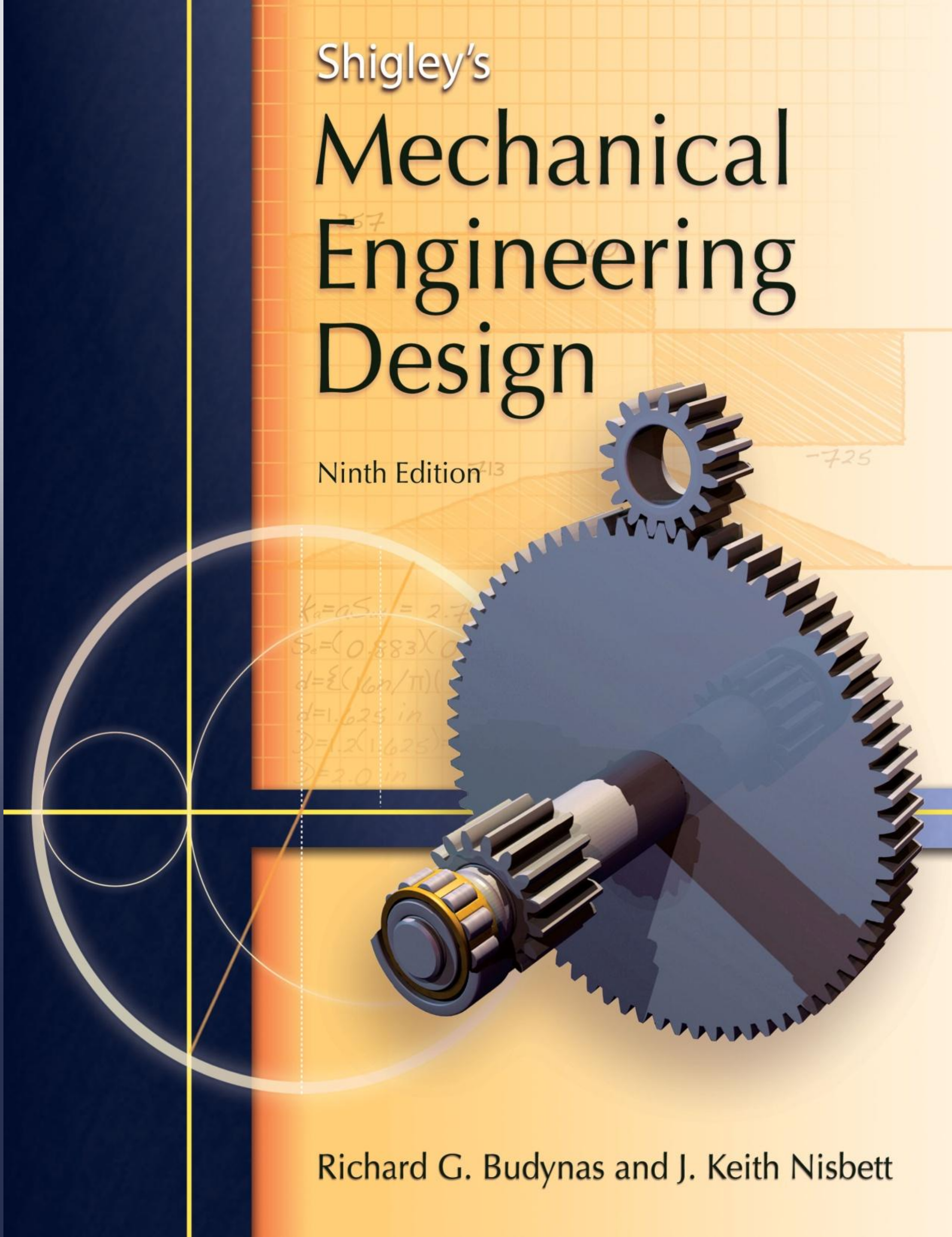
Lecture Slides

Chapter 9

Welding, Bonding, and the Design of Permanent Joints



Chapter Outline

Welding Symbols

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

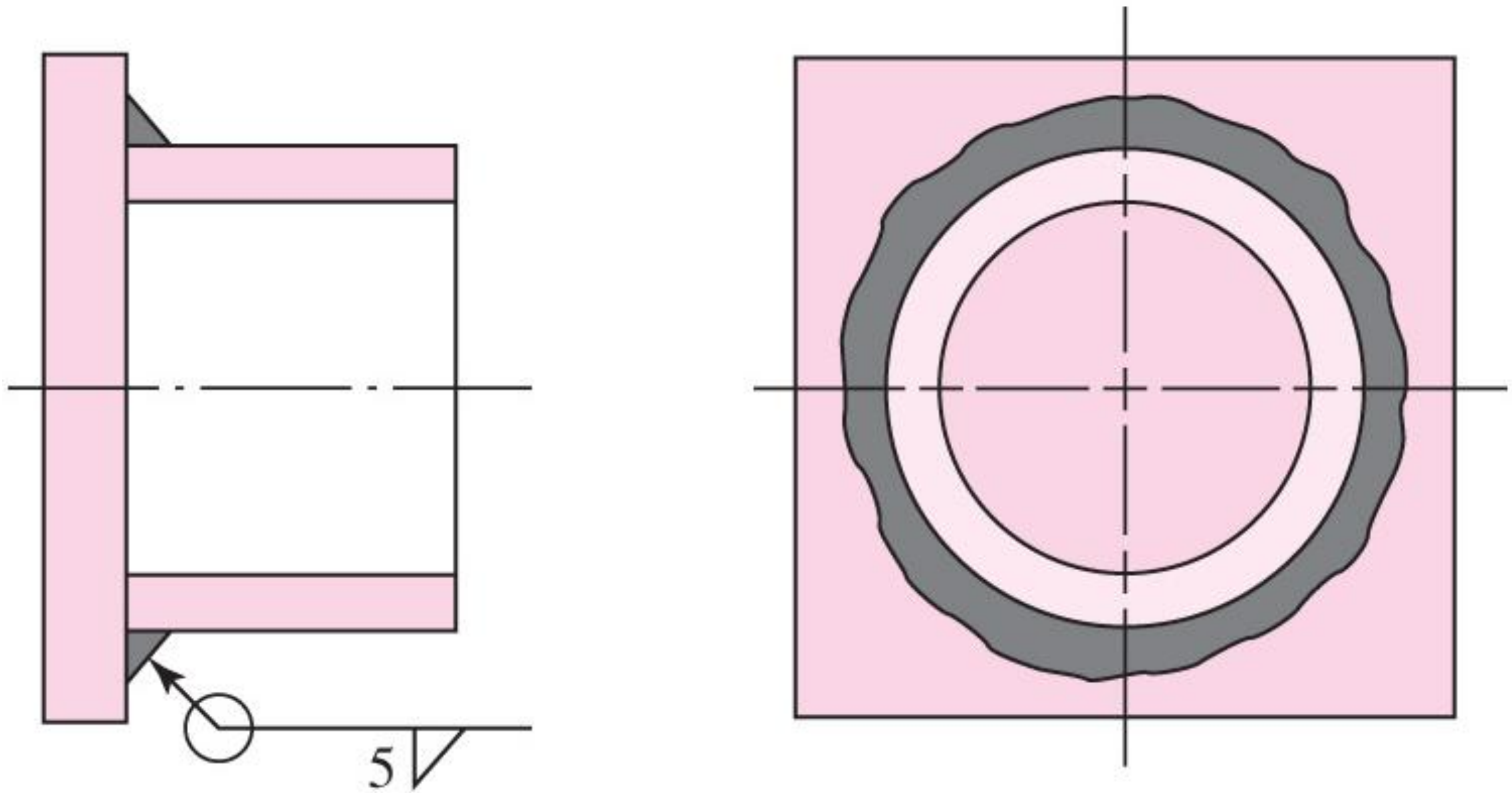


Fig. 9–4

Welding Symbols

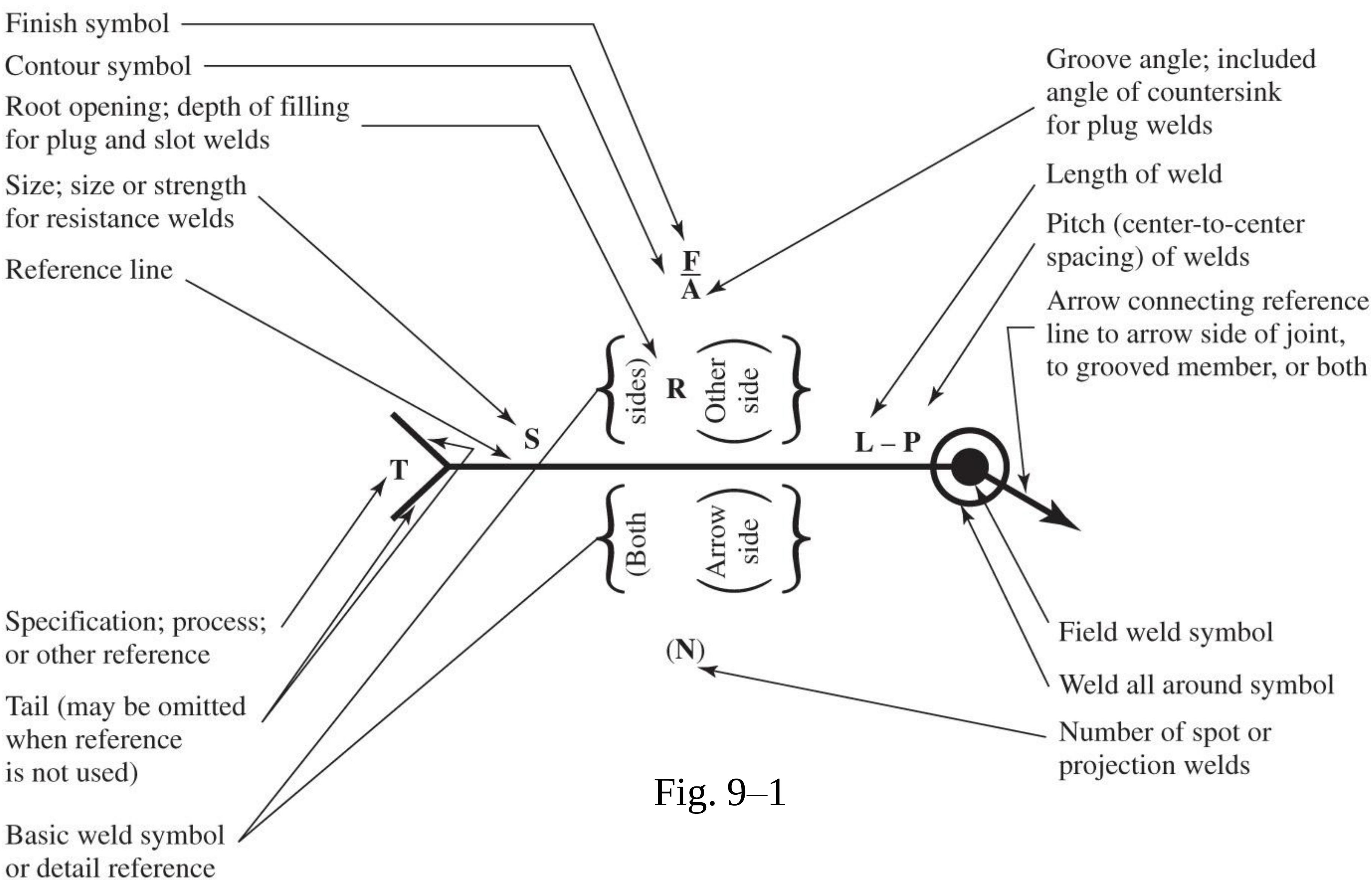


Fig. 9–1

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

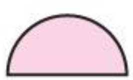
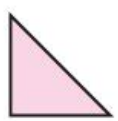
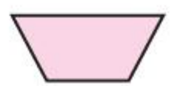
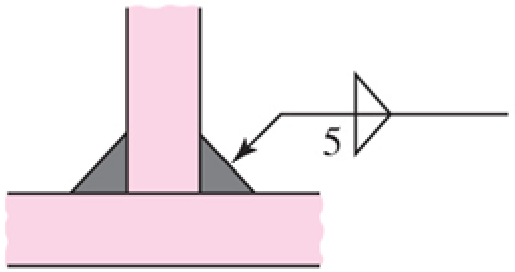
Type of weld							
Bead	Fillet	Plug or slot	Groove				
			Square	V	Bevel	U	J
							

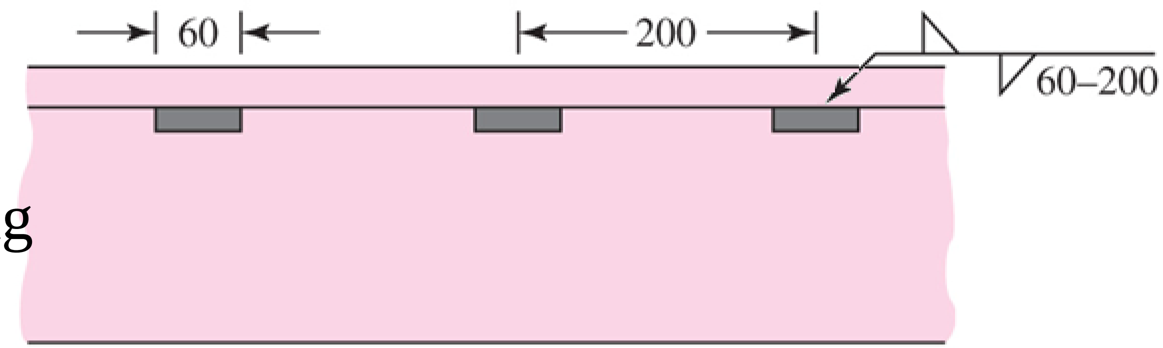
Fig. 9–2

Welding Symbol Examples

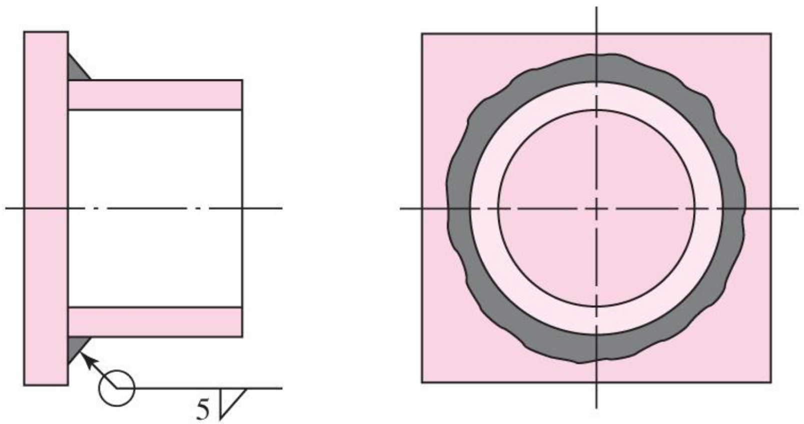
- Weld leg size of 5 mm
- Fillet weld
- Both sides



