Chapter 9

Welding, Bonding, and the Design of Permanent Joints

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</table>
Welding Symbols

- Welding symbol standardized by American Welding Society
- Specifies details of weld on machine drawings

Fig. 9–4
Welding Symbols

- *Arrow side* of a joint is the line, side, area, or near member to which the arrow points.
- The side opposite the arrow side is the *other side*.
- Shape of weld is shown with the symbols below.

<table>
<thead>
<tr>
<th>Bead</th>
<th>Fillet</th>
<th>Plug or slot</th>
<th>Groove</th>
</tr>
</thead>
</table>

Fig. 9–2
Welding Symbol Examples

- Weld leg size of 5 mm
- Fillet weld
- Both sides

- Intermittent and staggered 60 mm along on 200 mm centers

- Leg size of 5 mm
- On one side only (outside)
- Circle indicates all the way around
Welding Symbol Examples

Fig. 9–5
Welding Symbol Examples

Fig. 9–6
Tensile Butt Joint

- Simple butt joint loaded in tension or compression
- Stress is normal stress
  \[ \sigma = \frac{F}{hl} \quad (9-1) \]
- Throat \( h \) does not include extra reinforcement
- Reinforcement adds some strength for static loaded joints
- Reinforcement adds stress concentration and should be ground off for fatigue loaded joints

Fig. 9–7a
Shear Butt Joint

- Simple butt joint loaded in shear
- Average shear stress

\[ \tau = \frac{F}{hl} \]  

Fig. 9–7b
Transverse Fillet Weld

- Joint loaded in tension
- Weld loading is complex

Fig. 9–8

Fig. 9–9
Transverse Fillet Weld

- **Summation of forces**
  \[ F_s = F \sin \theta \]
  \[ F_n = F \cos \theta \]

- **Law of sines**
  \[
  t = \frac{h}{\sin(180^\circ - 45^\circ - \theta)} = \frac{h}{\sin(135^\circ - \theta)} = \frac{\sqrt{2}h}{\cos \theta + \sin \theta}
  \]

- **Solving for throat thickness** \( t \)
  \[ t = \frac{h}{\cos \theta + \sin \theta} \]

Fig. 9–9
Transverse Fillet Weld

- Nominal stresses at angle $\theta$

\[
\tau = \frac{F_s}{A} = \frac{F \sin \theta (\cos \theta + \sin \theta)}{hl} = \frac{F}{hl} (\sin \theta \cos \theta + \sin^2 \theta)
\]

\[
\sigma = \frac{F_n}{A} = \frac{F \cos \theta (\cos \theta + \sin \theta)}{hl} = \frac{F}{hl} (\cos^2 \theta + \sin \theta \cos \theta)
\]

- Von Mises Stress at angle $\theta$

\[
\sigma' = (\sigma^2 + 3\tau^2)^{1/2} = \frac{F}{hl} \left[ (\cos^2 \theta + \sin \theta \cos \theta)^2 + 3(\sin^2 \theta + \sin \theta \cos \theta)^2 \right]^{1/2}
\]

Fig. 9–9
Transverse Fillet Weld

- Largest von Mises stress occurs at $\theta = 62.5^\circ$ with value of $\sigma' = 2.16F/(hl)$
- Maximum shear stress occurs at $\theta = 67.5^\circ$ with value of $\tau_{\text{max}} = 1.207F/(hl)$
Experimental Stresses in Transverse Fillet Weld

- Experimental results are more complex
Transverse Fillet Weld Simplified Model

- No analytical approach accurately predicts the experimentally measured stresses.
- Standard practice is to use a simple and conservative model
- Assume the external load is carried entirely by shear forces on the minimum throat area.

\[ \tau = \frac{F}{0.707hl} = \frac{1.414F}{hl} \]  \hspace{1cm} (9-3)

- By ignoring normal stress on throat, the shearing stresses are inflated sufficiently to render the model conservative.
- By comparison with previous maximum shear stress model, this inflates estimated shear stress by factor of \( \frac{1.414}{1.207} = 1.17 \).
Parallel Fillet Welds

- Same equation also applies for simpler case of simple shear loading in fillet weld

\[
\tau = \frac{F}{0.707hl} = \frac{1.414F}{hl}
\]

