Chapter 9

Welding, Bonding, and the Design of Permanent Joints
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Welding is the process of joining two pieces of metal together by hammering, pressure or fusion. Filler metal may or may not be used.

The strongest and most common method of permanently joining steel components together.

Arc welding is the most important since it is adaptable to various manufacturing environments and is relatively cheap.

A weldment is fabricated by welding together a collection of metal shapes.
Introduction

- A pool of molten metal in which the components and electrode material coalesce, forming a homogeneous whole (ideally) when the pool later resolidifies.

- The materials of components and electrode must be compatible from the point of view of strength, ductility and metallurgy.
Welding Symbols

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

Fig. 9–4
AWS Standard

- **Weld symbol**
  - Graphic symbol that indicates weld required

- **Welding symbol**
  - Following eight elements:
    - Reference line (required)
    - Arrow (required)
    - Basic weld symbols
    - Dimensions and other data
    - Supplementary symbols
    - Finish symbols
    - Tail
    - Other specifications
Welding Symbol Components

Reference line (required) is always drawn horizontally.

Arrow (required) can point up or down and can be moved to other end of reference line.

Tail (optional; only required if welding references are needed such as a WPS no.) is always on the end of the reference line opposite the arrow.
Welding Symbols

- Finish symbol
- Contour symbol
- Root opening; depth of filling for plug and slot welds
- Size; size or strength for resistance welds
- Reference line
- Groove angle; included angle of countersink for plug welds
- Length of weld
- Pitch (center-to-center spacing) of welds
- Arrow connecting reference line to arrow side of joint, to grooved member, or both
- Specification; process; or other reference
- Tail (may be omitted when reference is not used)
- Basic weld symbol or detail reference

Fig. 9–1

Shigley’s Mechanical Engineering Design
Weld symbols on drawings

Joints in drawings may be indicated:

• by detailed sketches, showing every dimension

• by symbolic representation
Elementary Weld Symbols

Square Groove Weld

Single V Groove Weld
Elementary Weld Symbols

Single Bevel Groove Weld

Single V Groove Weld with Broad Root Face
Elementary Weld Symbols

Single Bevel Groove Weld with Broad Root Face

Single U Groove Weld
Elementary Weld Symbols

Single J Groove Weld

Backing Weld
Elementary Weld Symbols

Fillet Weld

Plug / Slot Weld
Elementary Weld Symbols

Spot Weld

Seam Weld
Elementary Weld Symbols

Edge Weld

Surfacing
SUPPLEMENTARY SYMBOLS

Weld Profile

- Flat
- Convex
- Concave
SUPPLEMENTARY SYMBOLS

Toes blended smoothly

Permanent Backing Strip

Removable Backing Strip
SUPPLEMENTARY SYMBOLS

Peripheral Welds
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.

![Welding Symbols](image)

<table>
<thead>
<tr>
<th>Type of weld</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bead</td>
</tr>
<tr>
<td>---</td>
</tr>
<tr>
<td>[Image]</td>
</tr>
</tbody>
</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
  \[ \sigma = \frac{F}{hl} \]  
  (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

\[ \tau = \frac{F}{hl} \]  \hspace{1cm} (9-2)

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

- 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 $\tau_{\text{max}} = 1.207F/(hl)$

Fig. 9–9

Shigley’s Mechanical Engineering Design
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
\[ I = \bar{I} + Ad^2 \quad \text{parallel axis theorem} \]
Example of Finding $A$ and $J$

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

\[ A = A_1 + A_2 = t_1d + t_2b \]

\[ I_x = \frac{t_1d^3}{12} \quad I_y = \frac{dt_1^3}{12} \]

\[ J_{G1} = I_x + I_y = \frac{t_1d^3}{12} + \frac{dt_1^3}{12} \]

\[ J_{G2} = \frac{bt_2^3}{12} + \frac{t_2b^3}{12} \]

\[ \bar{x} = \frac{A_1x_1 + A_2x_2}{A} \quad \bar{y} = \frac{A_1y_1 + A_2y_2}{A} \]