- 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

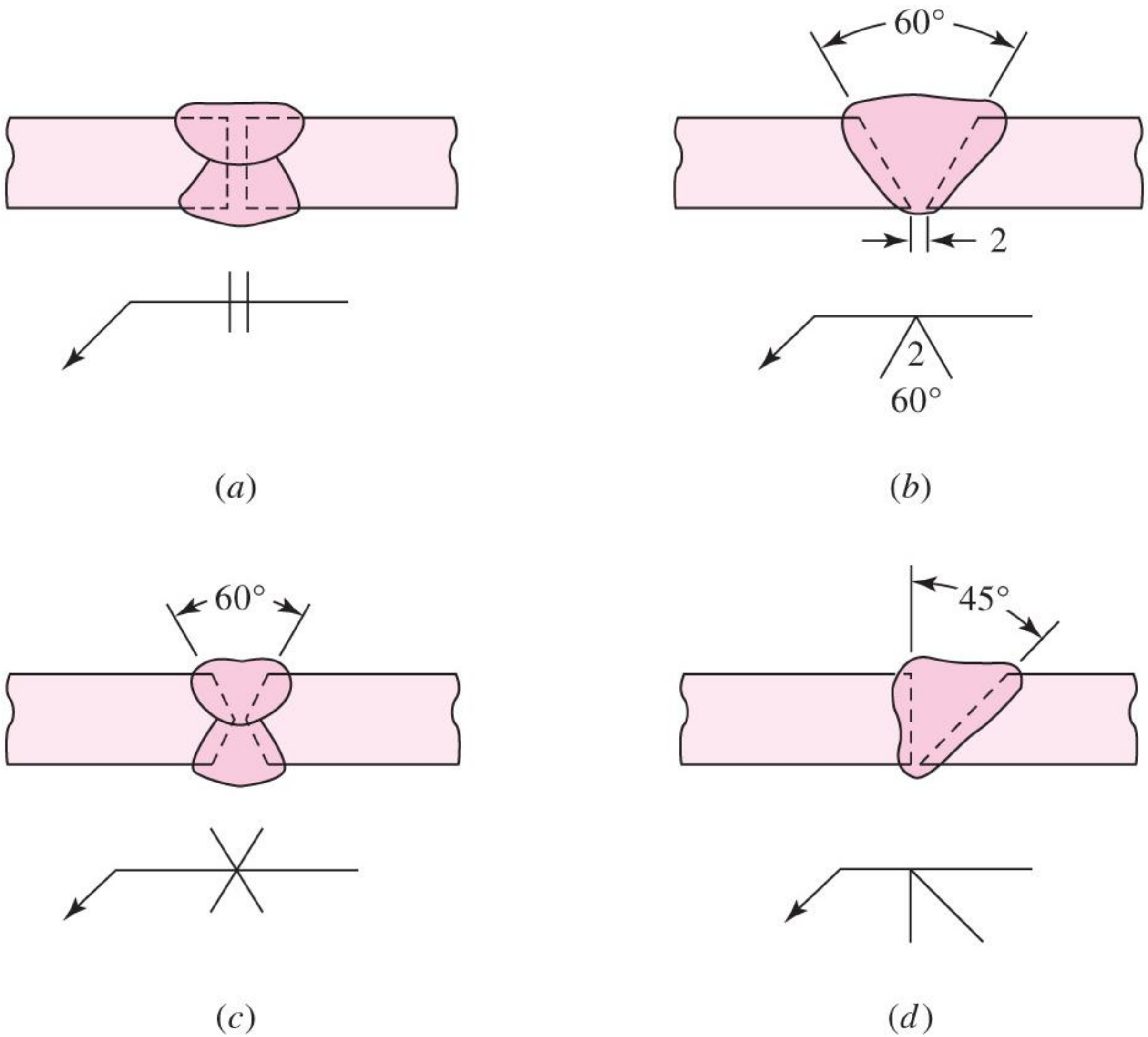


Fig. 9–5

Welding Symbol Examples

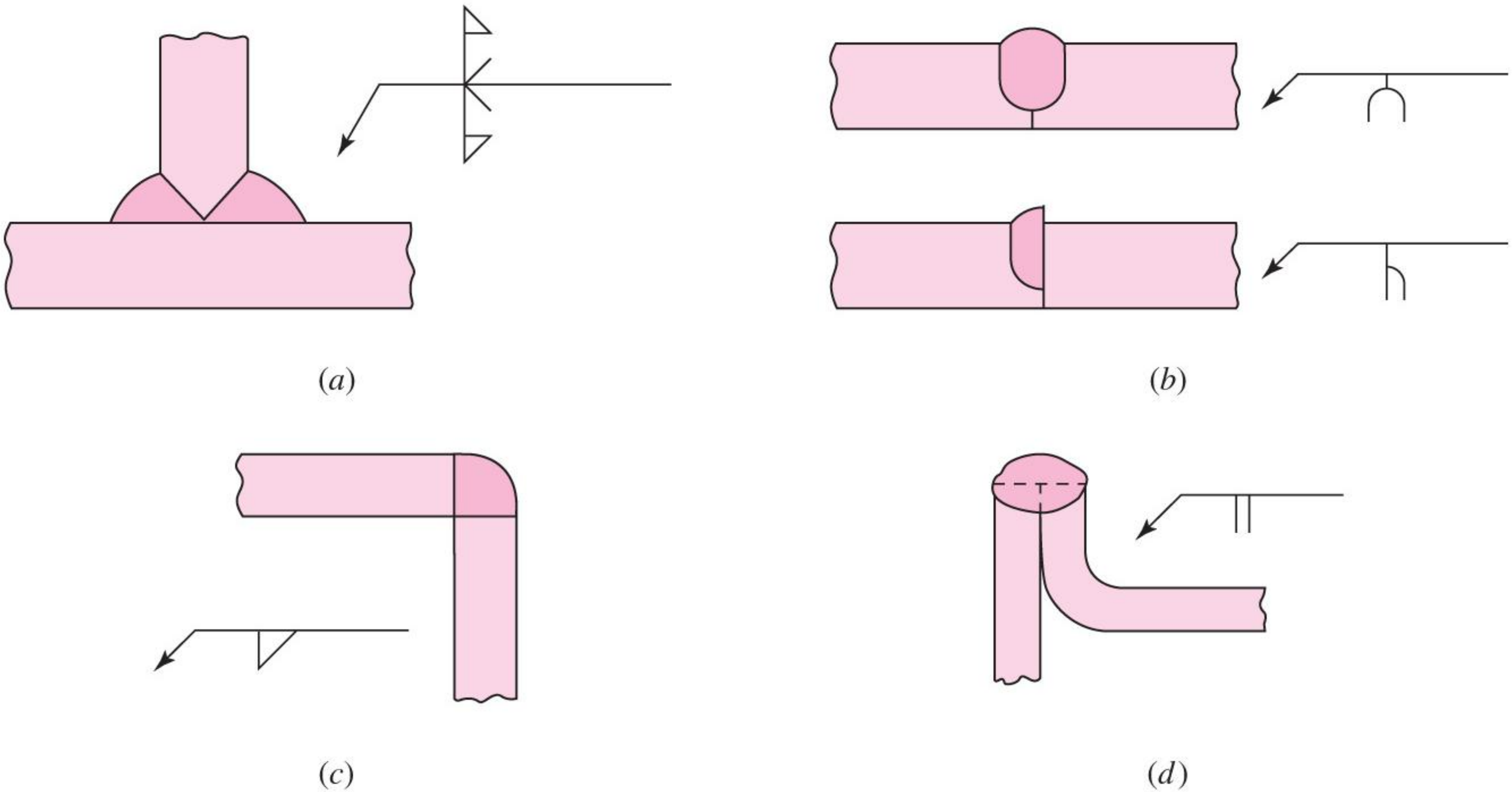


Fig. 9–6

Tensile Butt Joint

- Simple butt joint loaded in tension or compression
- Stress is normal stress

$$\sigma = \frac{F}{hl} \qquad (9\text{–}1)$$

- 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

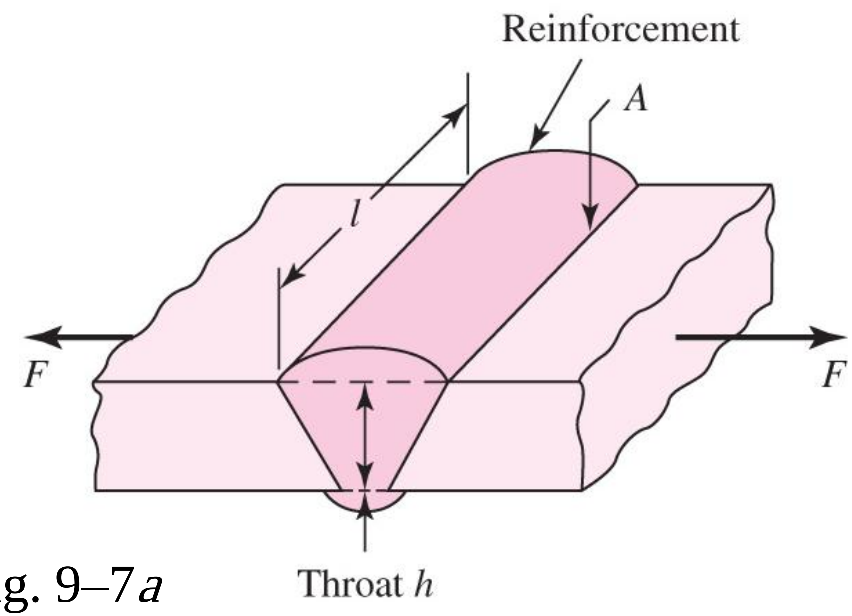


Fig. 9–7*a*

Shear Butt Joint

- Simple butt joint loaded in shear
- Average shear stress

$$\tau = \frac{F}{hl} \qquad (9\text{–}2)$$

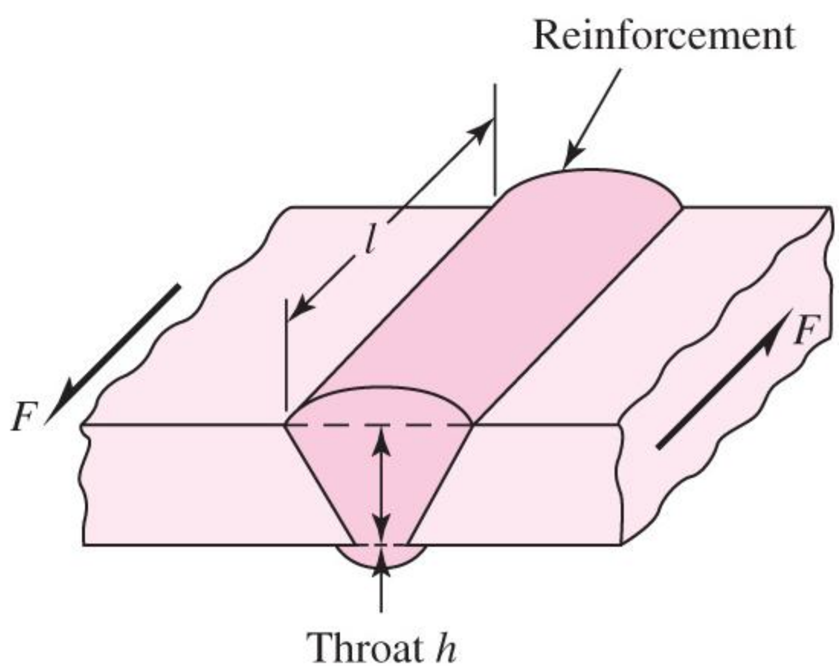


Fig. 9–7*b*

Transverse Fillet Weld

- Joint loaded in tension
- Weld loading is complex

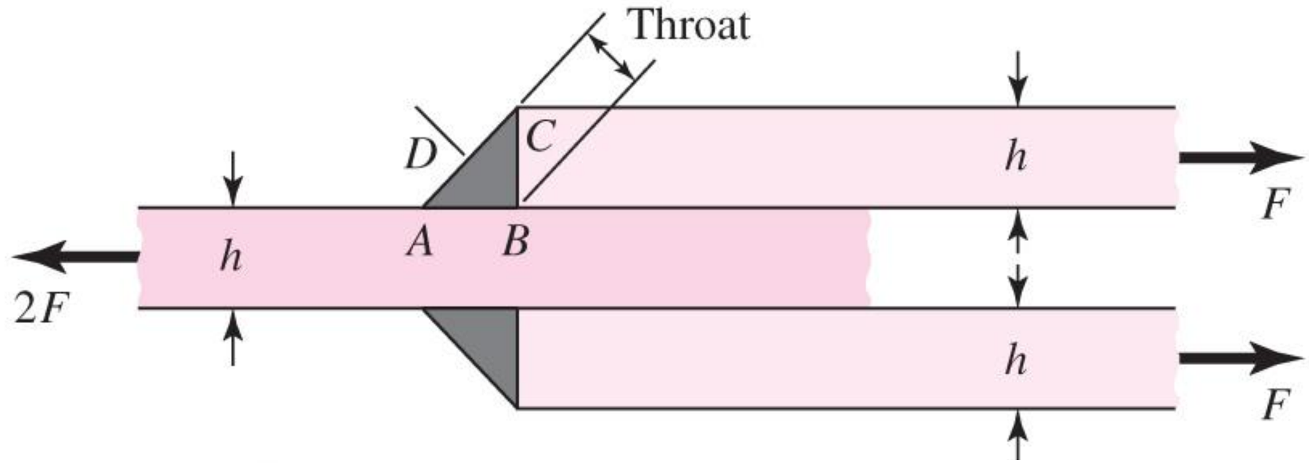


Fig. 9–8

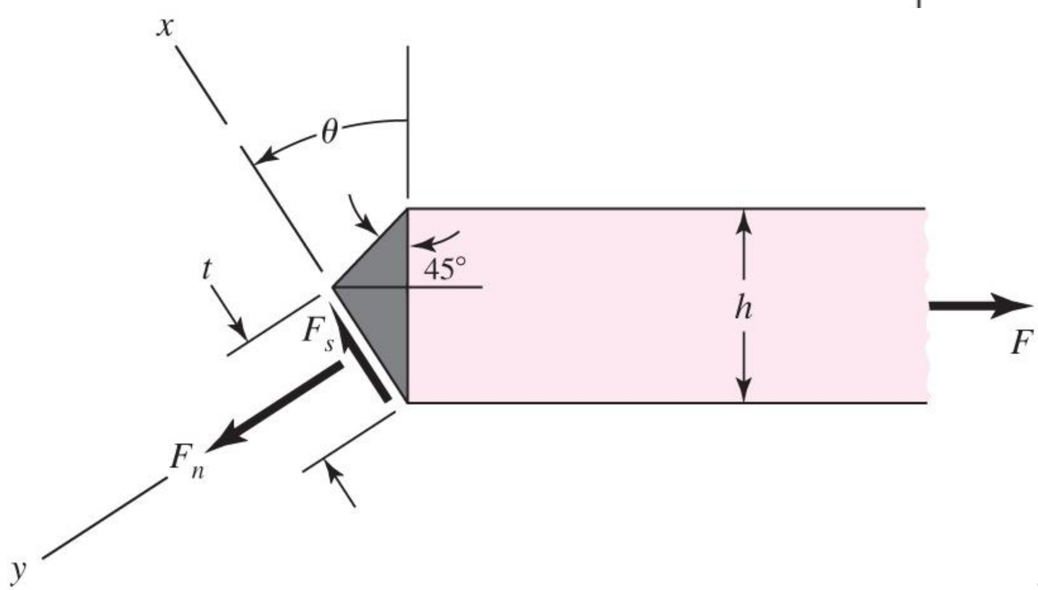


Fig. 9–9

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

$$t = \frac{h}{\cos\theta + \sin\theta}$$

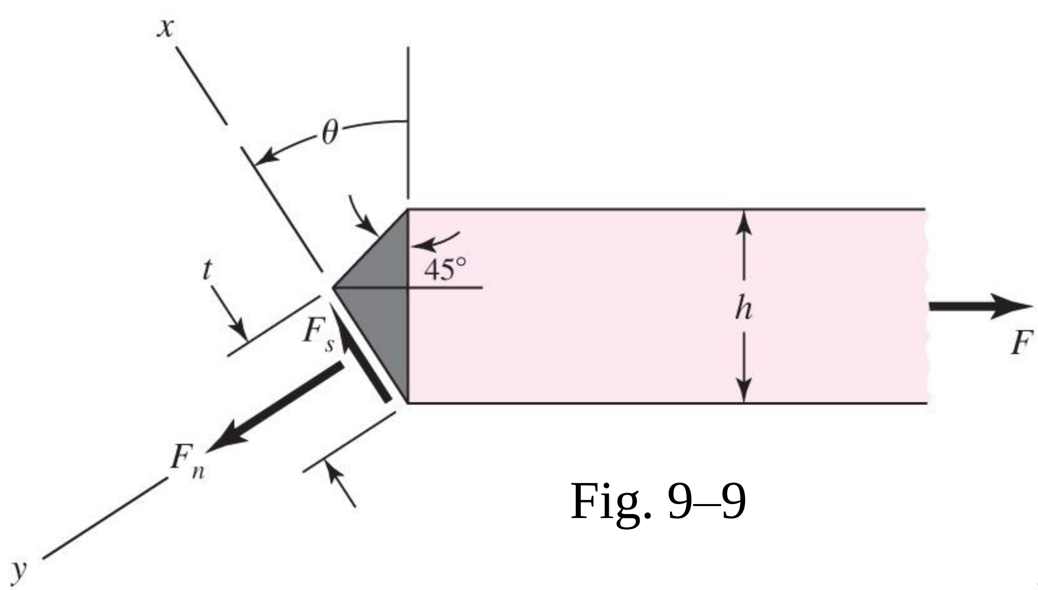


Fig. 9–9

Transverse Fillet Weld

- Nominal stresses at angle θ

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

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

- Von Mises Stress at angle θ

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

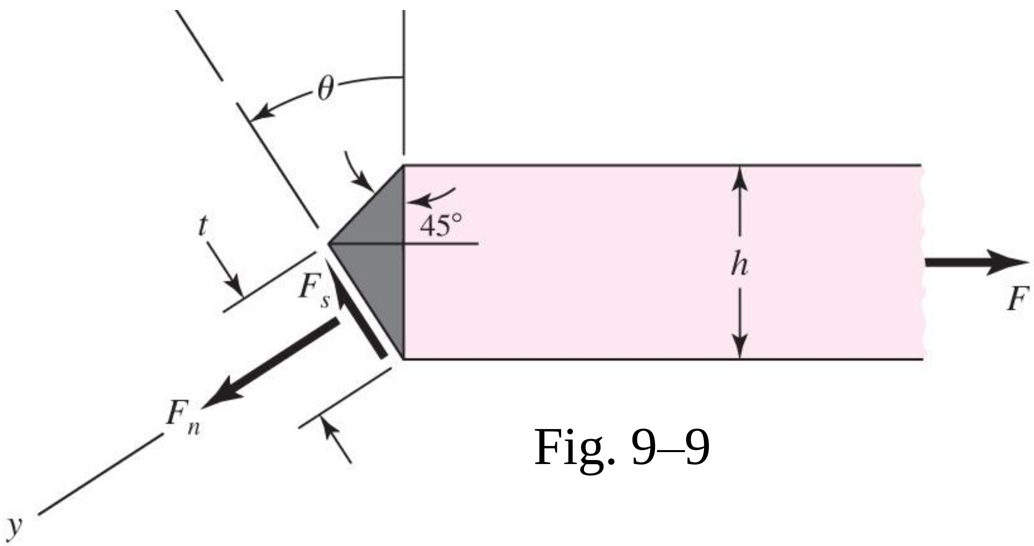


Fig. 9–9

Transverse Fillet Weld

- Largest von Mises stress occurs at $\theta = 62.5°$ with value of $\sigma' = 2.16F/(hl)$
- Maximum shear stress occurs at $\theta = 67.5°$ with value of $\tau_{max} = 1.207F/(hl)$