(9–3)

Fig. 9–11
Fillet Welds Loaded in Torsion

- Fillet welds carrying both direct shear $V$ and moment $M$

- **Primary shear**
  \[
  \tau' = \frac{V}{A}
  \]

- **Secondary shear**
  \[
  \tau'' = \frac{Mr}{J}
  \]

- $A$ is the throat area of all welds
- $r$ is distance from centroid of weld group to point of interest
- $J$ is second polar moment of area of weld group about centroid of group

Fig. 9–12
Example of Finding $A$ and $J$

- Rectangles represent throat areas. $t = 0.707 \, h$

\[ A = A_1 + A_2 = t_1 d + t_2 b \]

\[ I_x = \frac{t_1 d^3}{12} \quad I_y = \frac{d t_1^3}{12} \]

\[ J_{G1} = I_x + I_y = \frac{t_1 d^3}{12} + \frac{d t_1^3}{12} \]

\[ J_{G2} = \frac{b t_2^3}{12} + \frac{t_2 b^3}{12} \]

\[ \bar{x} = \frac{A_1 x_1 + A_2 x_2}{A} \quad \bar{y} = \frac{A_1 y_1 + A_2 y_2}{A} \]

\[ r_1 = [(\bar{x} - x_1)^2 + (\bar{y} - y)^2]^{1/2} \quad r_2 = [(y_2 - \bar{y})^2 + (x_2 - \bar{x})^2]^{1/2} \]

\[ J = (J_{G1} + A_1 r_1^2) + (J_{G2} + A_2 r_2^2) \]
Example of Finding $A$ and $J$

- Note that $t^3$ terms will be very small compared to $b^3$ and $d^3$
- Usually neglected
- Leaves $J_{G1}$ and $J_{G2}$ linear in weld width
- Can normalize by treating each weld as a line with unit thickness $t$
- Results in *unit second polar moment of area*, $J_u$
- Since $t = 0.707h$,

$$J = 0.707hJ_u$$
# Common Torsional Properties of Fillet Welds (Table 9–1)

<table>
<thead>
<tr>
<th>Weld</th>
<th>Throat Area</th>
<th>Location of G</th>
<th>Unit Second Polar Moment of Area</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>$A = 0.707 , h d$</td>
<td>$\bar{x} = 0$</td>
<td>$J_u = d^3/12$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$\bar{y} = d/2$</td>
<td></td>
</tr>
<tr>
<td>2.</td>
<td>$A = 1.414 , h d$</td>
<td>$\bar{x} = b/2$</td>
<td>$J_u = \frac{d(3b^2 + d^2)}{6}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$\bar{y} = d/2$</td>
<td></td>
</tr>
<tr>
<td>3.</td>
<td>$A = 0.707h(b + d)$</td>
<td>$\bar{x} = \frac{b^2}{2(b + d)}$</td>
<td>$J_u = \frac{(b + d)^4 - 6b^2d^2}{12(b + d)}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$\bar{y} = \frac{d^2}{2(b + d)}$</td>
<td></td>
</tr>
</tbody>
</table>
# Common Torsional Properties of Fillet Welds (Table 9–1)

4.\[
A = 0.707h(2b + d) \quad \bar{x} = \frac{b^2}{2b + d} \quad J_u = \frac{8b^3 + 6bd^2 + d^3}{12} - \frac{b^4}{2b + d}
\]
\[
\bar{y} = d/2
\]

5.\[
A = 1.414h(b + d) \quad \bar{x} = b/2 \quad J_u = \frac{(b + d)^3}{6}
\]
\[
\bar{y} = d/2
\]

6.\[
A = 1.414 \pi hr \quad J_u = 2\pi r^3
\]

*G is centroid of weld group; \( h \) is weld size; plane of torque couple is in the plane of the paper; all welds are of unit width.*
Example 9–1

A 50-kN load is transferred from a welded fitting into a 200-mm steel channel as illustrated in Fig. 9–14. Estimate the maximum stress in the weld.
Example 9–1

(a) Label the ends and corners of each weld by letter. See Fig. 9–15. Sometimes it is desirable to label each weld of a set by number.
Example 9–1