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

Using the parallel axis theorem, the second polar moment of area of the weld group is

\[ J = (J_{G1} + A_1r_1^2) + (J_{G2} + A_2r_2^2) \]
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 $t = 0.707h$, 

$$J = 0.707hJ_u$$

$$J_{G1} = I_x + I_y = \frac{t_1 d^3}{12} + \frac{dt_1^3}{12}$$

$$J_{G2} = \frac{bt_2^3}{12} + \frac{t_2 b^3}{12}$$

Fig. 9–13
**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>$J_u = \frac{d^3}{12}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$\bar{y} = \frac{d}{2}$</td>
<td></td>
</tr>
<tr>
<td>2.</td>
<td>$A = 1.414 , h , d$</td>
<td>$\bar{x} = \frac{b}{2}$</td>
<td>$J_u = \frac{d(3b^2 + d^2)}{6}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$\bar{y} = \frac{d}{2}$</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.  
\[ A = 0.707h(2b + d) \quad \bar{x} = \frac{b^2}{2b + d} \quad J_u = \frac{8b^3 + 6bd^2 + d^3}{12} - \frac{b^4}{2b + d} \]

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

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.

Fig. 9–15
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):

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

\[
r_C = r_D = \left[\frac{(190/2)^2 + (10.4)^2}{2}\right]^{1/2} = 95.6 \text{ mm}
\]
(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

(j) 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

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

(k) 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 \( M \)

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

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

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

\[
\tau = (\tau'^2 + \tau''^2)^{1/2}
\]
## 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 ) , ( \bar{y} = d/2 )</td>
<td>( I_u = \frac{d^3}{12} )</td>
</tr>
<tr>
<td>2.</td>
<td>( A = 1.414hd )</td>
<td>( \bar{x} = b/2 ) , ( \bar{y} = d/2 )</td>
<td>( I_u = \frac{d^3}{6} )</td>
</tr>
<tr>
<td>3.</td>
<td>( A = 1.414hb )</td>
<td>( \bar{x} = b/2 ) , ( \bar{y} = d/2 )</td>
<td>( I_u = \frac{bd^2}{2} )</td>
</tr>
<tr>
<td>4.</td>
<td>( A = 0.707h(2b + d) )</td>
<td>( \bar{x} = \frac{b^2}{2b + d} ) , ( \bar{y} = d/2 )</td>
<td>( I_u = \frac{d^2}{12}(6b + d) )</td>
</tr>
</tbody>
</table>
Bending Properties of Fillet Welds (Table 9–2)

5. 

\[ A = 0.707h(b + 2d) \]
\[ \bar{x} = b/2 \]
\[ \bar{y} = \frac{d^2}{b + 2d} \]
\[ I_u = \frac{2d^3}{3} - 2d^2 \bar{y} + (b + 2d)\bar{y}^2 \]

6. 

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

7. 

\[ A = 0.707h(b + 2d) \]
\[ \bar{x} = b/2 \]
\[ \bar{y} = \frac{d^2}{b + 2d} \]
\[ I_u = \frac{2d^3}{3} - 2d^2 \bar{y} + (b + 2d)\bar{y}^2 \]

8. 

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

9. 

\[ A = 1.414\pi hr \]
\[ l_u = \pi r^3 \]
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>0.60( S_y )</td>
<td>1.67</td>
</tr>
<tr>
<td>Bearing</td>
<td>Butt</td>
<td>0.90( S_y )</td>
<td>1.11</td>
</tr>
<tr>
<td>Bending</td>
<td>Butt</td>
<td>0.60–0.66( S_y )</td>
<td>1.52–1.67</td>
</tr>
<tr>
<td>Simple compression</td>
<td>Butt</td>
<td>0.60( S_y )</td>
<td>1.67</td>
</tr>
<tr>
<td>Shear</td>
<td>Butt or fillet</td>
<td>0.30( S_{utt} )</td>
<td></td>
</tr>
</tbody>
</table>

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

\( ^\dagger \)Shear stress on base metal should not exceed 0.40\( S_y \) of base metal.
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>Allowable Unit Force on Fillet Weld, kip/linear in</td>
<td>$f = 12.73h$</td>
<td>$14.85h$</td>
<td>$16.97h$</td>
<td>$19.09h$</td>
<td>$21.21h$</td>
<td>$23.33h$</td>
<td>$25.45h$</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.