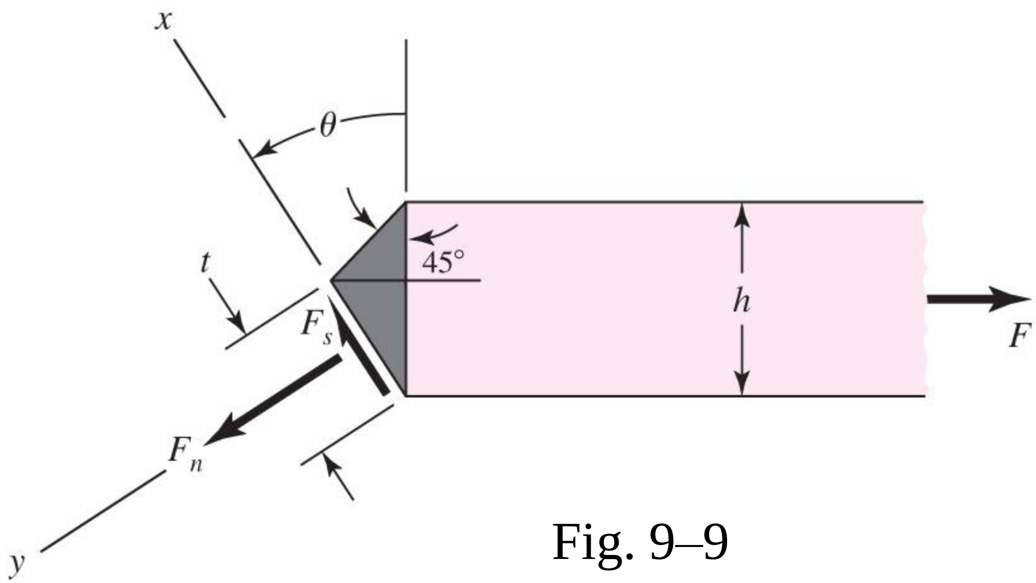


Fig. 9–9

Experimental Stresses in Transverse Fillet Weld

- Experimental results are more complex

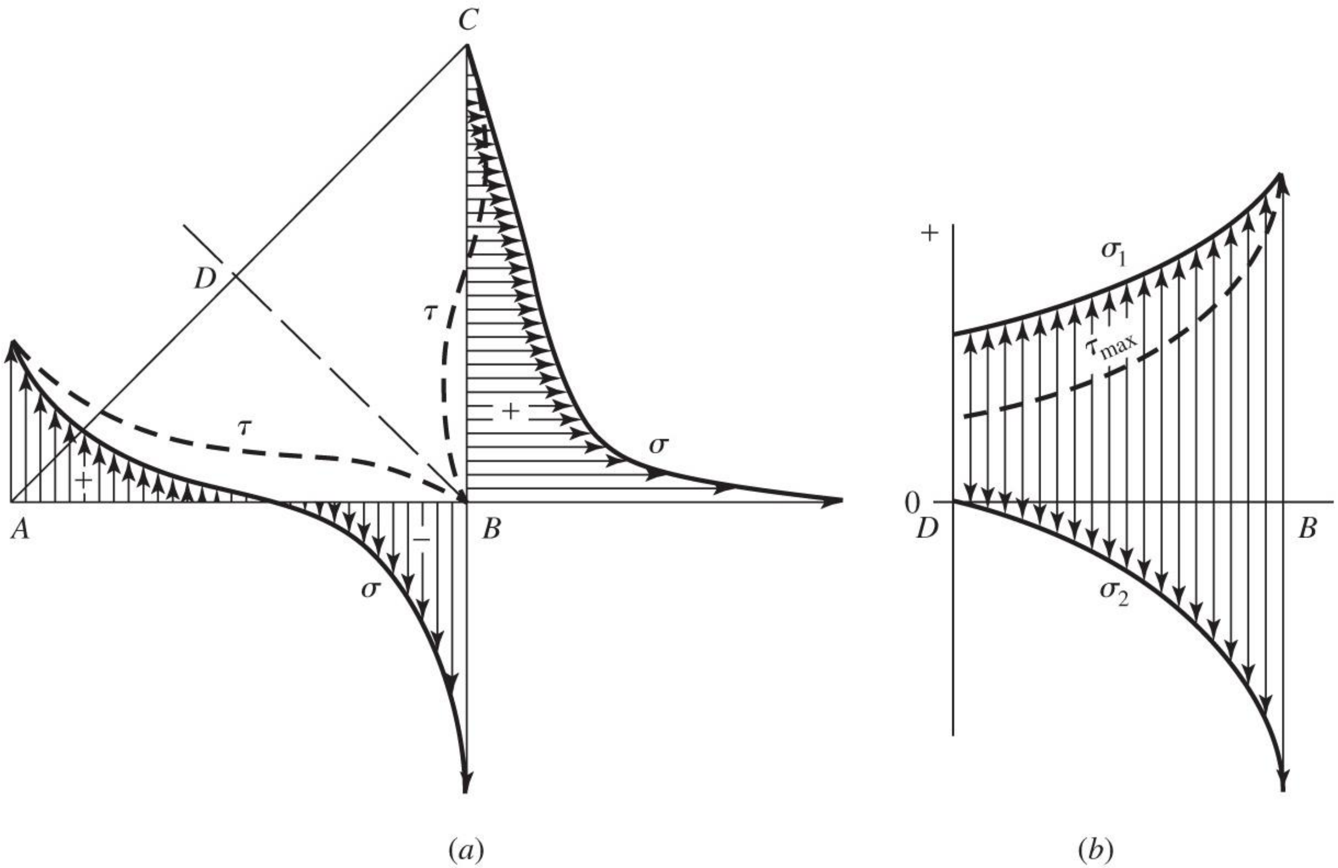


Fig. 9–10

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.

$$\tau = \frac{F}{0.707hl} = \frac{1.414F}{hl} \qquad (9\text{–}3)$$

- 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 1.414/1.207 = 1.17.

Parallel Fillet Welds

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

$$\tau = \frac{F}{0.707hl} = \frac{1.414F}{hl} \qquad (9\text{–}3)$$

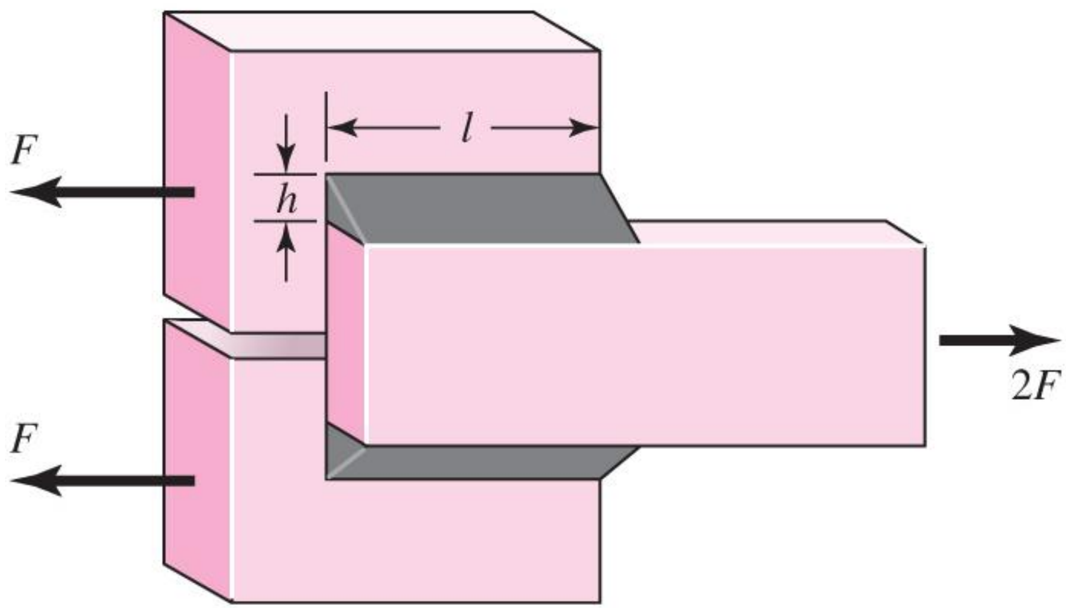


Fig. 9–11

Fillet Welds Loaded in Torsion

- Fillet welds carrying both direct shear *V* and moment *M*
- *P*... *hear*

$$\tau' = \frac{V}{A}$$

- *Se*... *hear*

$$\tau'' = \frac{Mr}{J}$$

- *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

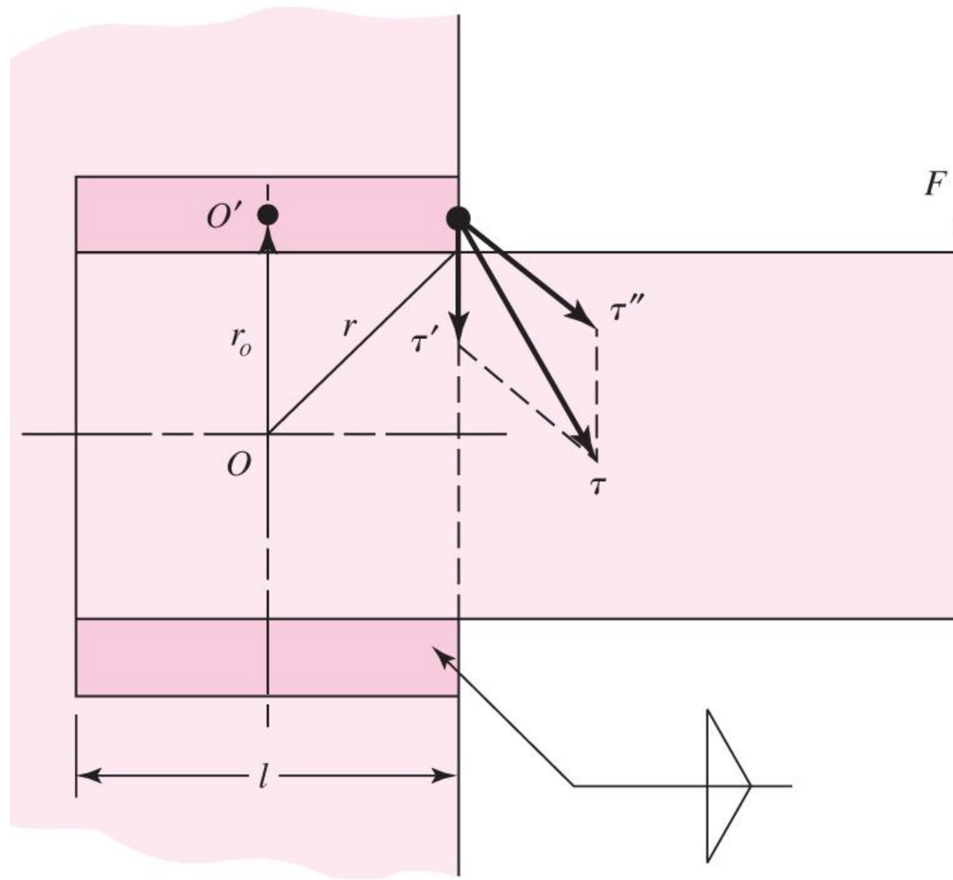


Fig. 9–12

Example of Finding *A* and *J*

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

$$A = A_1 + A_2 = t_1 d + t_2 b$$

$$I_x = \frac{t_1 d^3}{12} \qquad I_y = \frac{d t_1^3}{12}$$

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

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

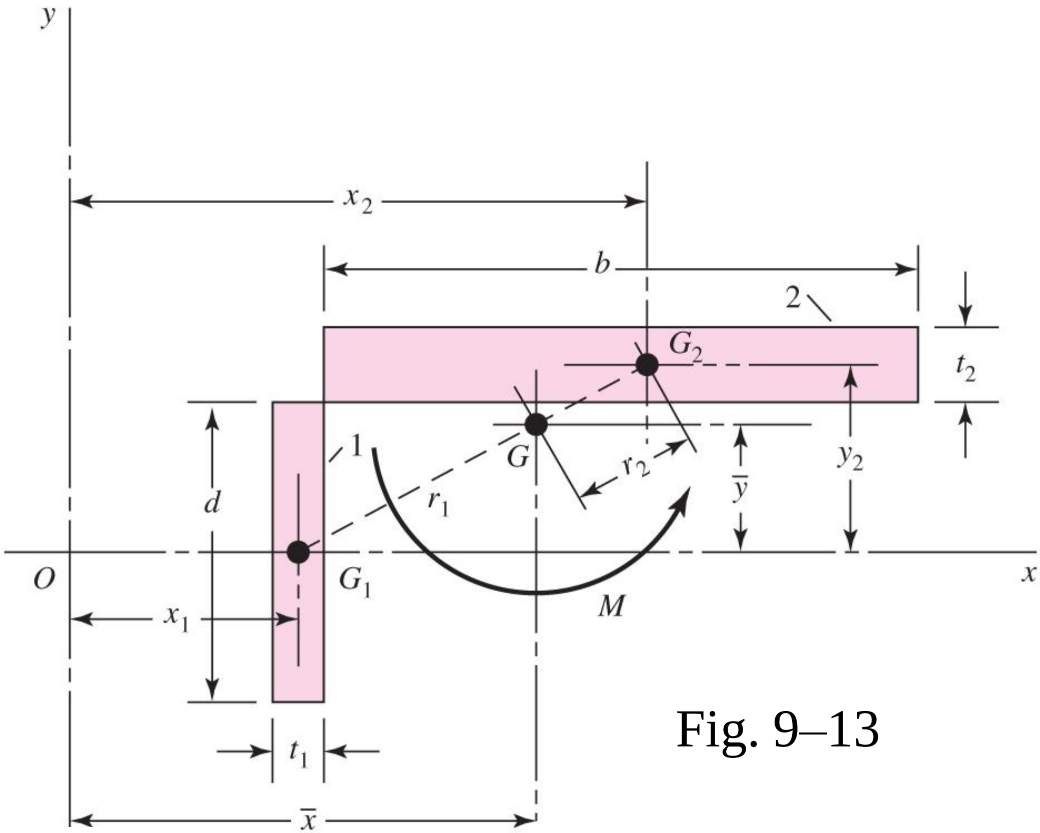


Fig. 9–13

$$\bar{x} = \frac{A_1 x_1 + A_2 x_2}{A} \qquad \bar{y} = \frac{A_1 y_1 + A_2 y_2}{A}$$

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

$$J = \left(J_{G1} + A_1 r_1^2\right) + \left(J_{G2} + A_2 r_2^2\right)$$

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 $t = 0.707h$,

$$J = 0.707hJ_u$$

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

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

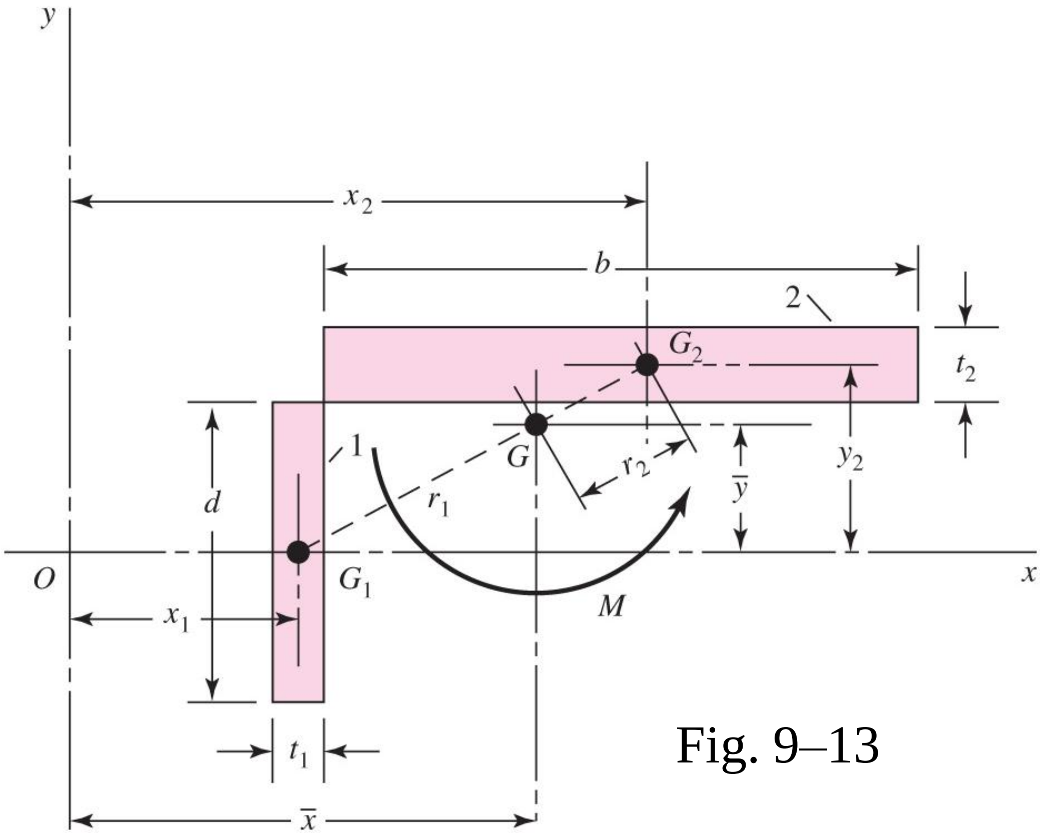
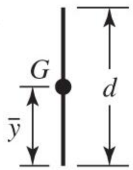
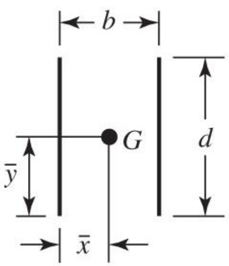
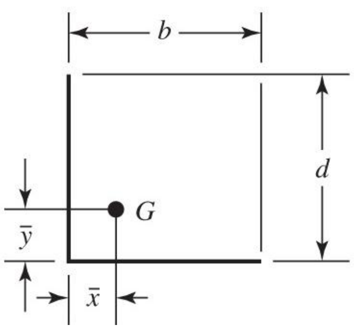
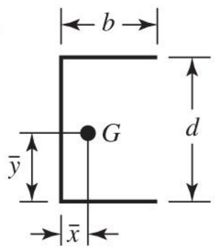
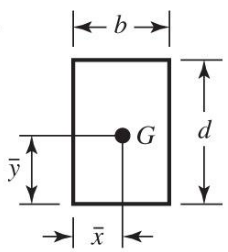
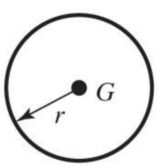


Fig. 9–13

Common Torsional Properties of Fillet Welds (Table 9–1)

Weld	Throat Area	Location of *G*	Unit Second Polar Moment of Area
1. 	$A = 0.707\,hd$	$\bar{x} = 0$ $\bar{y} = d/2$	$J_u = d^3/12$
2. 	$A = 1.414\,hd$	$\bar{x} = b/2$ $\bar{y} = d/2$	$J_u = \dfrac{d(3b^2 + d^2)}{6}$
3. 	$A = 0.707h(b + d)$	$\bar{x} = \dfrac{b^2}{2(b+d)}$ $\bar{y} = \dfrac{d^2}{2(b+d)}$	$J_u = \dfrac{(b+d)^4 - 6b^2d^2}{12(b+d)}$

Common Torsional Properties of Fillet Welds (Table 9–1)

4.		$A = 0.707h(2b + d)$	$\bar{x} = \dfrac{b^2}{2b + d}$ $\bar{y} = d/2$	$J_u = \dfrac{8b^3 + 6bd^2 + d^3}{12} - \dfrac{b^4}{2b + d}$
5.		$A = 1.414h(b + d)$	$\bar{x} = b/2$ $\bar{y} = d/2$	$J_u = \dfrac{(b + d)^3}{6}$
6.		$A = 1.414\,\pi hr$		$J_u = 2\pi r^3$

*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.