(b) Estimate the primary shear stress \( \tau' \). As shown in Fig. 9–14, each plate is welded to the channel by means of three 6-mm fillet welds. Figure 9–15 shows that we have divided the load in half and are considering only a single plate. From case 4 of Table 9–1 we find the throat area as

\[
A = 0.707(6)[2(56) + 190] = 1280 \text{ mm}^2
\]

Then the primary shear stress is

\[
\tau' = \frac{V}{A} = \frac{25(10)^3}{1280} = 19.5 \text{ MPa}
\]
Example 9–1

(c) Draw the $\tau'$ stress, to scale, at each lettered corner or end. See Fig. 9–16.
(d) Locate the centroid of the weld pattern. Using case 4 of Table 9–1, we find

$$\bar{x} = \frac{(56)^2}{2(56) + 190} = 10.4 \text{ mm}$$

This is shown as point $O$ on Figs. 9–15 and 9–16.
(e) Find the distances $r_i$ (see Fig. 9–16):

$$r_A = r_B = \left[\frac{(190/2)^2 + (56 - 10.4)^2}{1/2} = 105 \text{ mm}\right.$$  

$$r_C = r_D = \left[\frac{(190/2)^2 + (10.4)^2}{1/2} = 95.6 \text{ mm}\right.$$
Example 9–1

(f) Find $J$. Using case 4 of Table 9–1 again, with Eq. (9–6), we get

$$J = 0.707(6) \left[ \frac{8(56)^3 + 6(56)(190)^2 + (190)^3}{12} - \frac{(56)^4}{2(56) + 190} \right]$$

$$= 7.07(10)^6 \text{ mm}^4$$

(g) Find $M$:

$$M = Fl = 25(100 + 10.4) = 2760 \text{ N} \cdot \text{m}$$

(h) Estimate the secondary shear stresses $\tau''$ at each lettered end or corner:

$$\tau''_A = \tau''_B = \frac{Mr}{J} = \frac{2760(10)^3(105)}{7.07(10)^6} = 41.0 \text{ MPa}$$

$$\tau''_C = \tau''_D = \frac{2760(10)^3(95.6)}{7.07(10)^6} = 37.3 \text{ MPa}$$
Example 9–1

(i) Draw the \( \tau'' \) stress at each corner and end. See Fig. 9–16. Note that this is a free-body diagram of one of the side plates, and therefore the \( \tau' \) and \( \tau'' \) stresses represent what the channel is doing to the plate (through the welds) to hold the plate in equilibrium.
Example 9–1

At each point labeled, combine the two stress components as vectors (since they apply to the same area). At point A, the angle that $\tau_A''$ makes with the vertical, $\alpha$, is also the angle $r_A$ makes with the horizontal, which is $\alpha = \tan^{-1}(45.6/95) = 25.64^\circ$. This angle also applies to point B. Thus

$$\tau_A = \tau_B = \sqrt{(19.5 - 41.0 \sin 25.64^\circ)^2 + (41.0 \cos 25.64^\circ)^2} = 37.0 \text{ MPa}$$

Similarly, for C and D, $\beta = \tan^{-1}(10.4/95) = 6.25^\circ$. Thus

$$\tau_C = \tau_D = \sqrt{(19.5 + 37.3 \sin 6.25^\circ)^2 + (37.3 \cos 6.25^\circ)^2} = 43.9 \text{ MPa}$$

Identify the most highly stressed point: $\tau_{\text{max}} = \tau_C = \tau_D = 43.9 \text{ MPa}$
Fillet Welds Loaded in Bending

- Fillet welds carry both shear $V$ and moment $M$

\[
\tau' = \frac{V}{A}
\]

\[
I_u = \frac{bd^2}{2} \quad I = 0.707h I_u = 0.707h \frac{bd^2}{2}
\]

\[
\tau'' = \frac{Mc}{I} = \frac{Md/2}{0.707hbd^2/2} = \frac{1.414M}{bdh}
\]

\[
\tau = (\tau'^2 + \tau''^2)^{1/2}
\]

Fig. 9–17
## Bending Properties of Fillet Welds (Table 9–2)

