peel will contribute more to the stiffness and strength of the joint than will the adhesive because of the relatively low mechanical properties of adhesives in these types of loading.

Any location on the body-in-white may be a candidate for weld bonding. Some of the most common locations are:

- Areas with difficult weld gun access. Weld flanges that are difficult to reach with spot welding equipment and maintain the desired weld pitch are natural candidates.

- Fatigue life applications. Any joint that is sensitive to fatigue could have structural adhesive added to improve its fatigue life. Weldbonded joints can improve fatigue life while containing the cost and mass incurred by alternatives such as increased metal thickness or added reinforcements. Components loaded by the suspension, or by trailer towing, and latch strikers are examples of fatigue sensitive areas.

- Energy absorption applications. Weldbonded joints have proven to absorb more energy than spot welded joints. Downgaging of front and rear rail sections may be possible if crash energy absorption can be maintained. Structural adhesive can be applied along the entire length of the rails or in strategic locations to control the crash energy absorbed and possibly the crush mode. Potential applications include:

  - Front body hinge pillars, center pillar, and rocker section to increase the energy absorbed during a side impact.
  - Roof rails to absorb more energy during a vehicle rollover.

## 3.4.8 GLOSSARY OF ADHESIVE TERMS

The glossary in Table 3.4.8-1 is applicable to adhesives that are used in automotive body applications. Some of the terms are not used in the text of this manual; they are included for the convenience of the reader who is utilizing other sources.

**Table 3.4.8-1** Glossary of adhesive terms

| Adhesive Term | Definition |
|---|---|
| Adherend | A material joined by an adhesive |
| Bondline Thickness | The gap between the adherends that is filled by adhesive |
| Cure, Curing | Cause and adhesive to develop its final chemical and mechanical properties; more appropriate with thermosetting adhesives (see set) |
| Curing Agent | A chemical component that accomplishes curing |
| Engineering adhesive | A structural adhesive meeting specific performance criteria; e.g., a minimum shear of 500 psi |
| Fixturing | Any technique that supports the joint until the adhesive develops handling strength |
| Gel | (See definition of set) |
| Handling Strength | Strength developed in and adhesive during curing that is less than full cure but sufficient to allow the bonded components to be moved or otherwise handled |
| Hot Melt | An adhesive that bonds upon cooling |
| Induction Curing | A heat curing process where heating is produced by an electromagnetic field |
| Net Fit | Metal-to-metal or adhered-to-adherend contact |
| Non Structural Adhesive | Monomer based composition that may or may not polymerize to form a permanent bonded barrier between two adherends forming a non load bearing joint |
| One Part Adhesive | An adhesive that requires no mixing |
| Open Time | The maximum allowable time period (for a two part adhesive) between mixing the adhesive and joining the adherends |
| Peel | A type of loading that imposes a very high edge stress to the boundary of bonded area (Figure 3.4-18) |
| Peel Strength | Resistance of an adhesively bonded joint to fail in a prying or peel mode; measured in force per unit area |
| Set (Gel) | Convert the adhesive from its dispensed state to a nonflowing state exhibiting some adhesion; more appropriate with thermoplastic adhesive (see cure) |
| Shear | A type of loading that imposes stress that is distributed across the bonded area in the plane of the joint |
| Shear Strength | Resistance to an applied load of an adhesively bonded joint in the plane of the bondline; measured in force per unit area |
| Structural Adhesive | Monomer based composition that polymerizes to form a high modulus, high strength, permanent adhesive to bond two relatively rigid adherends, forming a load bearing joint |
| Thermoplastic | A polymer that repeatedly softens when heated |
| Thermoset | A polymer that cures by crosslinking and does not soften upon reheating |
| Toughening | Decreasing the brittle characteristics of an adhesive by adding appropriate constituents to the formulation |
| Two-Part Adhesive | An adhesive consisting of a resin and a curing agent, which begins curing when the components are mixed |
| Viscosity | The resistance to flow of a liquid |
| Weldbonding | Joining process using adhesive bonding supported by resistance spot welding |

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