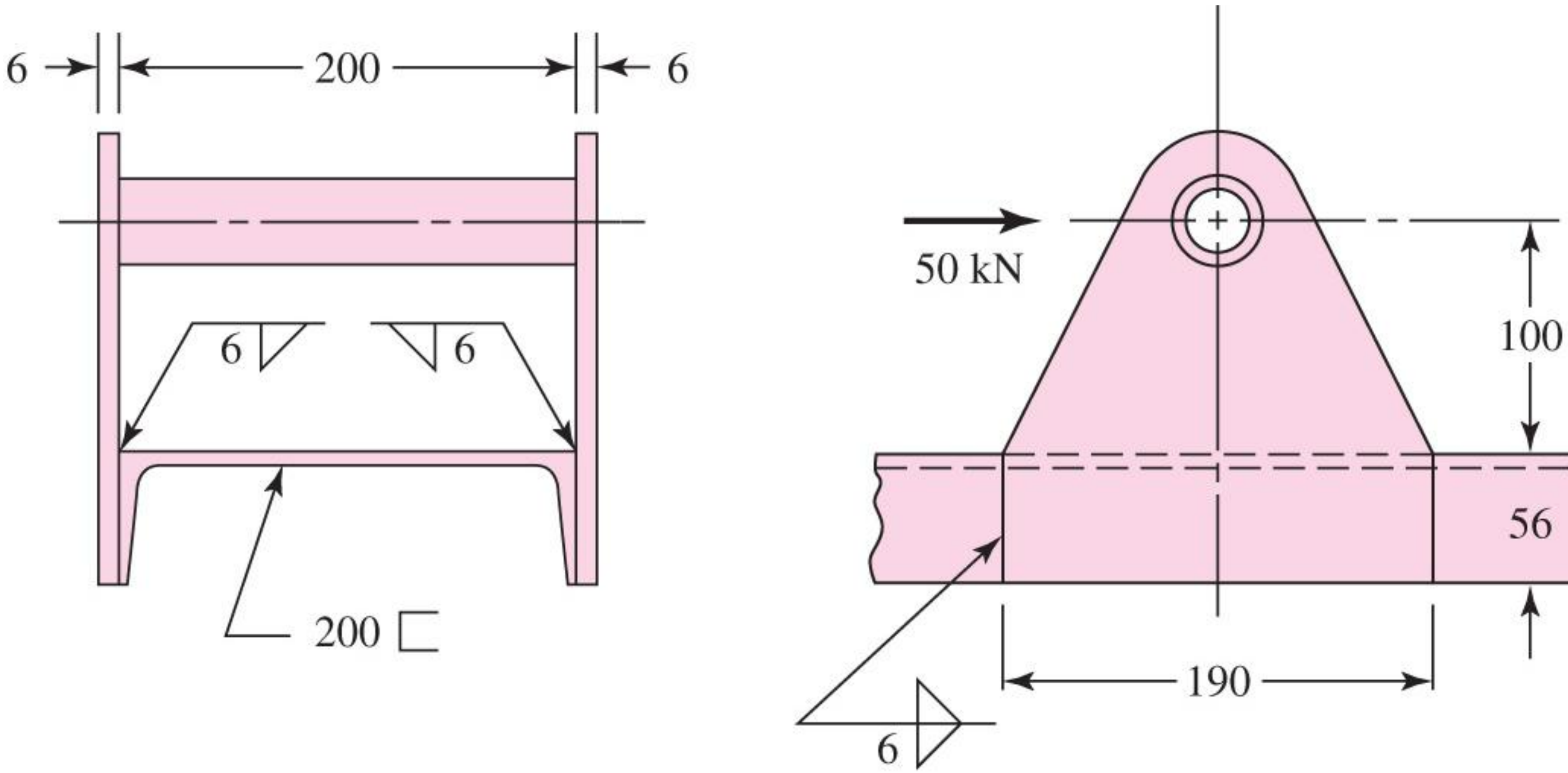


Fig. 9–14

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.

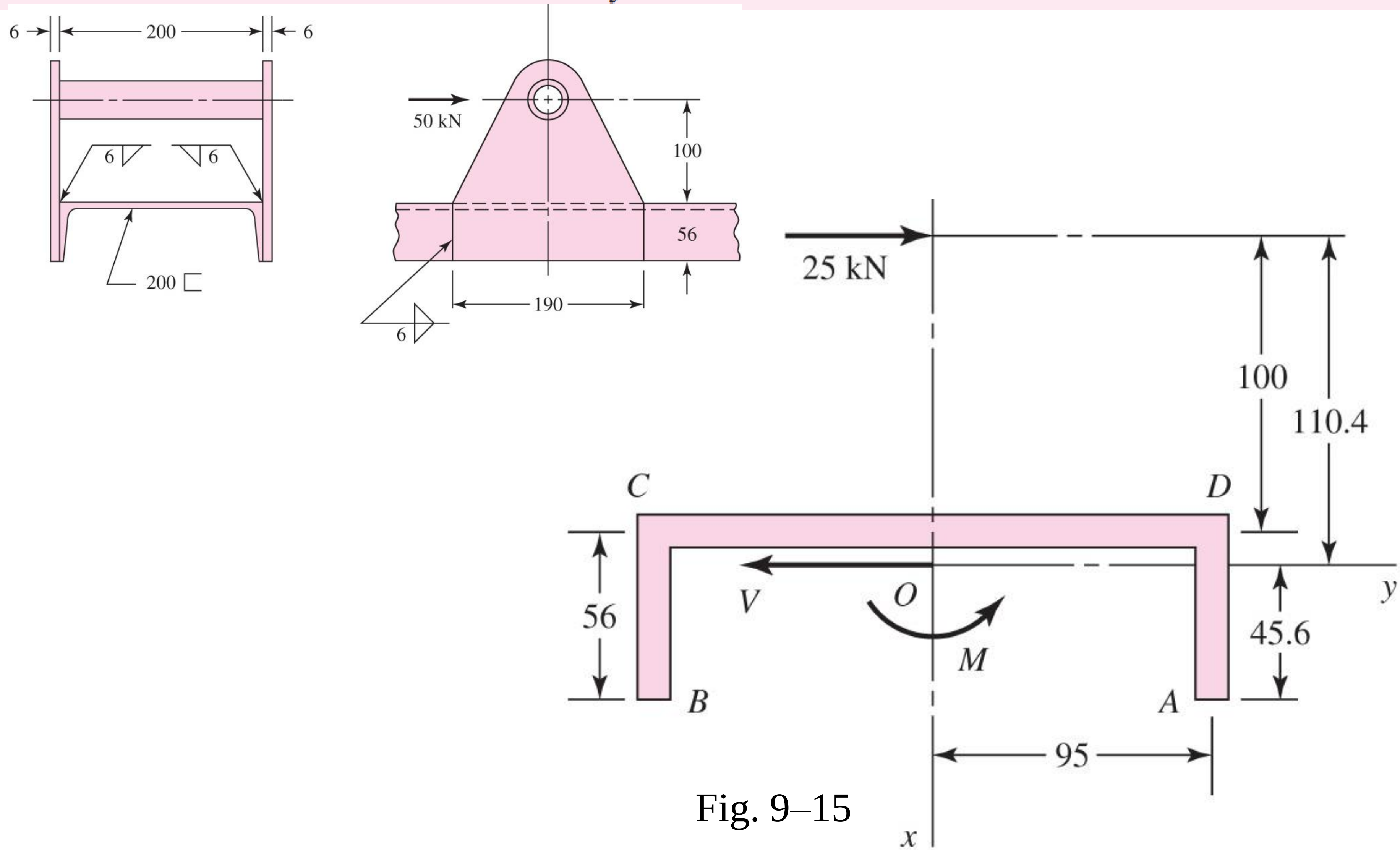


Fig. 9–15

Example 9–1

(*b*) Estimate the primary shear stress τ'. 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

$$\tau' = \frac{V}{A} = \frac{25(10)^3}{1280} = 19.5 \text{ MPa}$$

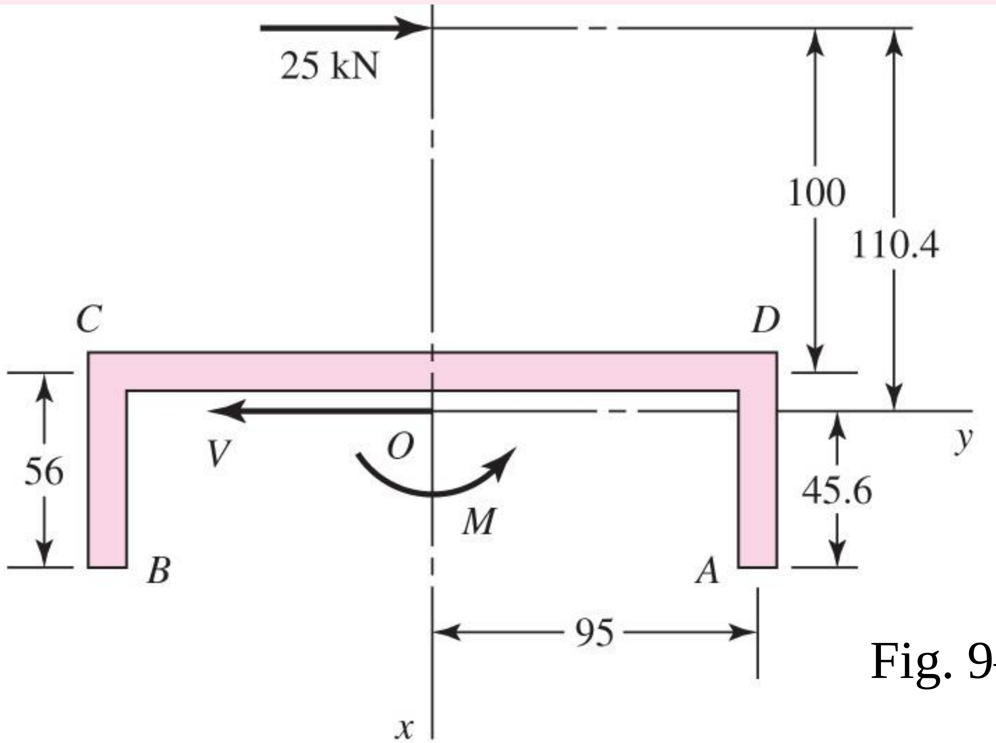


Fig. 9–15

Example 9–1

(*c*) Draw the τ' 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):

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

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

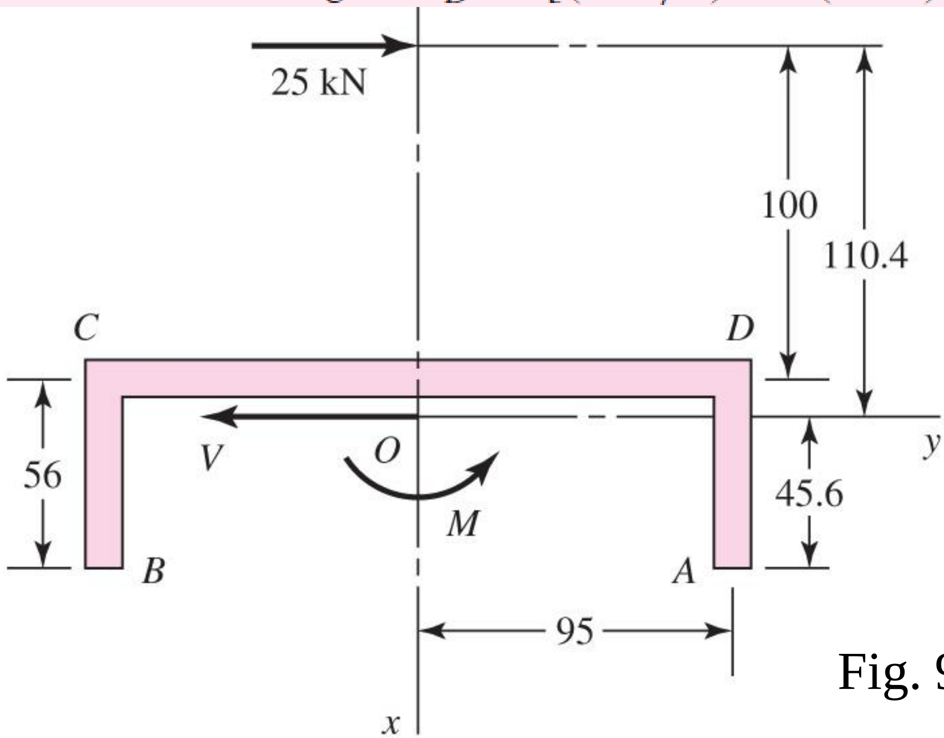


Fig. 9–15

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 τ'' 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 τ'' 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 τ' and τ'' stresses represent what the channel is doing to the plate (through the welds) to hold the plate in equilibrium.

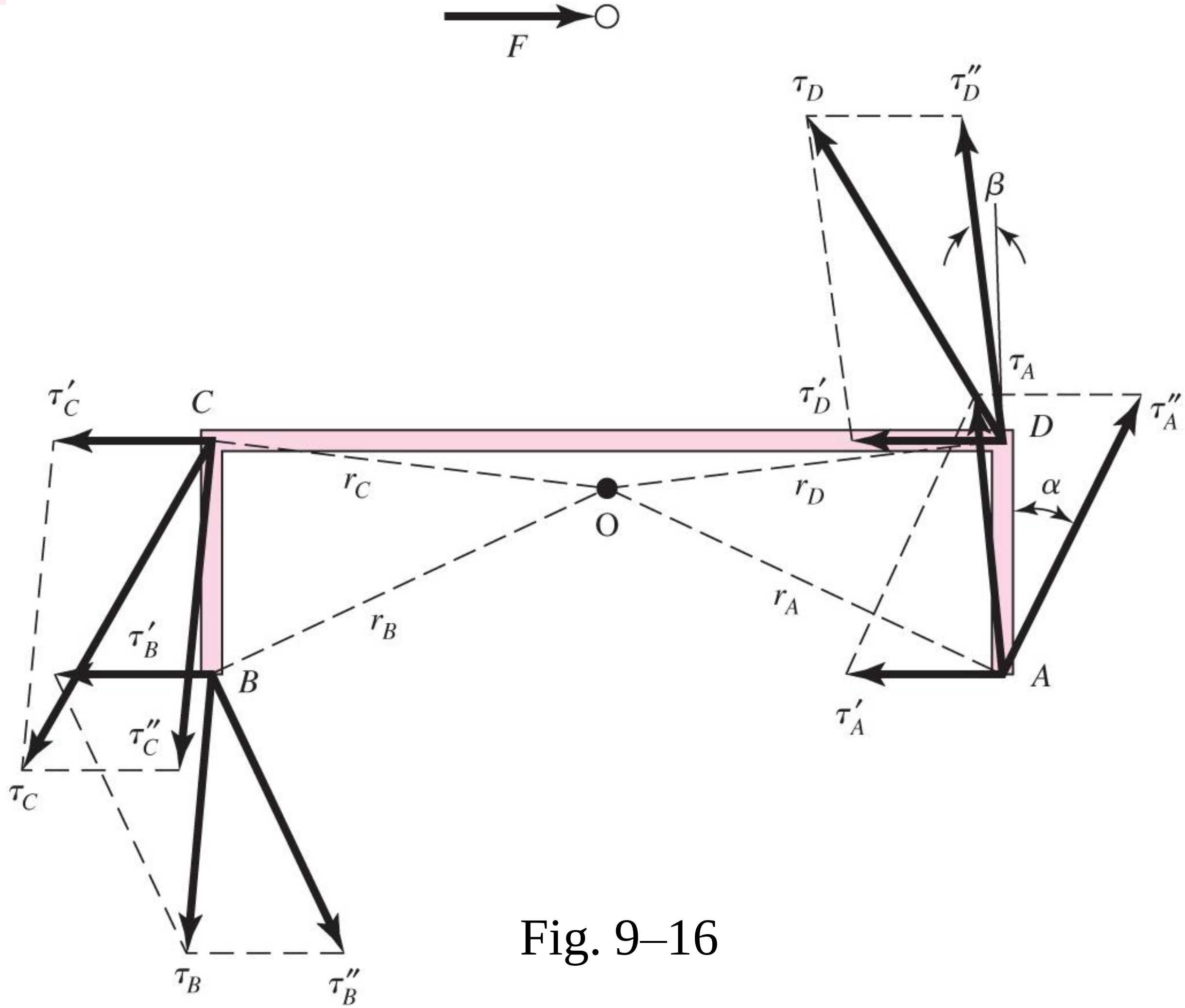


Fig. 9–16

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 τ_A'' makes with the vertical, α, is also the angle r_A makes with the horizontal, which is $\alpha = \tan^{-1}(45.6/95) = 25.64°$. This angle also applies to point B. Thus