<table>
<thead>
<tr>
<th>Weld</th>
<th>Throat Area</th>
<th>Location of $G$</th>
<th>Unit Second Moment of Area</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.</td>
<td>$A = 0.707hd$</td>
<td>$\bar{x} = 0$</td>
<td>$I_u = \frac{d^3}{12}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$\bar{y} = d/2$</td>
<td></td>
</tr>
<tr>
<td>2.</td>
<td>$A = 1.414hd$</td>
<td>$\bar{x} = b/2$</td>
<td>$I_u = \frac{d^3}{6}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$\bar{y} = d/2$</td>
<td></td>
</tr>
<tr>
<td>3.</td>
<td>$A = 1.414hb$</td>
<td>$\bar{x} = b/2$</td>
<td>$I_u = \frac{bd^2}{2}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$\bar{y} = d/2$</td>
<td></td>
</tr>
<tr>
<td>4.</td>
<td>$A = 0.707h(2b + d)$</td>
<td>$\bar{x} = \frac{b^2}{2b + d}$</td>
<td>$I_u = \frac{d^2}{12}(6b + d)$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$\bar{y} = d/2$</td>
<td></td>
</tr>
</tbody>
</table>
# Bending Properties of Fillet Welds (Table 9–2)

<table>
<thead>
<tr>
<th>Case</th>
<th>Area Formula</th>
<th>Centroidal y-coordinates</th>
<th>Moment of Inertia Formula</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.</td>
<td>( A = 0.707h(b + 2d) )</td>
<td>( \bar{y} = \frac{d^2}{b + 2d} )</td>
<td>( I_u = \frac{2d^3}{3} - 2d^2\bar{y} + (b + 2d)\bar{y}^2 )</td>
</tr>
<tr>
<td>6.</td>
<td>( A = 1.414h(b + d) )</td>
<td>( \bar{y} = \frac{d}{2} )</td>
<td>( I_u = \frac{d^2}{6}(3b + d) )</td>
</tr>
<tr>
<td>7.</td>
<td>( A = 0.707h(b + 2d) )</td>
<td>( \bar{y} = \frac{d^2}{b + 2d} )</td>
<td>( I_u = \frac{2d^3}{3} - 2d^2\bar{y} + (b + 2d)\bar{y}^2 )</td>
</tr>
<tr>
<td>8.</td>
<td>( A = 1.414h(b + d) )</td>
<td>( \bar{y} = \frac{d}{2} )</td>
<td>( I_u = \frac{d^2}{6}(3b + d) )</td>
</tr>
<tr>
<td>9.</td>
<td>( A = 1.414\pi hr )</td>
<td></td>
<td>( l_u = \pi r^3 )</td>
</tr>
</tbody>
</table>
Strength of Welded Joints

- Must check for failure in parent material and in weld
- Weld strength is dependent on choice of electrode material
- Weld material is often stronger than parent material
- Parent material experiences heat treatment near weld
- Cold drawn parent material may become more like hot rolled in vicinity of weld
- Often welded joints are designed by following codes rather than designing by the conventional factor of safety method
## Minimum Weld-Metal Properties (Table 9–3)

<table>
<thead>
<tr>
<th>AWS Electrode Number*</th>
<th>Tensile Strength kpsi (MPa)</th>
<th>Yield Strength, kpsi (MPa)</th>
<th>Percent Elongation</th>
</tr>
</thead>
<tbody>
<tr>
<td>E60xx</td>
<td>62 (427)</td>
<td>50 (345)</td>
<td>17–25</td>
</tr>
<tr>
<td>E70xx</td>
<td>70 (482)</td>
<td>57 (393)</td>
<td>22</td>
</tr>
<tr>
<td>E80xx</td>
<td>80 (551)</td>
<td>67 (462)</td>
<td>19</td>
</tr>
<tr>
<td>E90xx</td>
<td>90 (620)</td>
<td>77 (531)</td>
<td>14–17</td>
</tr>
<tr>
<td>E100xx</td>
<td>100 (689)</td>
<td>87 (600)</td>
<td>13–16</td>
</tr>
<tr>
<td>E120xx</td>
<td>120 (827)</td>
<td>107 (737)</td>
<td>14</td>
</tr>
</tbody>
</table>

