Welded and Bonded Joints

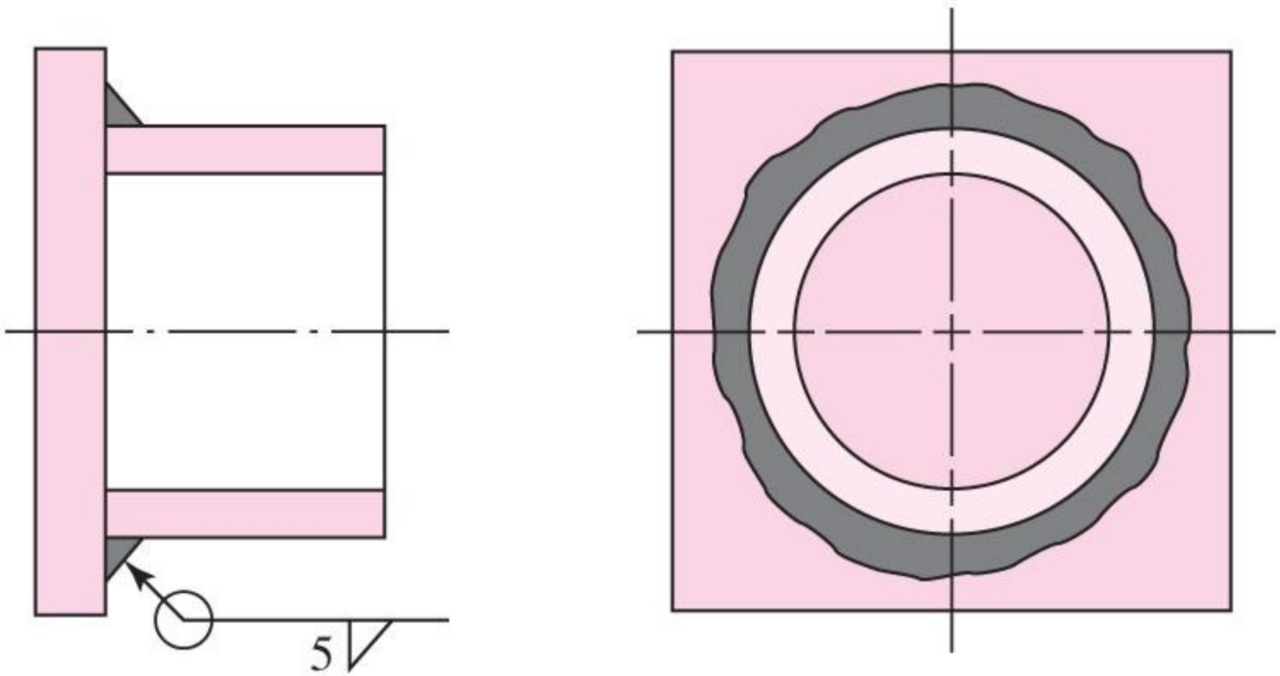
ME 423: Machine Design
Instructor: Ramesh Singh

Outline

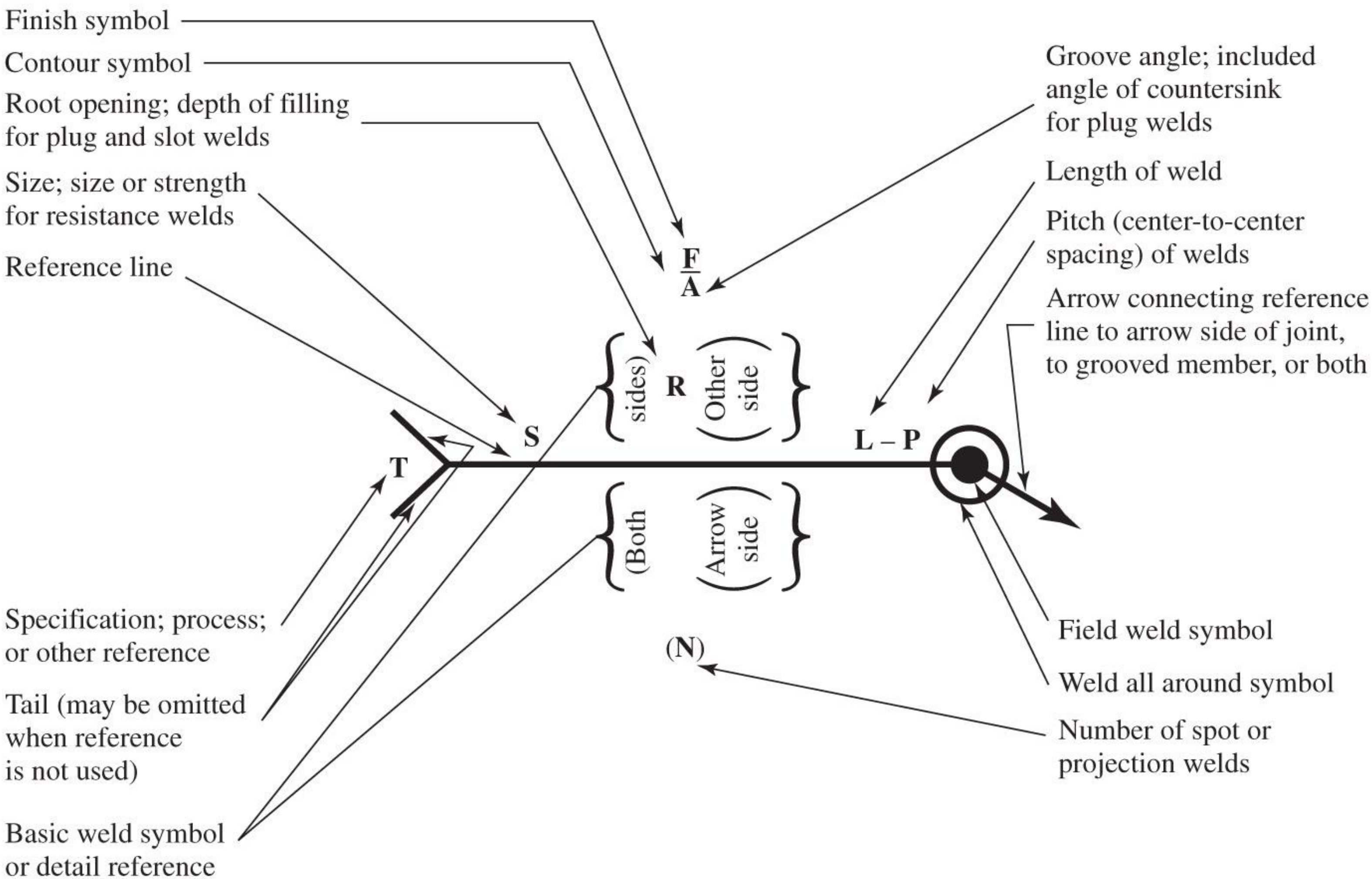
- Welding Symbols
- Butt and Fillet Welds
- Stresses in Welded Joints in Torsion
- Stresses in Welded Joints in Bending
- The Strength of Welded Joints
- Static Loading
- Fatigue Loading