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

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

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

(k) Identify the most highly stressed point: $\tau_{max} = \tau_C = \tau_D = 43.9$ MPa

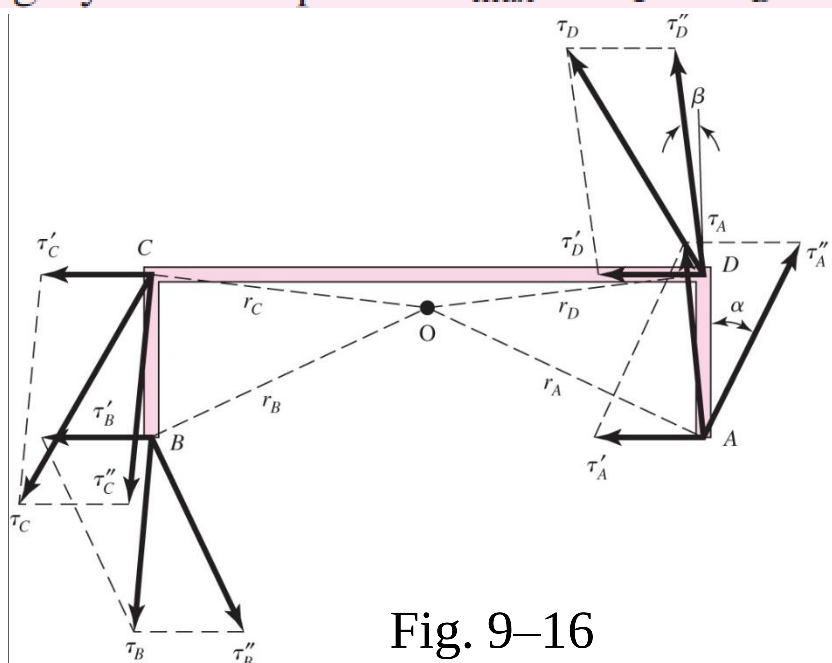


Fig. 9–16

Fillet Welds Loaded in Bending

- Fillet welds carry both shear V and moment M

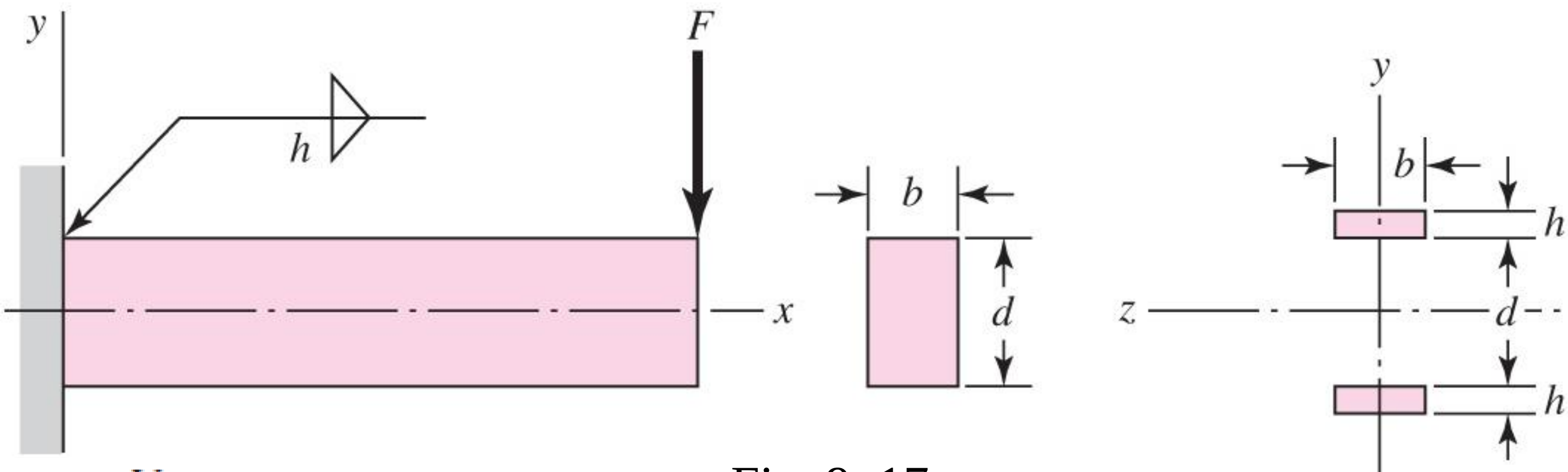


Fig. 9–17

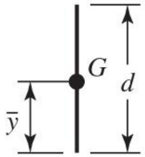
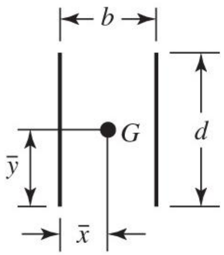
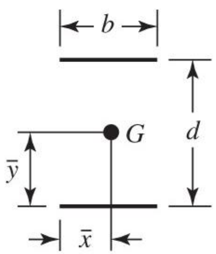
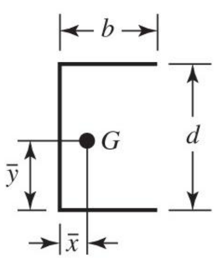
$$\tau' = \frac{V}{A}$$

$$I_u = \frac{bd^2}{2} \qquad I = 0.707hI_u = 0.707h\frac{bd^2}{2}$$

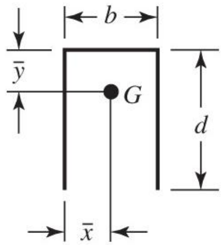
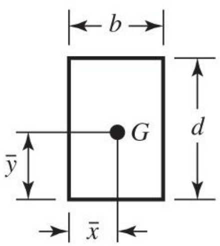
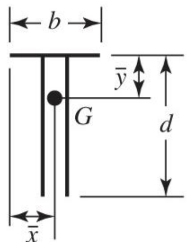
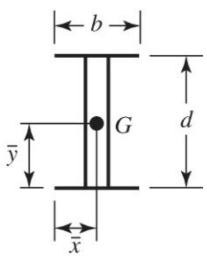
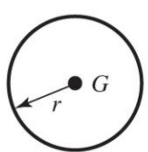
$$\tau'' = \frac{Mc}{I} = \frac{Md/2}{0.707hbd^2/2} = \frac{1.414M}{bdh}$$

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

Bending Properties of Fillet Welds (Table 9–2)

Weld	Throat Area	Location of G	Unit Second Moment of Area
1. 	$A = 0.707hd$	$\bar{x} = 0$ $\bar{y} = d/2$	$I_u = \dfrac{d^3}{12}$
2. 	$A = 1.414hd$	$\bar{x} = b/2$ $\bar{y} = d/2$	$I_u = \dfrac{d^3}{6}$
3. 	$A = 1.414hb$	$\bar{x} = b/2$ $\bar{y} = d/2$	$I_u = \dfrac{bd^2}{2}$
4. 	$A = 0.707h(2b + d)$	$\bar{x} = \dfrac{b^2}{2b + d}$ $\bar{y} = d/2$	$I_u = \dfrac{d^2}{12}(6b + d)$

Bending Properties of Fillet Welds (Table 9–2)

5.		$A = 0.707h(b + 2d)$	$\bar{x} = b/2$ $\bar{y} = \dfrac{d^2}{b + 2d}$	$I_u = \dfrac{2d^3}{3} - 2d^2\bar{y} + (b + 2d)\bar{y}^2$
6.		$A = 1.414h(b + d)$	$\bar{x} = b/2$ $\bar{y} = d/2$	$I_u = \dfrac{d^2}{6}(3b + d)$
7.		$A = 0.707h(b + 2d)$	$\bar{x} = b/2$ $\bar{y} = \dfrac{d^2}{b + 2d}$	$I_u = \dfrac{2d^3}{3} - 2d^2\bar{y} + (b + 2d)\bar{y}^2$
8.		$A = 1.414h(b + d)$	$\bar{x} = b/2$ $\bar{y} = d/2$	$I_u = \dfrac{d^2}{6}(3b + d)$
9.		$A = 1.414\pi hr$		$I_u = \pi r^3$

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)

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

*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

Type of Loading	Type of Weld	Permissible Stress	n^*
Tension	Butt	$0.60S_y$	1.67
Bearing	Butt	$0.90S_y$	1.11
Bending	Butt	0.60–$0.66S_y$	1.52–1.67
Simple compression	Butt	$0.60S_y$	1.67
Shear	Butt or fillet	$0.30S_{ut}^{\dagger}$	

*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

Fatigue Stress-Concentration Factors, K_{fs}

Type of Weld	K_{fs}
Reinforced butt weld	1.2
Toe of transverse fillet weld	1.5
End of parallel fillet weld	2.7
T-butt joint with sharp corners	2.0

Allowable Load or Various Sizes of Fillet Welds (Table 9–6)

Strength Level of Weld Metal (EXX)							
	60*	70*	80	90*	100	110*	120
Allowable shear stress on throat, ksi (1000 psi) of fillet weld or partial penetration groove weld							
$\tau =$	18.0	21.0	24.0	27.0	30.0	33.0	36.0
Allowable Unit Force on Fillet Weld, kip/linear in							
$^{\dagger}f =$	$12.73h$	$14.85h$	$16.97h$	$19.09h$	$21.21h$	$23.33h$	$25.45h$
Leg Size h, in	Allowable Unit Force for Various Sizes of Fillet Welds kip/linear in						
1	12.73	14.85	16.97	19.09	21.21	23.33	25.45
7/8	11.14	12.99	14.85	16.70	18.57	20.41	22.27
3/4	9.55	11.14	12.73	14.32	15.92	17.50	19.09
5/8	7.96	9.28	10.61	11.93	13.27	14.58	15.91
1/2	6.37	7.42	8.48	9.54	10.61	11.67	12.73
7/16	5.57	6.50	7.42	8.35	9.28	10.21	11.14
3/8	4.77	5.57	6.36	7.16	7.95	8.75	9.54
5/16	3.98	4.64	5.30	5.97	6.63	7.29	7.95
1/4	3.18	3.71	4.24	4.77	5.30	5.83	6.36
3/16	2.39	2.78	3.18	3.58	3.98	4.38	4.77
1/8	1.59	1.86	2.12	2.39	2.65	2.92	3.18
1/16	0.795	0.930	1.06	1.19	1.33	1.46	1.59

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

$^{\dagger}f = 0.707h\,\tau_{\text{all}}$.

Minimum Fillet Weld Size, *h* (Table 9–6)

Material Thickness of Thicker Part Joined, in		Weld Size, in
*To $\frac{1}{4}$ incl.		$\frac{1}{8}$
Over $\frac{1}{4}$	To $\frac{1}{2}$	$\frac{3}{16}$
Over $\frac{1}{2}$	To $\frac{3}{4}$	$\frac{1}{4}$
†Over $\frac{3}{4}$	To $1\frac{1}{2}$	$\frac{5}{16}$
Over $1\frac{1}{2}$	To $2\frac{1}{4}$	$\frac{3}{8}$
Over $2\frac{1}{4}$	To 6	$\frac{1}{2}$
Over 6		$\frac{5}{8}$

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?

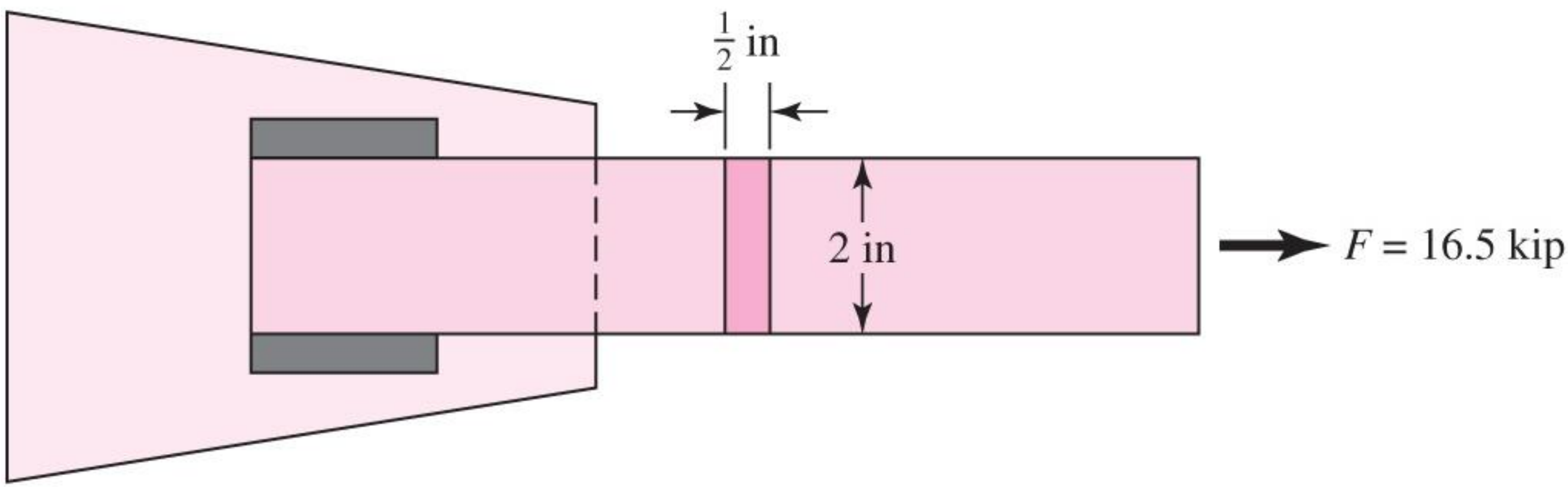


Fig. 9–18

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

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

The shear stress τ on the base metal adjacent to the weld is

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

Example 9–2

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

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

The allowable tensile stress σ_{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 9–3

A specially rolled A36 structural steel section for the attachment has a cross section as shown in Fig. 9–19 and has yield and ultimate tensile strengths of 36 and 58 kpsi, respectively. It is statically loaded through the attachment centroid by a load of $F = 24$ kip. Unsymmetrical weld tracks can compensate for eccentricity such that there is no moment to be resisted by the welds. Specify the weld track lengths l_1 and l_2 for a $\frac{5}{16}$-in fillet weld using an E70XX electrode. This is part of a design problem in which the design variables include weld lengths and the fillet leg size.

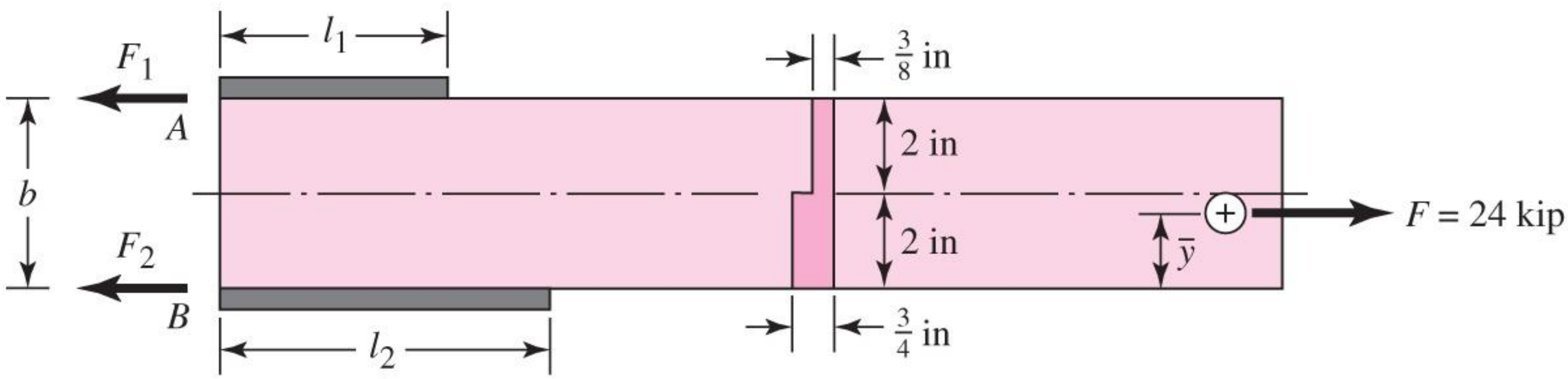


Fig. 9–19

Example 9–3

The y coordinate of the section centroid of the attachment is

$$\bar{y} = \frac{\sum y_i A_i}{\sum A_i} = \frac{1(0.75)2 + 3(0.375)2}{0.75(2) + 0.375(2)} = 1.67 \text{ in}$$