*The American Welding Society (AWS) specification code numbering system for electrodes. This system uses an E prefixed to a four- or five-digit numbering system in which the first two or three digits designate the approximate tensile strength. The last digit includes variables in the welding technique, such as current supply. The next-to-last digit indicates the welding position, as, for example, flat, or vertical, or overhead. The complete set of specifications may be obtained from the AWS upon request.
### Stresses Permitted by the AISC Code for Weld Metal

#### Table 9–4

<table>
<thead>
<tr>
<th>Type of Loading</th>
<th>Type of Weld</th>
<th>Permissible Stress</th>
<th>n*</th>
</tr>
</thead>
<tbody>
<tr>
<td>Tension</td>
<td>Butt</td>
<td>0.60S_y</td>
<td>1.67</td>
</tr>
<tr>
<td>Bearing</td>
<td>Butt</td>
<td>0.90S_y</td>
<td>1.11</td>
</tr>
<tr>
<td>Bending</td>
<td>Butt</td>
<td>0.60–0.66S_y</td>
<td>1.52–1.67</td>
</tr>
<tr>
<td>Simple compression</td>
<td>Butt</td>
<td>0.60S_y</td>
<td>1.67</td>
</tr>
<tr>
<td>Shear</td>
<td>Butt or fillet</td>
<td>0.30S_{utt}</td>
<td></td>
</tr>
</tbody>
</table>

*The factor of safety n has been computed by using the distortion-energy theory.

†Shear stress on base metal should not exceed 0.40S_y of base metal.
Fatigue Stress-Concentration Factors

- $K_{fs}$ appropriate for application to shear stresses
- Use for parent metal and for weld metal

**Table 9-5**

<table>
<thead>
<tr>
<th>Type of Weld</th>
<th>$K_{fs}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reinforced butt weld</td>
<td>1.2</td>
</tr>
<tr>
<td>Toe of transverse fillet weld</td>
<td>1.5</td>
</tr>
<tr>
<td>End of parallel fillet weld</td>
<td>2.7</td>
</tr>
<tr>
<td>T-butt joint with sharp corners</td>
<td>2.0</td>
</tr>
</tbody>
</table>
### Allowable Load or Various Sizes of Fillet Welds (Table 9–6)

<table>
<thead>
<tr>
<th>Strength Level of Weld Metal (EXX)</th>
<th>60*</th>
<th>70*</th>
<th>80</th>
<th>90*</th>
<th>100</th>
<th>110*</th>
<th>120</th>
</tr>
</thead>
<tbody>
<tr>
<td>Allowable shear stress on throat, ksi (1000 psi) of fillet weld or partial penetration groove weld</td>
<td>18.0</td>
<td>21.0</td>
<td>24.0</td>
<td>27.0</td>
<td>30.0</td>
<td>33.0</td>
<td>36.0</td>
</tr>
<tr>
<td>( \tau = )</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Allowable Unit Force on Fillet Weld, kip/linear in</th>
</tr>
</thead>
<tbody>
<tr>
<td>( f = )</td>
</tr>
</tbody>
</table>

<table>
<thead>
<tr>
<th>Leg Size ( h ), in</th>
<th>Allowable Unit Force for Various Sizes of Fillet Welds, kip/linear in</th>
</tr>
</thead>
<tbody>
<tr>
<td>5/8</td>
<td>7.96</td>
</tr>
<tr>
<td>1/2</td>
<td>6.37</td>
</tr>
<tr>
<td>7/16</td>
<td>5.57</td>
</tr>
<tr>
<td>3/8</td>
<td>4.77</td>
</tr>
<tr>
<td>5/16</td>
<td>3.98</td>
</tr>
<tr>
<td>1/4</td>
<td>3.18</td>
</tr>
<tr>
<td>3/16</td>
<td>2.39</td>
</tr>
<tr>
<td>1/8</td>
<td>1.59</td>
</tr>
<tr>
<td>1/16</td>
<td>0.795</td>
</tr>
</tbody>
</table>

*Fillet welds actually tested by the joint AISC-AWS Task Committee.