$\tau$ = 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.
Example 9–2

A $\frac{1}{2}$-in by 2-in rectangular-cross-section 1015 bar carries a static load of 16.5 kip. It is welded to a gusset plate with a $\frac{3}{8}$-in fillet weld 2 in long on both sides with an E70XX electrode as depicted in Fig. 9–18. Use the welding code method.

(a) Is the weld metal strength satisfactory?
(b) Is the attachment strength satisfactory?

Table A-20, $S_y = 27.5$ kpsi

Fig. 9–18
Example 9–2

(a) From Table 9–6, allowable force per unit length for a $\frac{3}{8}$-in E70 electrode metal is 5.57 kip/in of weldment; thus

$$F = 5.57l = 5.57(4) = 22.28 \text{ kip}$$

Since 22.28 > 16.5 kip, weld metal strength is satisfactory.

(b) Check shear in attachment adjacent to the welds. From Table A–20, $S_y = 27.5$ kpsi.

Then, from Table 9–4, the allowable attachment shear stress is

$$\tau_{all} = 0.4S_y = 0.4(27.5) = 11 \text{ kpsi}$$

The shear stress $\tau$ on the base metal adjacent to the weld is

$$\tau = \frac{F}{2hl} = \frac{16.5}{2(0.375)2} = 11 \text{ kpsi}$$

$\boxed{h = \frac{3}{8} = 0.375 \text{ in}}$

$\boxed{t = \frac{1}{2} \text{ in}}$

$\boxed{l = 2 \text{ in}}$
Example 9–2

Since $\tau_{all} > \tau$, the attachment is satisfactory near the weld beads. The tensile stress in the shank of the attachment $\sigma$ is

$$\sigma = \frac{F}{tl} = \frac{16.5}{(1/2)2} = 16.5 \text{ kpsi}$$

The allowable tensile stress $\sigma_{all}$, from Table 9–4, is $0.6S_y$ and, with welding code safety level preserved,

$$\sigma_{all} = 0.6S_y = 0.6(27.5) = 16.5 \text{ kpsi}$$

Since $\sigma \leq \sigma_{all}$, the shank tensile stress is satisfactory.
Example 9–4

Perform an adequacy assessment of the statically loaded welded cantilever carrying 500 lbf depicted in Fig. 9–20. The cantilever is made of AISI 1018 HR steel and welded with a $\frac{3}{8}$-in fillet weld as shown in the figure. An E6010 electrode was used, and the design factor was 3.0.

(a) Use the conventional method for the weld metal.
(b) Use the conventional method for the attachment (cantilever) metal.
(c) Use a welding code for the weld metal.

Table A-20, $S_y = 32$ kpsi, $S_{ut} = 58$ kpsi

Table 9-3, $S_y = 50$ kpsi, $S_{ut} = 62$ kpsi

$$\bar{x} = b/2$$

$$\bar{y} = d/2$$

$$A = 1.414hd$$

$$I_u = \frac{d^3}{6}$$

Fig. 9–20
Example 9–4

(a) From Table 9–3, $S_y = 50$ kpsi, $S_{ut} = 62$ kpsi. From Table 9–2, second pattern, $b = 0.375$ in, $d = 2$ in, so

$$A = 1.414hd = 1.414(0.375)2 = 1.06 \text{ in}^2$$

$$I_u = d^3/6 = 2^3/6 = 1.33 \text{ in}^3$$

$$I = 0.707hI_u = 0.707(0.375)1.33 = 0.353 \text{ in}^4$$

Primary shear:

$$\tau' = \frac{F}{A} = \frac{500(10^{-3})}{1.06} = 0.472 \text{ kpsi}$$

Secondary shear:

$$\tau'' = \frac{Mr}{I} = \frac{500(10^{-3})(6)(1)}{0.353} = 8.50 \text{ kpsi}$$
Example 9–4