Summing moments about point B to zero gives

$$\sum M_B = 0 = -F_1 b + F\bar{y} = -F_1(4) + 24(1.67)$$

from which

$$F_1 = 10 \text{ kip}$$

It follows that

$$F_2 = 24 - 10.0 = 14.0 \text{ kip}$$

The weld throat areas have to be in the ratio $14/10 = 1.4$, that is, $l_2 = 1.4 l_1$. The weld length design variables are coupled by this relation, so l_1 is the weld length design variable. The other design variable is the fillet weld leg size h, which has been decided by the problem statement. From Table 9–4, the allowable shear stress on the throat τ_{all} is

$$\tau_{all} = 0.3(70) = 21 \text{ kpsi}$$

Example 9–3

The shear stress τ on the 45° throat is

$$\tau = \frac{F}{(0.707)h(l_1 + l_2)} = \frac{F}{(0.707)h(l_1 + 1.4l_1)}$$

$$= \frac{F}{(0.707)h(2.4l_1)} = \tau_{\text{all}} = 21\,\text{kpsi}$$

from which the weld length l_1 is

$$l_1 = \frac{24}{21(0.707)0.3125(2.4)} = 2.16\ \text{in}$$

and

$$l_2 = 1.4l_1 = 1.4(2.16) = 3.02\ \text{in}$$

Example 9–3

The shear stress τ on the 45° throat is

$$\tau = \frac{F}{(0.707)h(l_1 + l_2)} = \frac{F}{(0.707)h(l_1 + 1.4l_1)}$$

$$= \frac{F}{(0.707)h(2.4l_1)} = \tau_{\text{all}} = 21\ \text{kpsi}$$

from which the weld length l_1 is

$$l_1 = \frac{24}{21(0.707)0.3125(2.4)} = 2.16\ \text{in}$$

and

$$l_2 = 1.4l_1 = 1.4(2.16) = 3.02\ \text{in}$$

These are the weld-bead lengths required by weld metal strength. The attachment shear stress allowable in the base metal, from Table 9–4, is

$$\tau_{\text{all}} = 0.4S_y = 0.4(36) = 14.4\ \text{kpsi}$$

Example 9–3

The shear stress τ in the base metal adjacent to the weld is

$$\tau = \frac{F}{h(l_1 + l_2)} = \frac{F}{h(l_1 + 1.4l_1)} = \frac{F}{h(2.4l_1)} = \tau_{\text{all}} = 14.4 \text{ kpsi}$$

from which

$$l_1 = \frac{F}{14.4h(2.4)} = \frac{24}{14.4(0.3125)2.4} = 2.22 \text{ in}$$

$$l_2 = 1.4l_1 = 1.4(2.22) = 3.11 \text{ in}$$

These are the weld-bead lengths required by base metal (attachment) strength. The base metal controls the weld lengths. For the allowable tensile stress σ_{all} in the shank of the attachment, the AISC allowable for tension members is $0.6S_y$; therefore,

$$\sigma_{\text{all}} = 0.6S_y = 0.6(36) = 21.6 \text{ kpsi}$$

Example 9–3

The nominal tensile stress σ is *uniform* across the attachment cross section because of the load application at the centroid. The stress σ is

$$\sigma = \frac{F}{A} = \frac{24}{0.75(2) + 2(0.375)} = 10.7 \text{ kpsi}$$

Since $\sigma \leq \sigma_{\text{all}}$, the shank section is satisfactory. With l_1 set to a nominal $2\frac{1}{4}$ in, l_2 should be $1.4(2.25) = 3.15$ in.

Set $l_1 = 2\frac{1}{4}$ in, $l_2 = 3\frac{1}{4}$ in. The small magnitude of the departure from $l_2/l_1 = 1.4$ is not serious. The joint is essentially moment-free.

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.

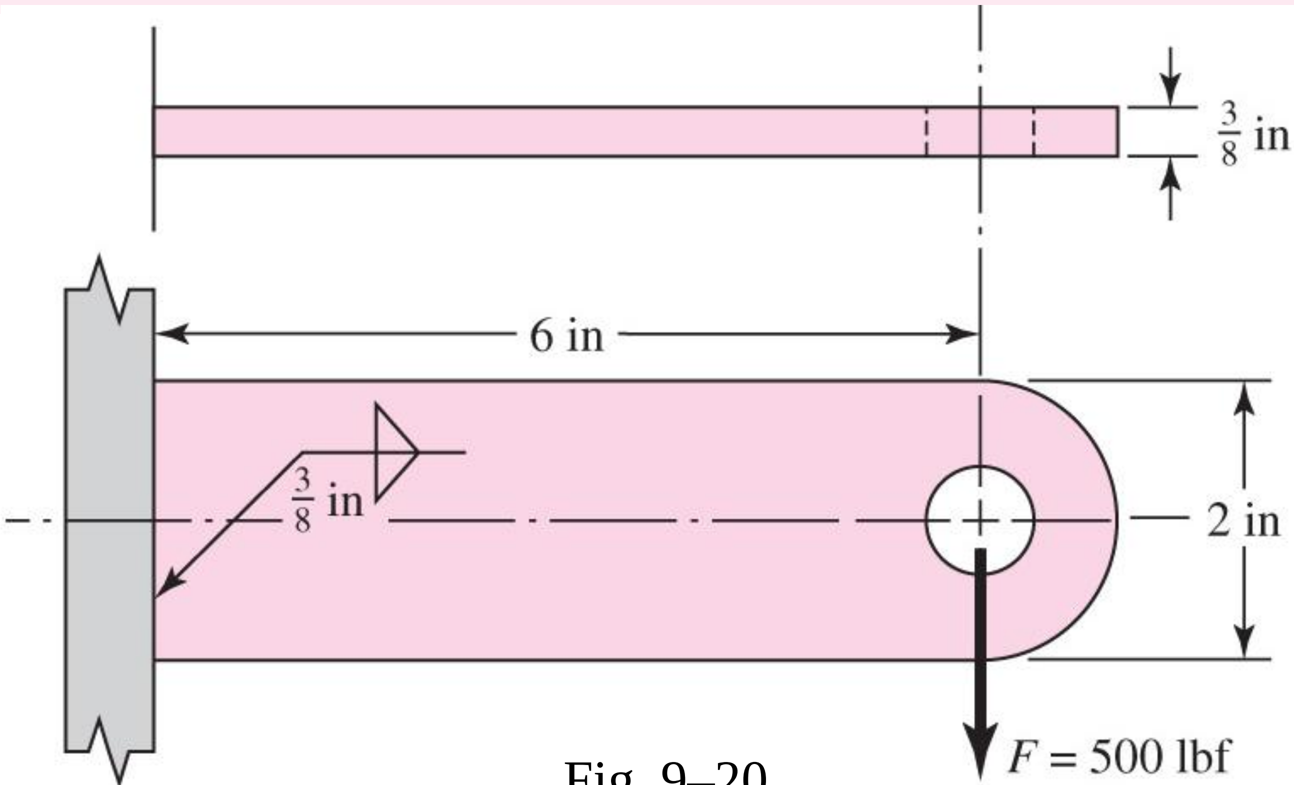


Fig. 9–20

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 τ 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

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

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

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 E6010 electrode Table 9–6 gives the allowable shear stress τ_{all} as 18 kpsi. Since $\tau < \tau_{\text{all}}$, 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.

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.

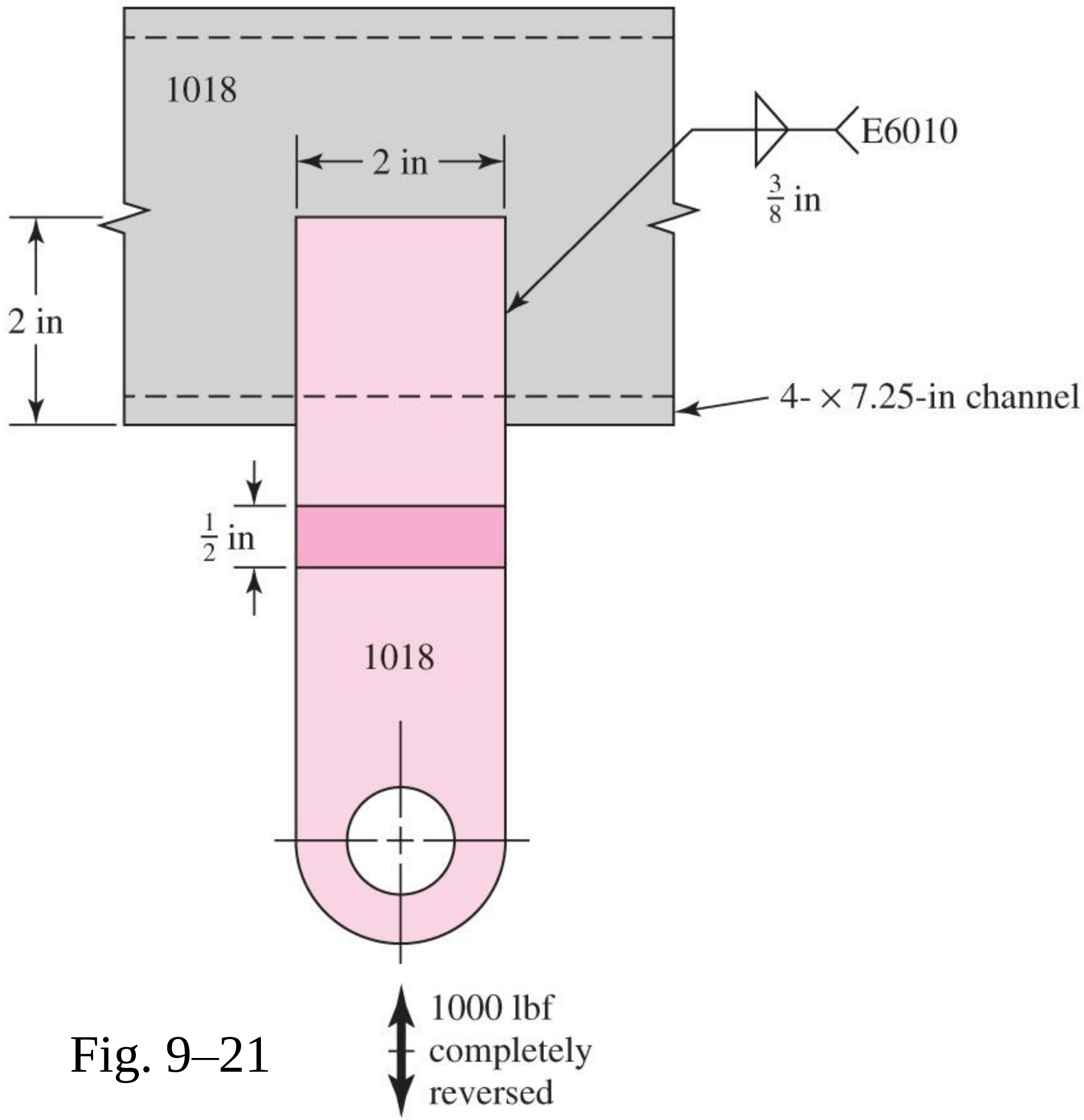


Fig. 9–21

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:

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

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

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

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

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

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

Example 9–5

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

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

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

$$n_f = \frac{S_{se}}{\tau'_a} = \frac{12\,000}{2545} = 4.72$$

Example 9–6

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

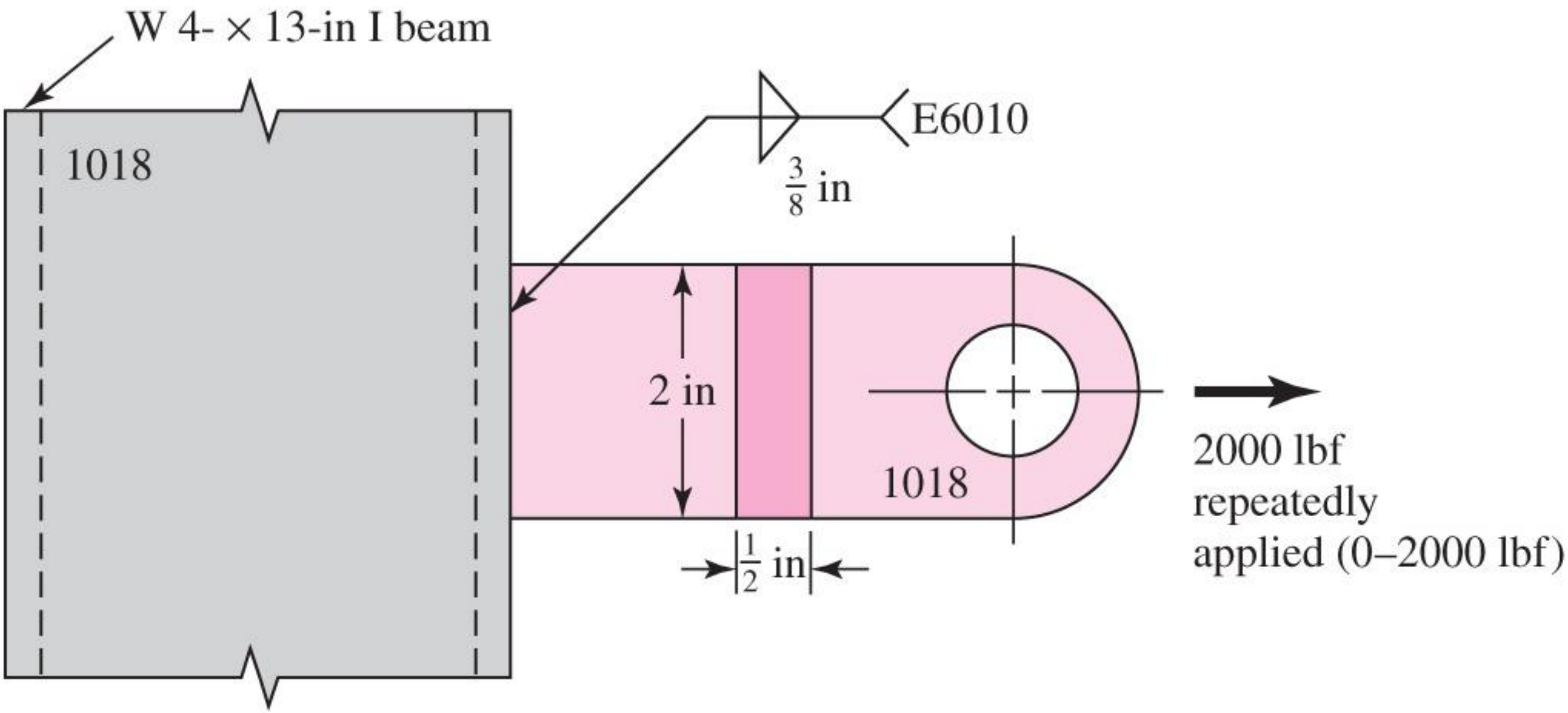


Fig. 9–22

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, $S_{su} \doteq 0.67S_{ut}$. This, together with the Gerber fatigue failure criterion for shear stresses from Table 6–7, p. 307, gives

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

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

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

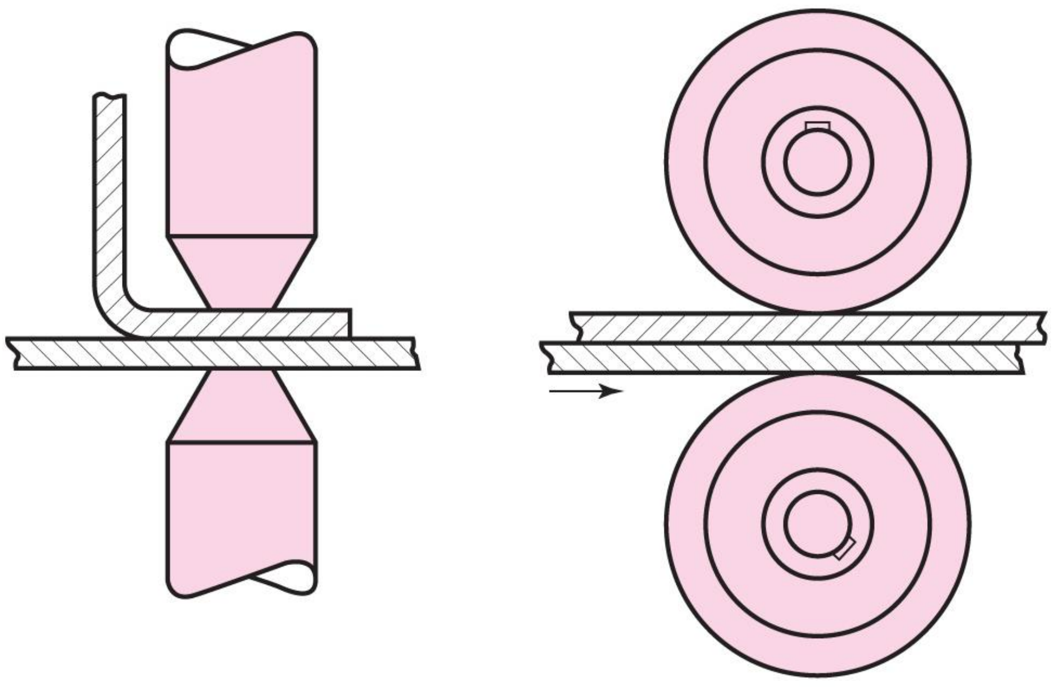


Fig. 9–23

(*a*) (*b*)