\( f = 0.707h \tau_{all} \).
## Minimum Fillet Weld Size, $h$ (Table 9–6)

<table>
<thead>
<tr>
<th>Material Thickness of Thicker Part Joined, in</th>
<th>Weld Size, in</th>
</tr>
</thead>
<tbody>
<tr>
<td>*To $\frac{1}{4}$ incl.</td>
<td>$\frac{1}{8}$</td>
</tr>
<tr>
<td>Over $\frac{1}{4}$</td>
<td>To $\frac{1}{2}$</td>
</tr>
<tr>
<td>Over $\frac{1}{2}$</td>
<td>To $\frac{3}{4}$</td>
</tr>
<tr>
<td>†Over $\frac{3}{4}$</td>
<td>To $1\frac{1}{2}$</td>
</tr>
<tr>
<td>Over $1\frac{1}{2}$</td>
<td>To $2\frac{1}{4}$</td>
</tr>
<tr>
<td>Over $2\frac{1}{4}$</td>
<td>To 6</td>
</tr>
<tr>
<td>Over 6</td>
<td></td>
</tr>
</tbody>
</table>

Not to exceed the thickness of the thinner part.

*Minimum size for bridge application does not go below $\frac{3}{16}$ in.

†For minimum fillet weld size, schedule does not go above $\frac{5}{16}$ in fillet weld for every $\frac{3}{4}$ in material.
Resistance Welding

- Welding by passing an electric current through parts that are pressed together
- Common forms are *spot welding* and *seam welding*
- Failure by shear of weld or tearing of member
- Avoid loading joint in tension to avoid tearing

Fig. 9–23
Adhesive Bonding

- Adhesive bonding has unique advantages
- Reduced weight, sealing capabilities, reduced part count, reduced assembly time, improved fatigue and corrosion resistance, reduced stress concentration associated with bolt holes

Fig. 9–24
Types of Adhesives

- May be classified by
  - Chemistry
    - Epoxies, polyurethanes, polyimides
  - Form
    - Paste, liquid, film, pellets, tape
  - Type
    - Hot melt, reactive hot melt, thermosetting, pressure sensitive, contact
  - Load-carrying capability
    - Structural, semi-structural, non-structural
# Mechanical Performance of Various Types of Adhesives

<table>
<thead>
<tr>
<th>Adhesive Chemistry or Type</th>
<th>Room Temperature Lap-Shear Strength, MPa (psi)</th>
<th>Peel Strength per Unit Width, kN/m (lbf/in)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Pressure-sensitive</td>
<td>0.01–0.07 (2–10)</td>
<td>0.18–0.88 (1–5)</td>
</tr>
<tr>
<td>Starch-based</td>
<td>0.07–0.7 (10–100)</td>
<td>0.18–0.88 (1–5)</td>
</tr>
<tr>
<td>Cellosics</td>
<td>0.35–3.5 (50–500)</td>
<td>0.18–1.8 (1–10)</td>
</tr>
<tr>
<td>Rubber-based</td>
<td>0.35–3.5 (50–500)</td>
<td>1.8–7 (10–40)</td>
</tr>
<tr>
<td>Formulated hot melt</td>
<td>0.35–4.8 (50–700)</td>
<td>0.88–3.5 (5–20)</td>
</tr>
<tr>
<td>Synthetically designed hot melt</td>
<td>0.7–6.9 (100–1000)</td>
<td>0.88–3.5 (5–20)</td>
</tr>
<tr>
<td>PVAc emulsion (white glue)</td>
<td>1.4–6.9 (200–1000)</td>
<td>0.88–1.8 (5–10)</td>
</tr>
<tr>
<td>Cyanoacrylate</td>
<td>6.9–13.8 (1000–2000)</td>
<td>0.18–3.5 (1–20)</td>
</tr>
<tr>
<td>Protein-based</td>
<td>6.9–13.8 (1000–2000)</td>
<td>0.18–1.8 (1–10)</td>
</tr>
<tr>
<td>Anaerobic acrylic</td>
<td>6.9–13.8 (1000–2000)</td>
<td>0.18–1.8 (1–10)</td>
</tr>
<tr>
<td>Urethane</td>
<td>6.9–17.2 (1000–2500)</td>
<td>1.8–8.8 (10–50)</td>
</tr>
<tr>
<td>Rubber-modified acrylic</td>
<td>13.8–24.1 (2000–3500)</td>
<td>1.8–8.8 (10–50)</td>
</tr>
<tr>
<td>Modified phenolic</td>
<td>13.8–27.6 (2000–4000)</td>
<td>3.6–7 (20–40)</td>
</tr>
<tr>
<td>Unmodified epoxy</td>
<td>10.3–27.6 (1500–4000)</td>
<td>0.35–1.8 (2–10)</td>
</tr>
<tr>
<td>Bis-maleimide</td>
<td>13.8–27.6 (2000–4000)</td>
<td>0.18–3.5 (1–20)</td>
</tr>
<tr>
<td>Polyimide</td>
<td>13.8–27.6 (2000–4000)</td>
<td>0.18–0.88 (1–5)</td>
</tr>
<tr>
<td>Rubber-modified epoxy</td>
<td>20.7–41.4 (3000–6000)</td>
<td>4.4–14 (25–80)</td>
</tr>
</tbody>
</table>