The shear magnitude $\tau$ is the Pythagorean combination

$$\tau = (\tau'^2 + \tau''^2)^{1/2} = (0.472^2 + 8.50^2)^{1/2} = 8.51 \text{ kpsi}$$

The factor of safety based on a minimum strength and the distortion-energy criterion is

$$n = \frac{S_{sy}}{\tau} = \frac{0.577(50)}{8.51} = 3.39$$

Eq. 5-19

Since $n \geq n_d$, that is, $3.39 \geq 3.0$, the weld metal has satisfactory strength.

Eq. 5-21

$S_{sy} = 0.577 \times S_y$
Example 9–4

(b) From Table A–20, minimum strengths are $S_{ut} = 58$ kpsi and $S_y = 32$ kpsi. Then

\[
\sigma = \frac{M}{I/c} = \frac{M}{bd^2/6} = \frac{500(10^{-3})6}{0.375(2^2)/6} = 12 \text{ kpsi}
\]

\[
n = \frac{S_y}{\sigma} = \frac{32}{12} = 2.67
\]

Since $n < n_d$, that is, $2.67 < 3.0$, the joint is unsatisfactory as to the attachment strength.

(c) From part (a), $\tau = 8.51$ kpsi. For an \textbf{E6010 electrode} Table 9–6 gives the allowable shear stress $\tau_{all}$ as 18 kpsi. Since $\tau < \tau_{all}$, the weld is satisfactory. Since the code already has a design factor of $0.577(50)/18 = 1.6$ included at the equality, the corresponding factor of safety to part (a) is

\[
n = 1.6 \frac{18}{8.51} = 3.38
\]

which is consistent.

### Strength Level of Weld Metal (EXX)

<table>
<thead>
<tr>
<th>Strength Level of Weld Metal (EXX)</th>
</tr>
</thead>
<tbody>
<tr>
<td>60*</td>
</tr>
<tr>
<td>-----</td>
</tr>
<tr>
<td>Allowable shear stress on throat, ksi (1000 psi) of fillet or partial penetration groove weld</td>
</tr>
<tr>
<td>18.0</td>
</tr>
</tbody>
</table>
Fatigue Stress-Concentration Factors

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

<table>
<thead>
<tr>
<th>Table 9-5</th>
<th>Type of Weld</th>
<th>$K_{fs}$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Fatigue Stress-Concentration Factors, $K_{fs}$</td>
<td>Reinforced butt weld</td>
<td>1.2</td>
</tr>
<tr>
<td></td>
<td>Toe of transverse fillet weld</td>
<td>1.5</td>
</tr>
<tr>
<td></td>
<td>End of parallel fillet weld</td>
<td>2.7</td>
</tr>
<tr>
<td></td>
<td>T-butt joint with sharp corners</td>
<td>2.0</td>
</tr>
</tbody>
</table>

🌟 For Welding codes, see the fatigue stress allowable in the AISI manual.
Example 9–5

The 1018 steel strap of Fig. 9–21 has a 1000 lbf, completely reversed load applied. Determine the factor of safety of the weldment for infinite life.

Fig. 9–21
Example 9–5

From Table A–20 for the 1018 attachment metal the strengths are $S_{ut} = 58$ kpsi and $S_y = 32$ kpsi. For the E6010 electrode, from Table 9–3 $S_{ut} = 62$ kpsi and $S_y = 50$ kpsi. The fatigue stress-concentration factor, from Table 9–5, is $K_{fs} = 2.7$. From Table 6–2, p. 288, $k_a = 39.9(58)^{-0.995} = 0.702$. For case 2 of Table 9–5, the shear area is:

$$k_a = a S_{ut}^b$$

$$A = 1.414(0.375)(2) = 1.061 \text{ in}^2$$

For a uniform shear stress on the throat, $k_b = 1$.