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

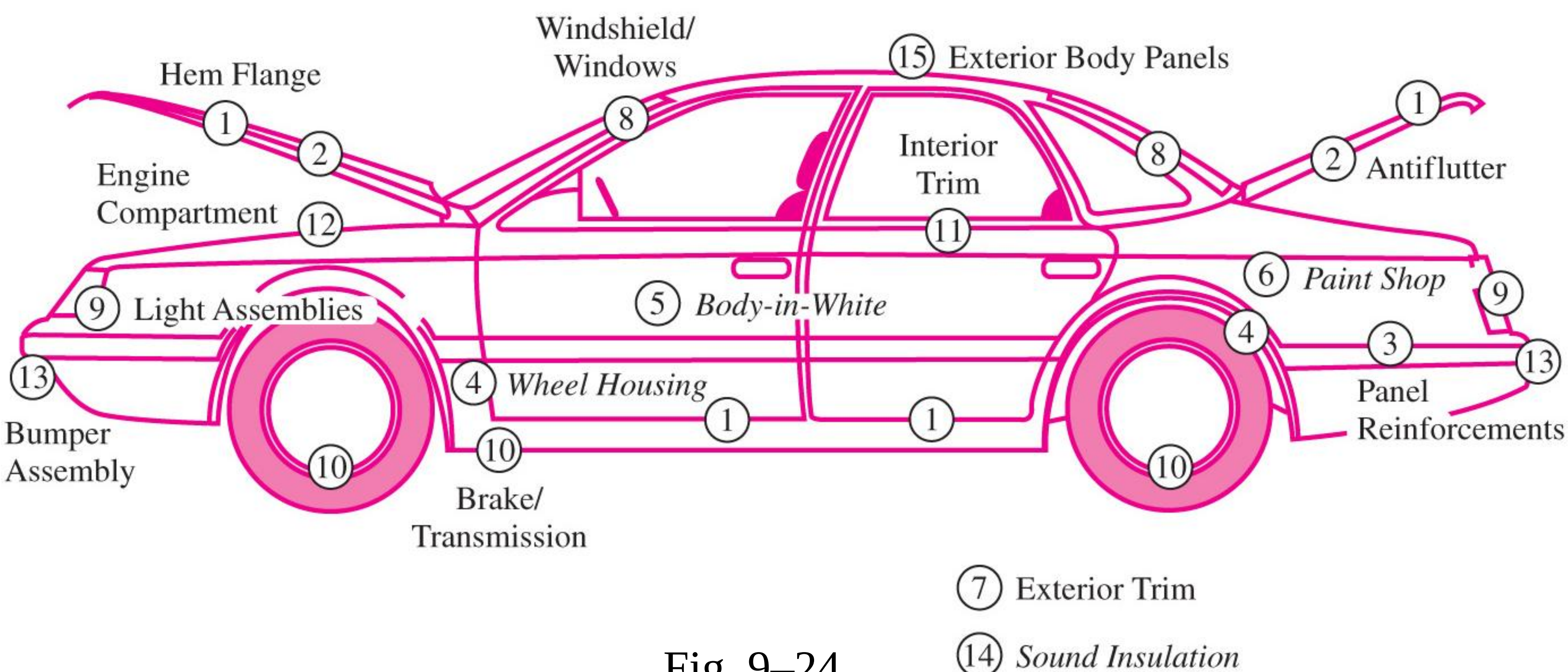


Fig. 9–24

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

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

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

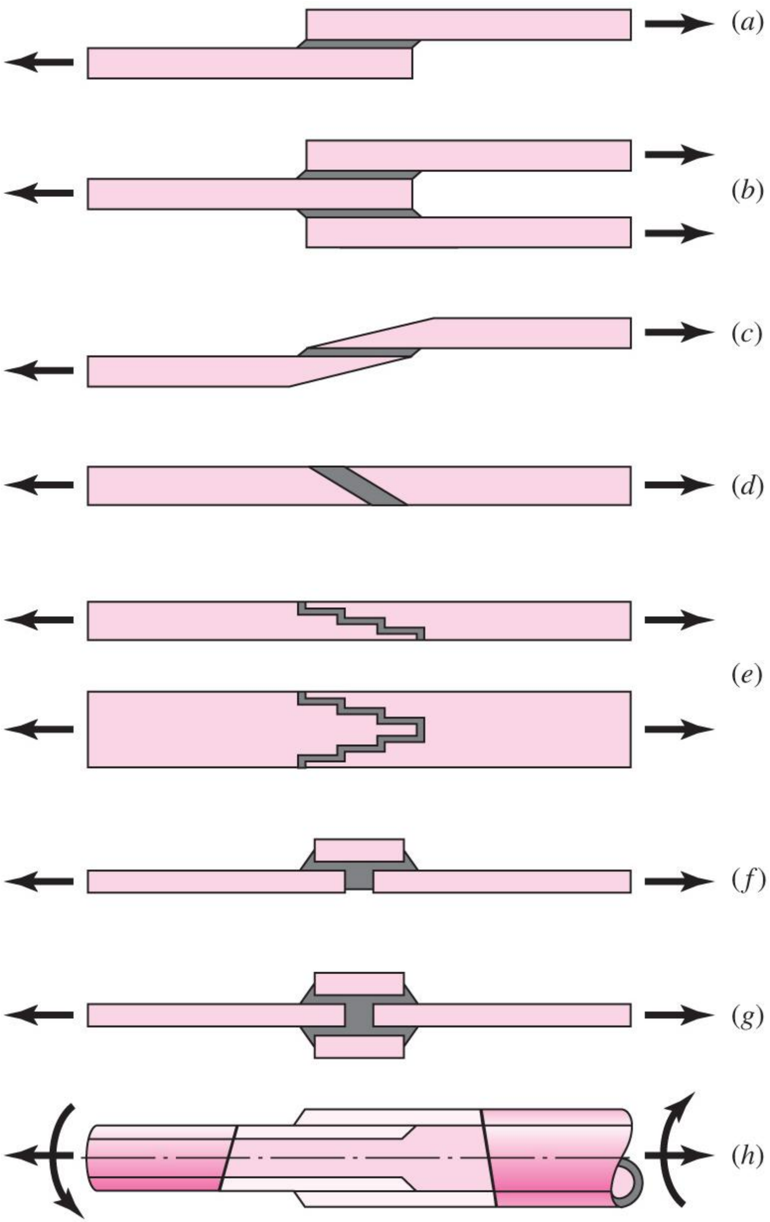


Fig. 9–25

Double-lap Joint

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

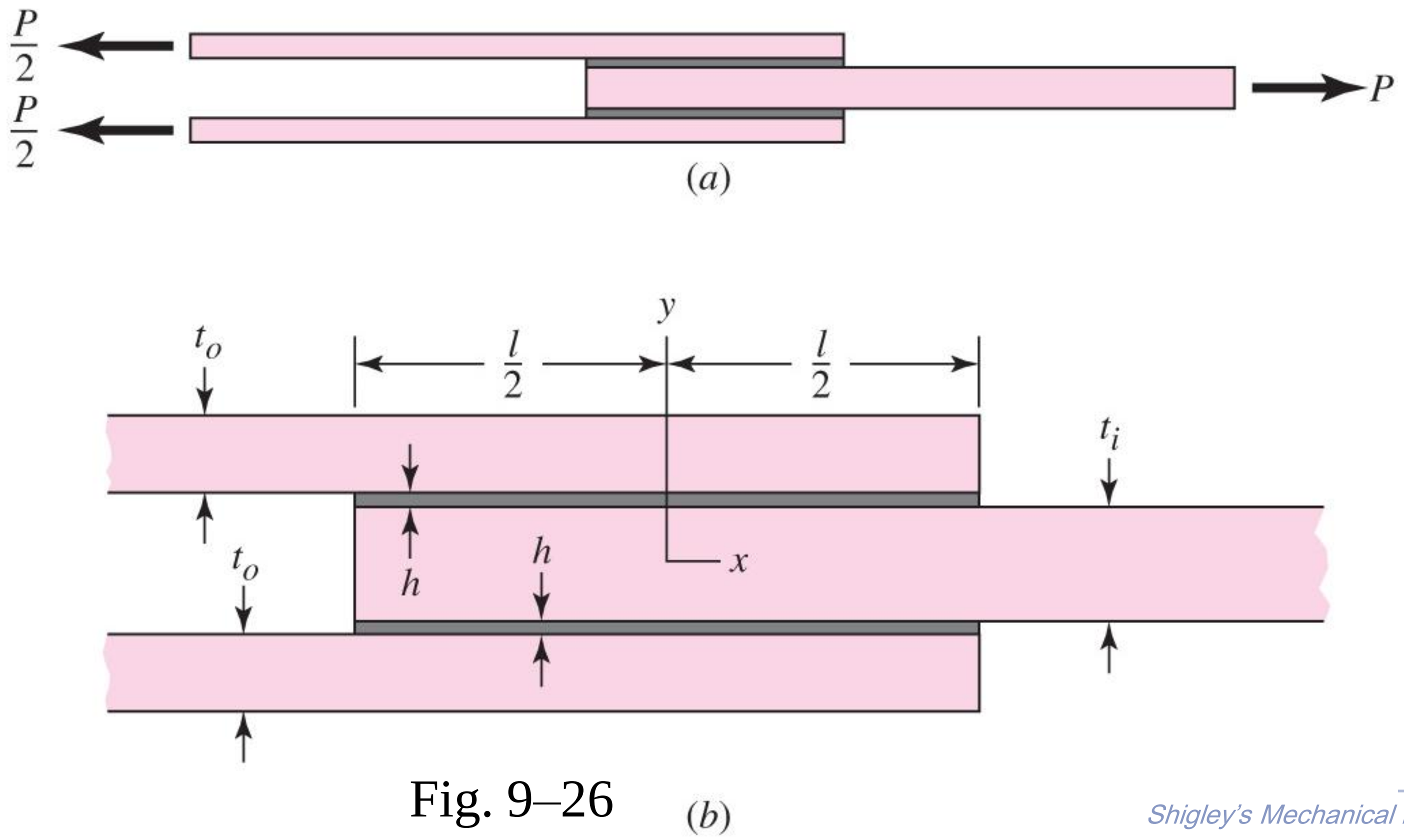


Fig. 9–26

Double-lap Joint

- Shear-stress distribution is given by

$$\tau(x) = \frac{P\omega}{4b\sinh(\omega l/2)}\cosh(\omega x) + \left[\frac{P\omega}{4b\cosh(\omega l/2)}\left(\frac{2E_o t_o - E_i t_i}{2E_o t_o + E_i t_i}\right) + \frac{(\alpha_i - \alpha_o)\,\Delta T\,\omega}{(1/E_o t_o + 2/E_i t_i)\cosh(\omega l/2)}\right]\sinh(\omega x) \qquad (9\text{–}7)$$

where

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

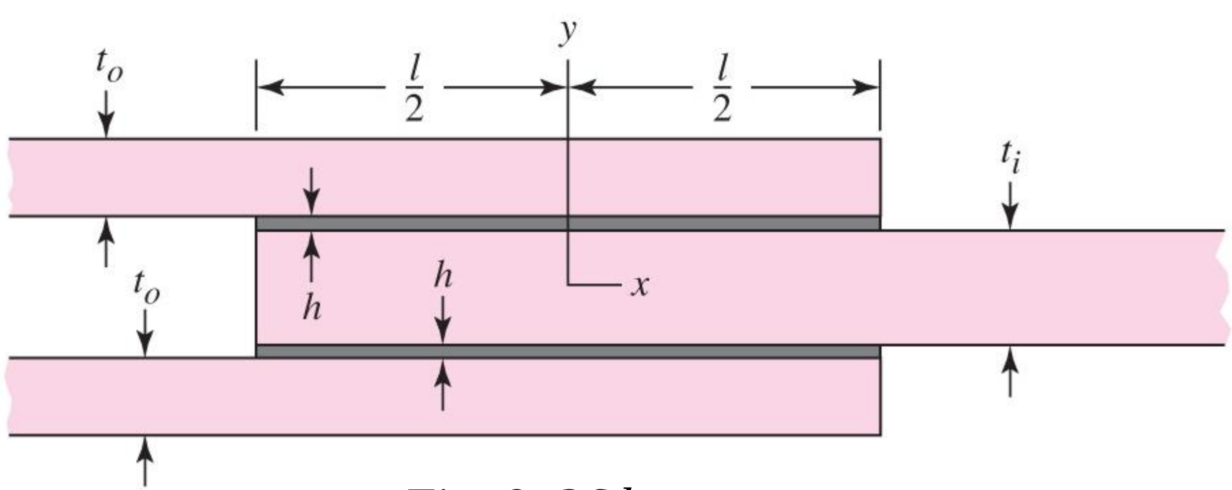


Fig. 9–26*b*

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:

	G, psi	E, psi	α, in/(in · °F)	Thickness, in
Adhesive	$0.2(10^6)$		$55(10^{-6})$	0.020
Outer adherend		$10(10^6)$	$13.3(10^{-6})$	0.150
Inner adherend		$30(10^6)$	$6.0(10^{-6})$	0.100

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.

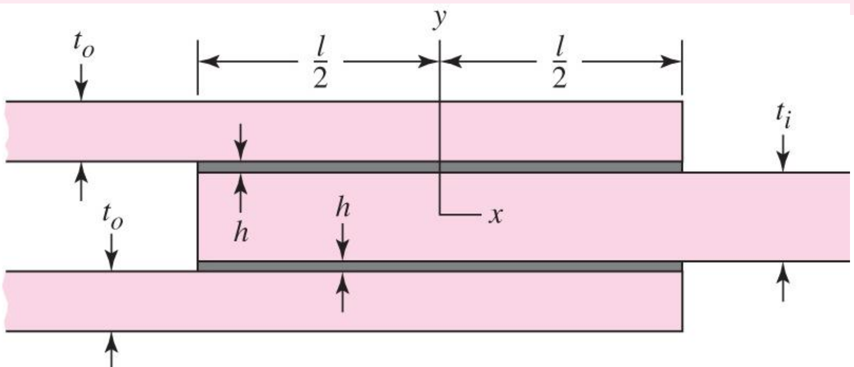


Fig. 9–26

Example 9–7

In Eq. (9–7) the parameter ω 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 · °F), $\Delta T = 70 - 200 = -130$°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.

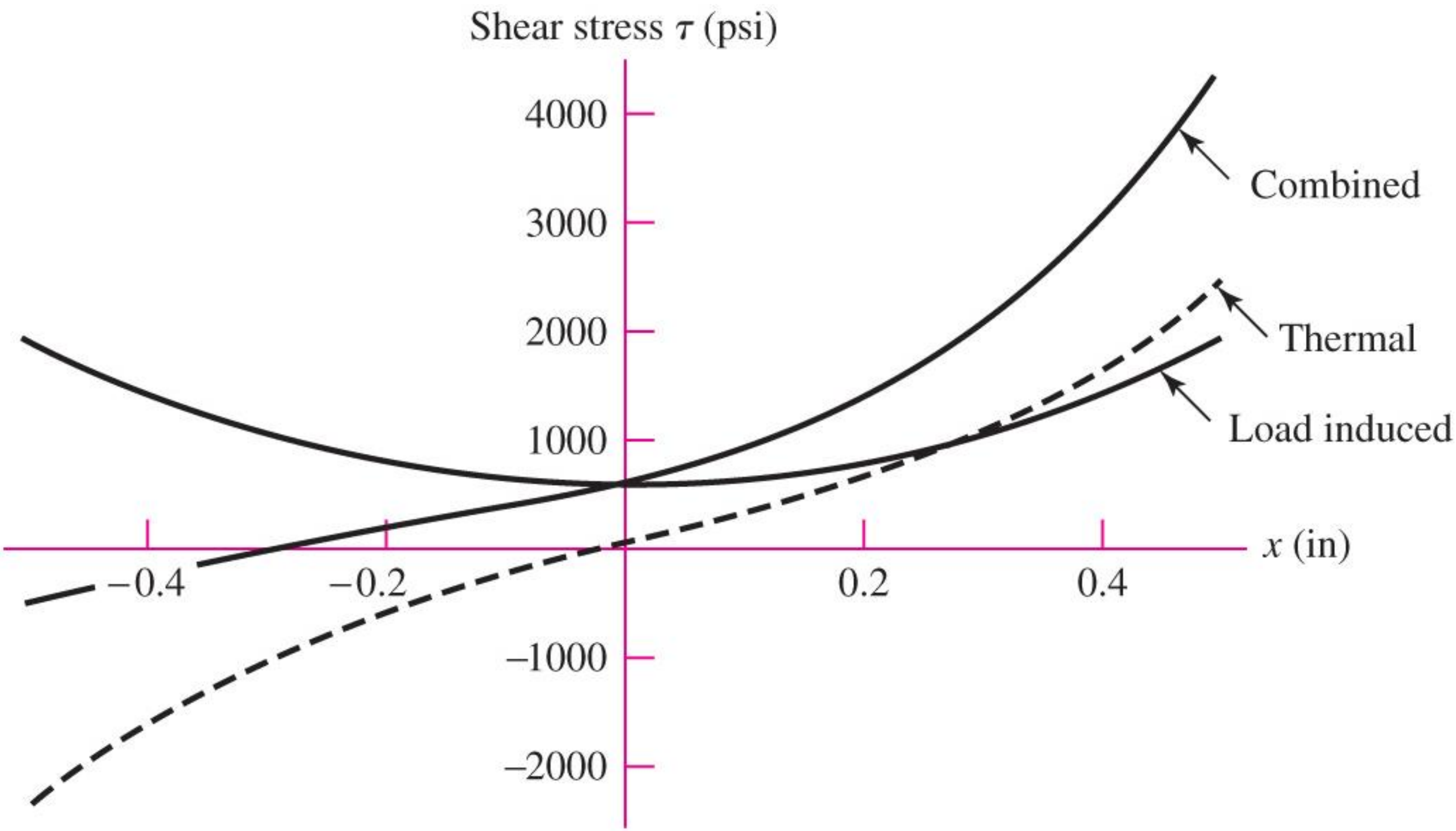


Fig. 9–27

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

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

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):

	$\tau(-0.5)$	$\tau(0)$	$\tau(0.5)$
Thermal only	−2466	0	2466
Load-induced only	1922	604	1922
Combined	−544	604	4388