Table 9–7
Stress Distributions

- Adhesive joints are much stronger in shear loading than tensile loading.
- Lap-shear joints are important for test specimens and for practical designs.
- Simplest analysis assumes uniform stress distribution over bonded area.
- Most joints actually experience significant peaks of stress.

Fig. 9-25
Double-lap Joint

- Classic analysis of double-lap joint known as shear-lag model
- Double joint eliminates complication of bending from eccentricity

![Diagram of double-lap joint](image-url)

Fig. 9–26
Double-lap Joint

- Shear-stress distribution is given by

\[
\tau(x) = \frac{P \omega}{4b \sinh(\omega l/2)} \cosh(\omega x) + \left[ \frac{P \omega}{4b \cosh(\omega l/2)} \left( \frac{2E_o t_o - E_i t_i}{2E_o t_o + E_i t_i} \right) \right] \sinh(\omega x) + \frac{(\alpha_i - \alpha_o) \Delta T \omega}{(1/E_o t_o + 2/E_i t_i) \cosh(\omega l/2)} \sinh(\omega x)
\]

where

\[
\omega = \sqrt{\frac{G}{h} \left( \frac{1}{E_o t_o} + \frac{2}{E_i t_i} \right)}
\]

Fig. 9–26b
Single-lap Joint

- Eccentricity introduces bending
- Bending can as much as double the resulting shear stresses
- Near ends of joint peel stresses can be large, causing joint failure

Fig. 9–28
Single-lap Joint

- Shear and peal stresses in single-lap joint, as calculated by Goland and Reissner

- Volkersen curve is for double-lap joint

ASTM D 1002-94

$l = 0.5$ in (12.7 mm)
$t = 0.064$ in (1.6 mm)
Aluminum: $E = 10$ Msi (70 GPa)
Epoxy: $E_a = 500$ ksi (3.5 GPa)

Stresses shown for an applied load of $P = 1000$ lbf (4.4 kN)

Note: For very long joints, Volkersen predicts only 50% of the G-R shear stress.

Shigley's Mechanical Engineering Design Fig. 9–28 (d)
Adhesive Joint Design Guidelines

- Design to place bondline in shear, not peel.
- Use adhesives with adequate ductility to reduce stress concentrations and increase toughness to resist debond propagation.
- Recognize environmental limitations of adhesives and surface preparation.
- Design to facilitate inspection.
- Allow sufficient bond area to tolerate some debonding before becoming critical.
- Attempt to bond to multiple surfaces to support loads in any direction.
- Consider using adhesives in conjunction with spot welds, rivets, or bolts.
Design Ideas for Improved Bonding

Original

Improved

Fig. 9–29
Design Ideas for Improved Bonding

Original

Improved

Fig. 9–29
Design Ideas for Improved Bonding

Peel stresses can be a problem at ends of lap joints of all types

Tapered to reduce peel
Mechanically reduce peel

Rivet, spot weld, or bolt to reduce peel
Larger bond area to reduce peel

Fig. 9–29