From Eq. (6–26), p. 290, for torsion (shear),

$$k_c = 0.59 \quad k_d = k_e = k_f = 1$$

From Eqs. (6–8), p. 282, and (6–18), p. 287,

$$S_{se} = 0.702(1)(0.59)(1)(1)(0.5)(58) = 12.0 \text{ kpsi}$$

$$S_e = k_a k_b k_c k_d k_e k_f S_e'$$

$$k_c = \begin{cases} 
1 & \text{bending} \\
0.85 & \text{axial} \\
0.59 & \text{torsion}
\end{cases}$$
Example 9–5

From Table 9–5, $K_{fs} = 2.7$. Only primary shear is present. So, with $F_a = 1000$ lbf and $F_m = 0$

$$\tau'_a = \frac{K_{fs}F_a}{A} = \frac{2.7(1000)}{1.061} = 2545 \text{ psi} \quad \tau'_m = 0 \text{ psi}$$

In the absence of a midrange component, the fatigue factor of safety $n_f$ is given by

$$n_f = \frac{S_{se}}{\tau'_a} = \frac{12000}{2545} = 4.72$$
Example 9–6

The 1018 steel strap of Fig. 9–22 has a repeatedly applied load of 2000 lbf \((F_a = F_m = 1000 \text{ lbf})\). Determine the fatigue factor of safety fatique strength of the weldment.

Fig. 9–22
Example 9–6

From Table 6–2, p. 288, \( k_a = 39.9(58)^{-0.995} = 0.702 \). From case 2 of Table 9–2

\[
A = 1.414(0.375)(2) = 1.061 \text{ in}^2
\]

For uniform shear stress on the throat \( k_b = 1 \).

From Eq. (6–26), p. 290, \( k_c = 0.59 \). From Eqs. (6–8), p. 282, and (6–18), p. 287,

\[
S_{se} = 0.702(1)0.59(1)(1)(1)0.5(58) = 12.0 \text{ kpsi}
\]

From Table 9–5, \( K_{fs} = 2 \). Only primary shear is present:

\[
\tau'_a = \tau'_m = \frac{K_{fs}F_a}{A} = \frac{2(1000)}{1.061} = 1885 \text{ psi}
\]
Example 9–6

From Eq. (6–54), p. 317, \( S_{su} \leq 0.67S_{ut} \). This, together with the Gerber fatigue failure criterion for shear stresses from Table 6–7, p. 307, gives

\[
n_f = \frac{1}{2} \left( \frac{0.67S_{ut}}{\tau_m} \right)^2 \frac{\tau_a}{S_{se}} \left[ -1 + \sqrt{1 + \left( \frac{2\tau_m S_{se}}{0.67S_{ut} \tau_a} \right)^2} \right]
\]

\[
n_f = \frac{1}{2} \left[ \frac{0.67(58)}{1.885} \right]^2 \frac{1.885}{12.0} \left\{ -1 + \sqrt{1 + \left[ \frac{2(1.885)12.0}{0.67(58)1.885} \right]^2} \right\} = 5.85
\]

Fatigue factor of safety

\[
n_f = \frac{1}{2} \left( \frac{S_{ut}}{\sigma_m} \right)^2 \frac{\sigma_a}{S_e} \left[ -1 + \sqrt{1 + \left( \frac{2\sigma_m S_e}{S_{ut} \sigma_a} \right)^2} \right] \quad \sigma_m > 0
\]
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](image-url)
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

Fig. 9–26

---

*Fig. 9–26* (b)
Double-lap Joint

- Shear-stress distribution is given by

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

where

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

Fig. 9–26b
Example 9–7

The double-lap joint depicted in Fig. 9–26 consists of aluminum outer adherends and an inner steel adherend. The assembly is cured at 250°F and is stress-free at 200°F. The completed bond is subjected to an axial load of 2000 lbf at a service temperature of 70°F. The width \( b \) is 1 in, the length of the bond \( l \) is 1 in. Additional information is tabulated below:

<table>
<thead>
<tr>
<th></th>
<th>( G, \text{ psi} )</th>
<th>( E, \text{ psi} )</th>
<th>( \alpha, \text{ in/(in \cdot °F)} )</th>
<th>Thickness, in</th>
</tr>
</thead>
<tbody>
<tr>
<td>Adhesive</td>
<td>0.2 ((10^6))</td>
<td></td>
<td>55 ((10^{-6}))</td>
<td>0.020</td>
</tr>
<tr>
<td>Outer adherend</td>
<td>10 ((10^6))</td>
<td>13.3 ((10^{-6}))</td>
<td></td>
<td>0.150</td>
</tr>
<tr>
<td>Inner adherend</td>
<td>30 ((10^6))</td>
<td>6.0 ((10^{-6}))</td>
<td></td>
<td>0.100</td>
</tr>
</tbody>
</table>

Sketch a plot of the shear stress as a function of the length of the bond due to (a) thermal stress, (b) load-induced stress, and (c) the sum of stresses in \( a \) and \( b \); and (d) find where the largest shear stress is maximum.

Fig. 9–26
Example 9–7

In Eq. (9–7) the parameter $\omega$ is given by

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

\[
= \sqrt{\frac{0.2(10^6)}{0.020} \left[ \frac{1}{10(10^6)0.15} + \frac{2}{30(10^6)0.10} \right]} = 3.65 \text{ in}^{-1}
\]

(a) For the thermal component, $\alpha_i - \alpha_o = 6(10^{-6}) - 13.3(10^{-6}) = -7.3(10^{-6})$ in/(in $\cdot$ °F), $\Delta T = 70 - 200 = -130^\circ F$,

\[
\tau_{th}(x) = \frac{(\alpha_i - \alpha_o) \Delta T \omega \sinh(\omega x)}{(1/E_o t_o + 2/E_i t_i) \cosh(\omega l/2)}
\]

\[
\tau_{th}(x) = \frac{-7.3(10^{-6})(-130)3.65 \sinh(3.65x)}{\left[ \frac{1}{10(10^6)0.150} + \frac{2}{30(10^6)0.100} \right] \cosh \left[ \frac{3.65(1)}{2} \right]}
\]

\[
= 816.4 \sinh(3.65x)
\]
Example 9–7

The thermal stress is plotted in Fig. (9–27) and tabulated at \( x = -0.5, 0, \) and 0.5 in the table below.
Example 9-7

(b) The bond is “balanced” \((E_o t_o = E_i t_i / 2)\), so the load-induced stress is given by

\[
\tau_P(x) = \frac{P \omega \cosh(\omega x)}{4b \sinh(\omega l / 2)} = \frac{2000(3.65) \cosh(3.65x)}{4(1)3.0208} = 604.1 \cosh(3.65x)
\]  

The load-induced stress is plotted in Fig. (9–27) and tabulated at \(x = -0.5\), 0, and 0.5 in the table below.

(c) Total stress table (in psi):

<table>
<thead>
<tr>
<th></th>
<th>(\tau(-0.5))</th>
<th>(\tau(0))</th>
<th>(\tau(0.5))</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal only</td>
<td>-2466</td>
<td>0</td>
<td>2466</td>
</tr>
<tr>
<td>Load-induced only</td>
<td>1922</td>
<td>604</td>
<td>1922</td>
</tr>
<tr>
<td>Combined</td>
<td>-544</td>
<td>604</td>
<td>4388</td>
</tr>
</tbody>
</table>
Example 9-7

(d) The maximum shear stress predicted by the shear-lag model will always occur at the ends. See the plot in Fig. 9–27. Since the residual stresses are always present, significant shear stresses may already exist prior to application of the load. The large stresses present for the combined-load case could result in local yielding of a ductile adhesive or failure of a more brittle one. The significance of the thermal stresses serves as a caution against joining dissimilar adherends when large temperature changes are involved. Note also that the average shear stress due to the load is \( \tau_{avg} = P/(2bl) = 1000 \) psi. Equation (1) produced a maximum of 1922 psi, almost double the average.
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

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

Fig. 9–28

Stresses shown for an applied load of
\( P = 1000 \text{ lbf (4.4 kN)} \)

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

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