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

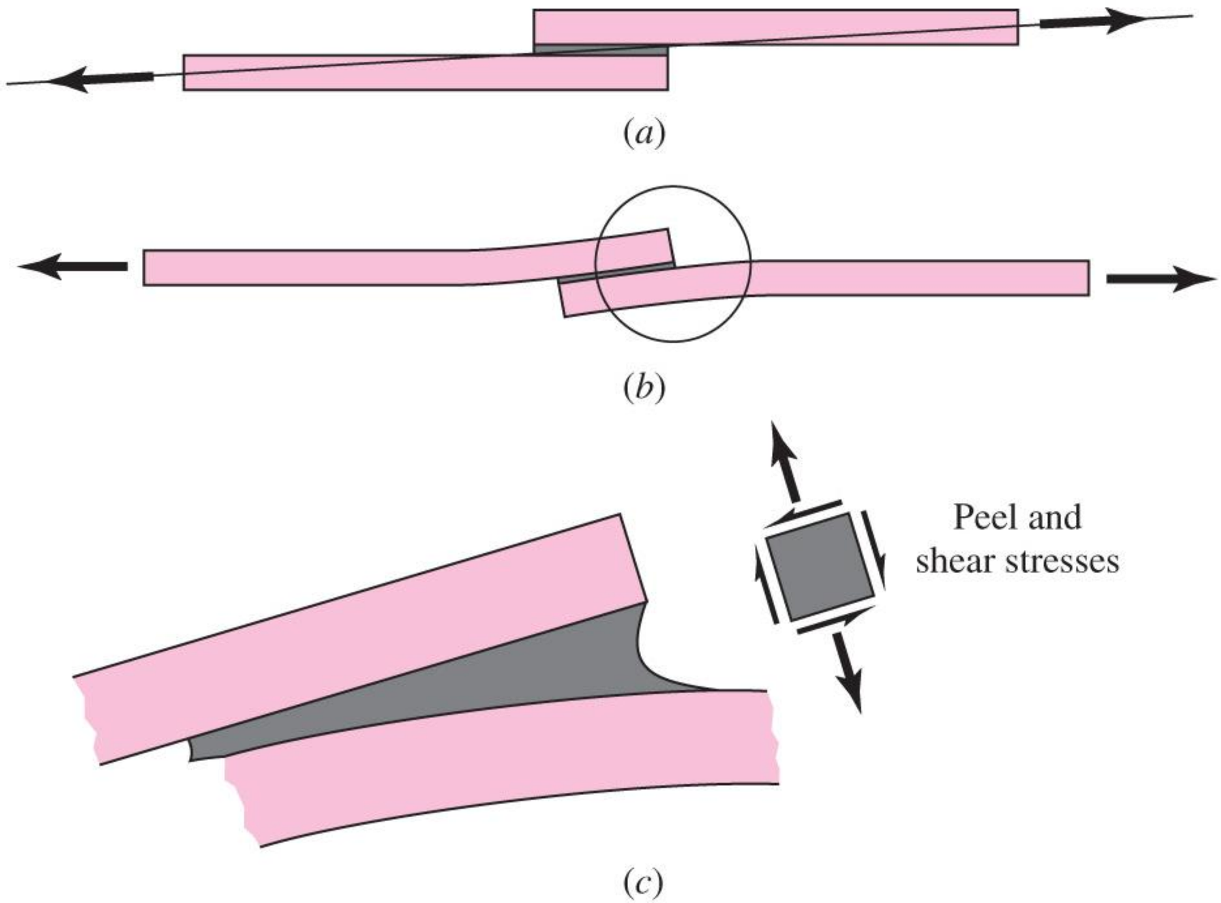


Fig. 9–28

Single-lap Joint

- Shear and peal stresses in single-lap joint, as calculated by Goland and Reissner
- Volkersen curve is for double-lap joint

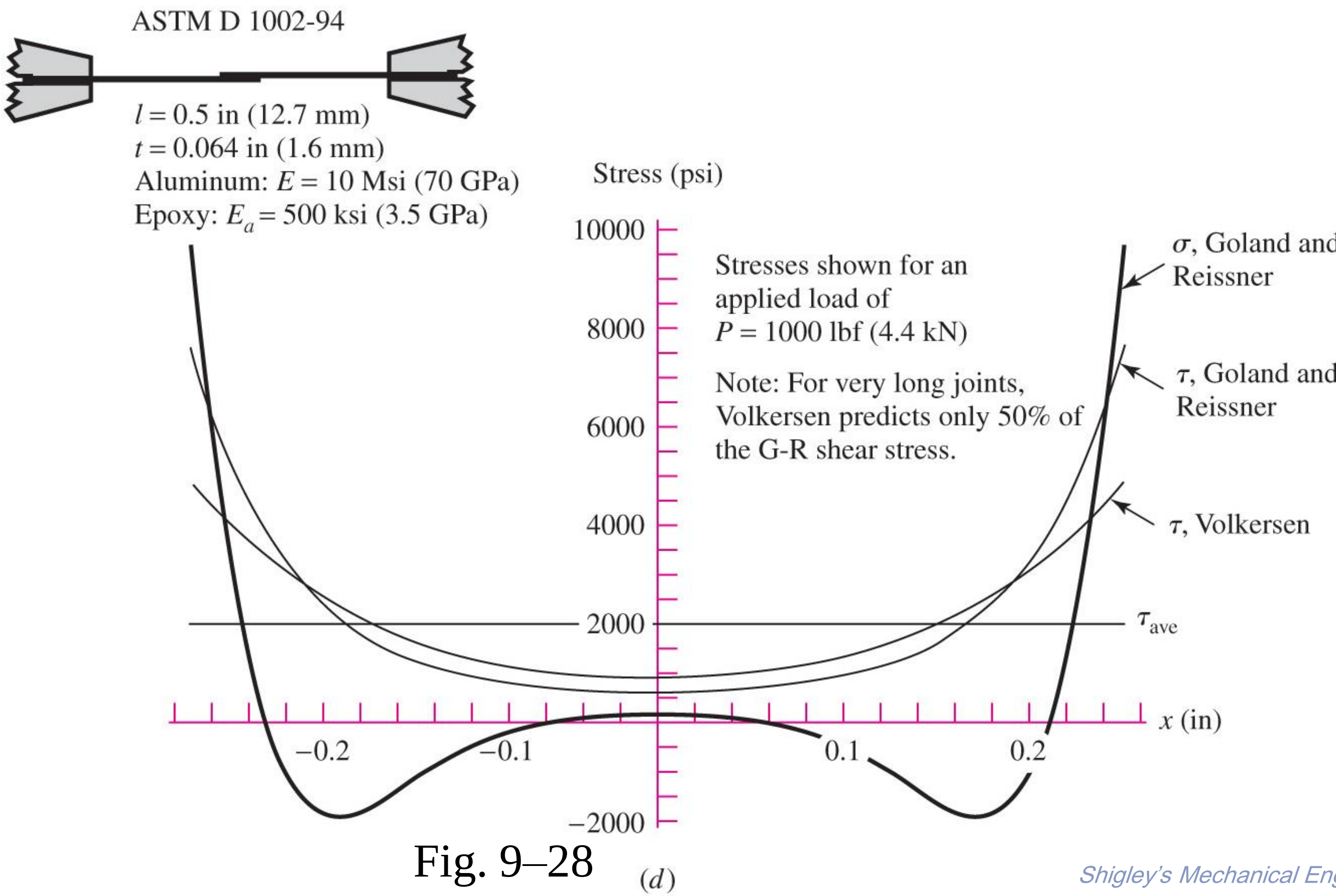


Fig. 9–28

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

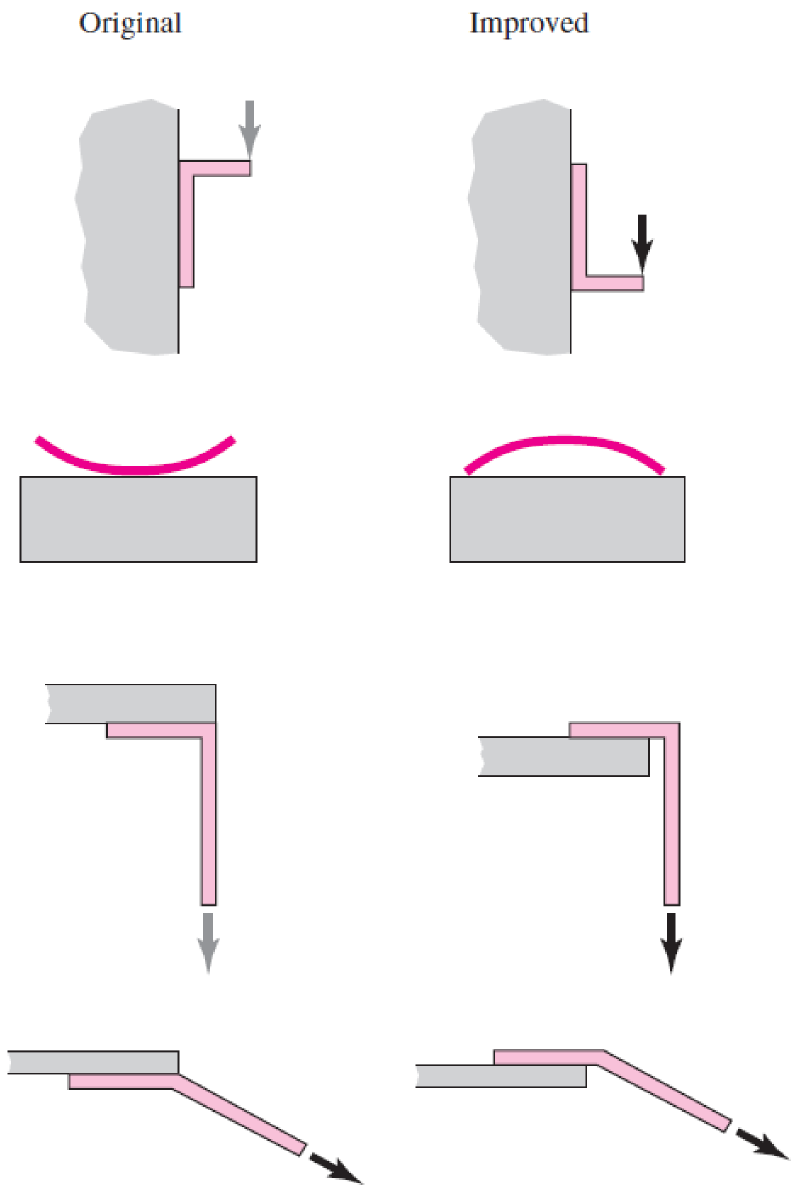


Fig. 9–29

Design Ideas for Improved Bonding

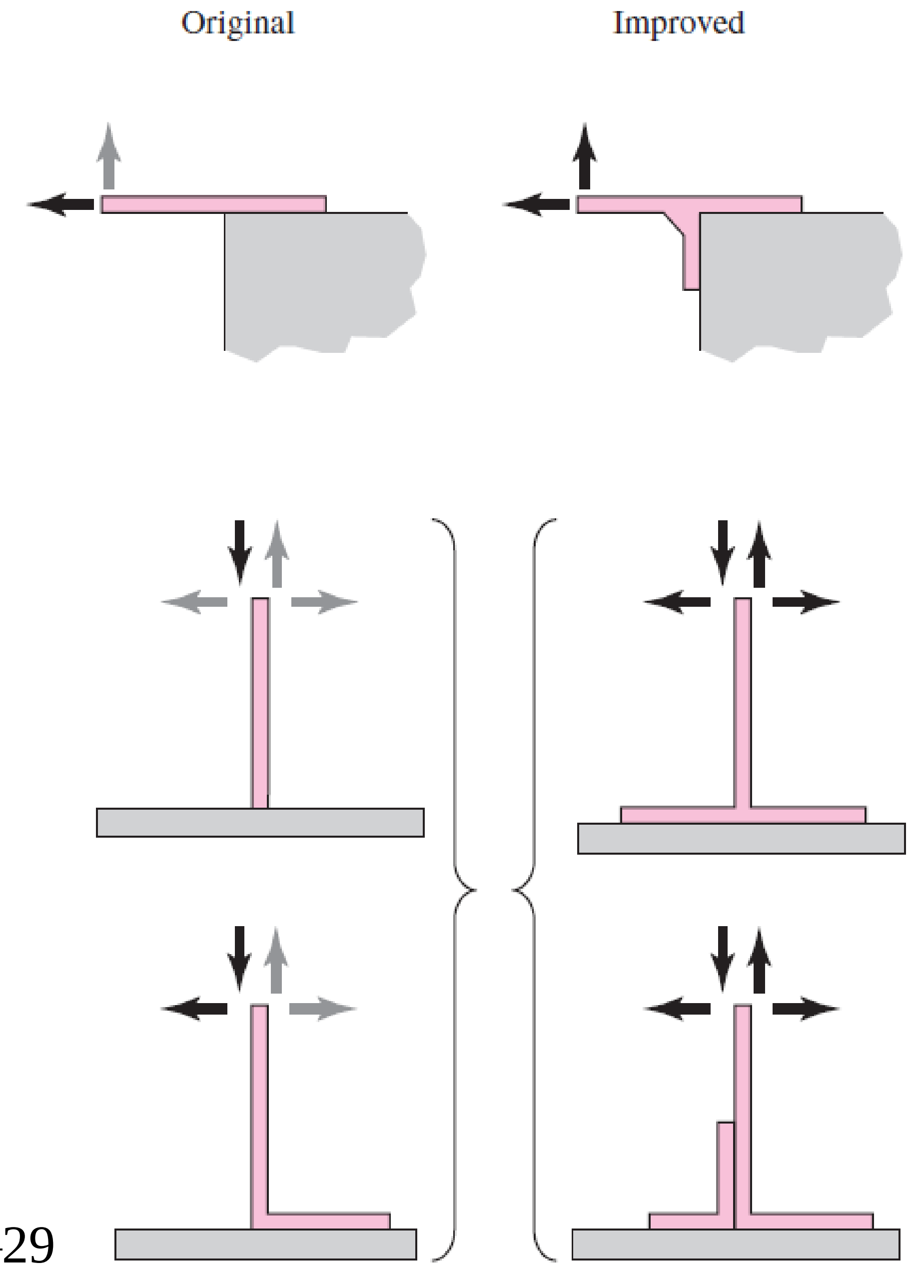


Fig. 9–29

Design Ideas for Improved Bonding

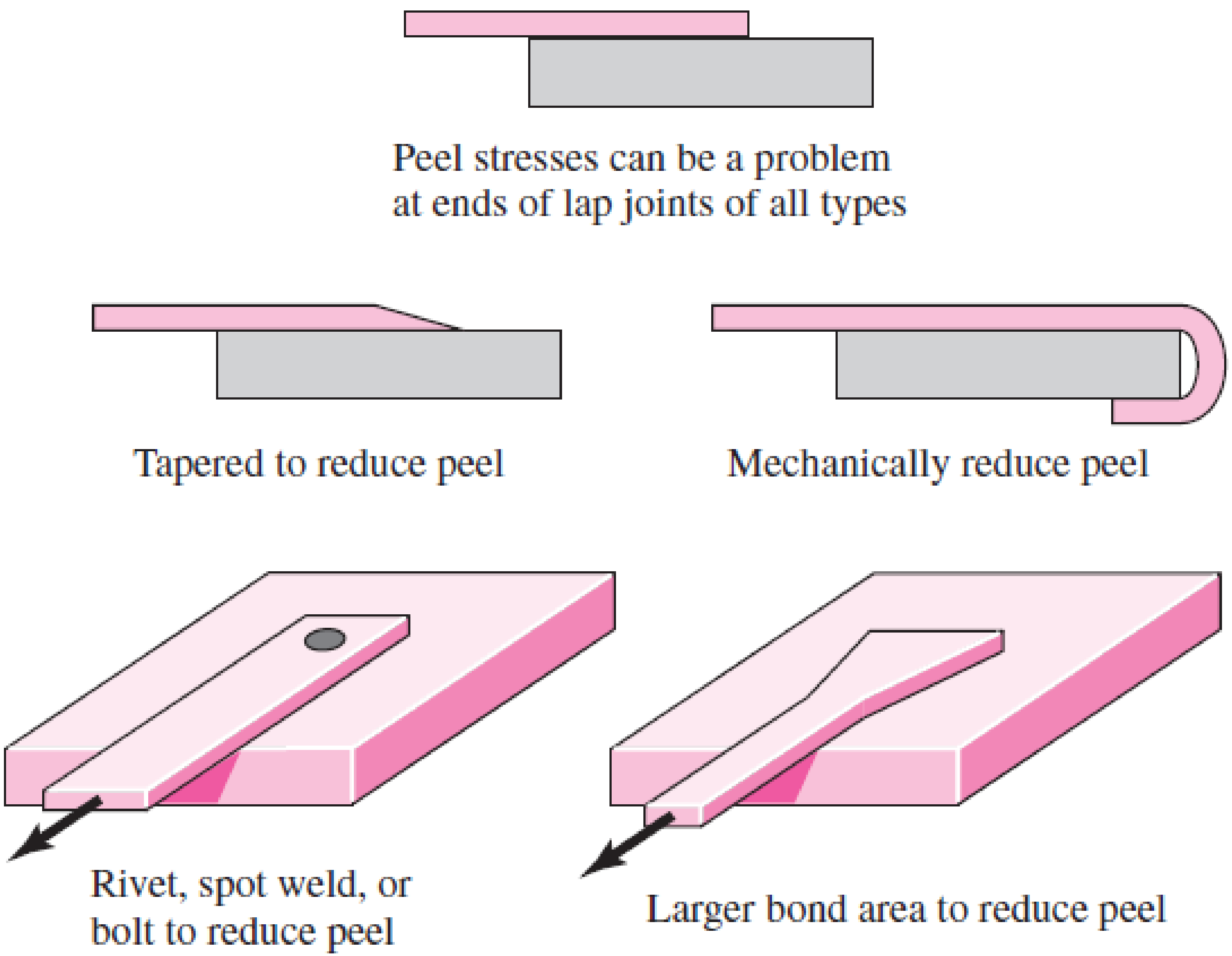


Fig. 9–29