Welding Symbols

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



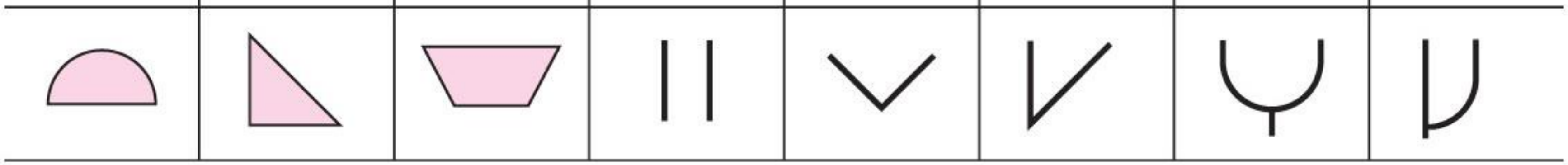
Welding Symbols



Welding Symbols

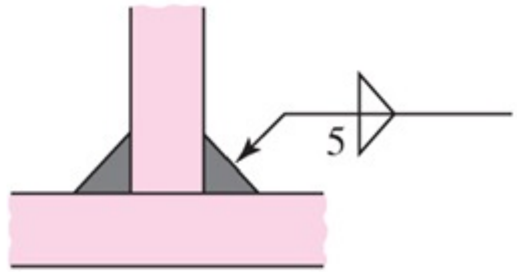
- Arrow side of a joint is the line, side, area, or near member to which the arrow points
- The side opposite the arrow side is the other side
- Shape of weld is shown with the symbols below

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

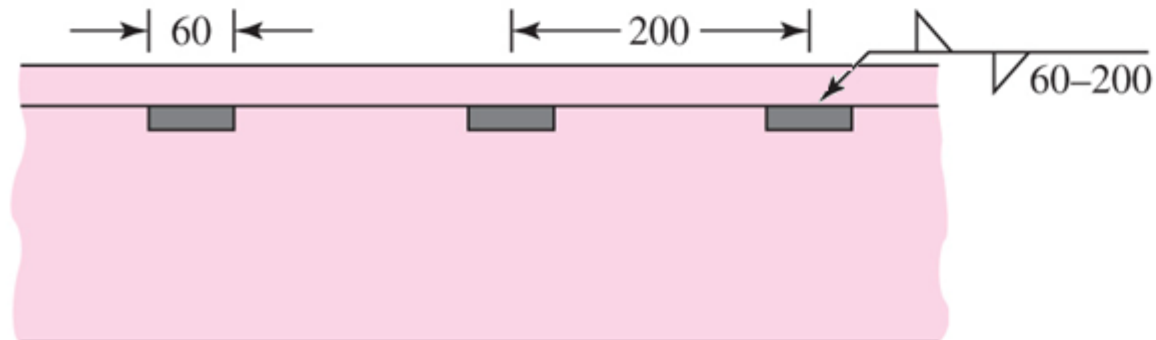


Symbols

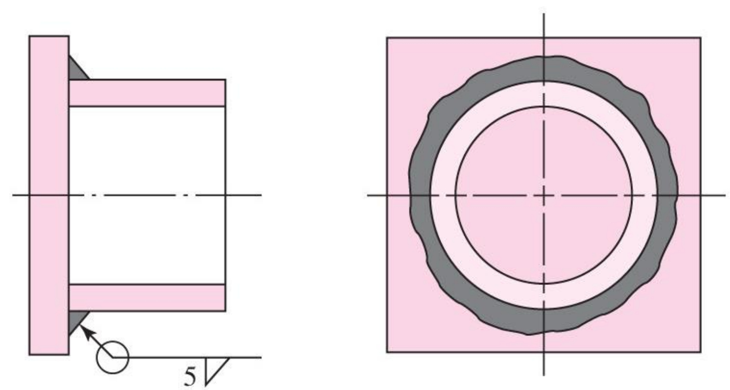
- Weld leg size of 5 mm
- Fillet weld
- Both sides



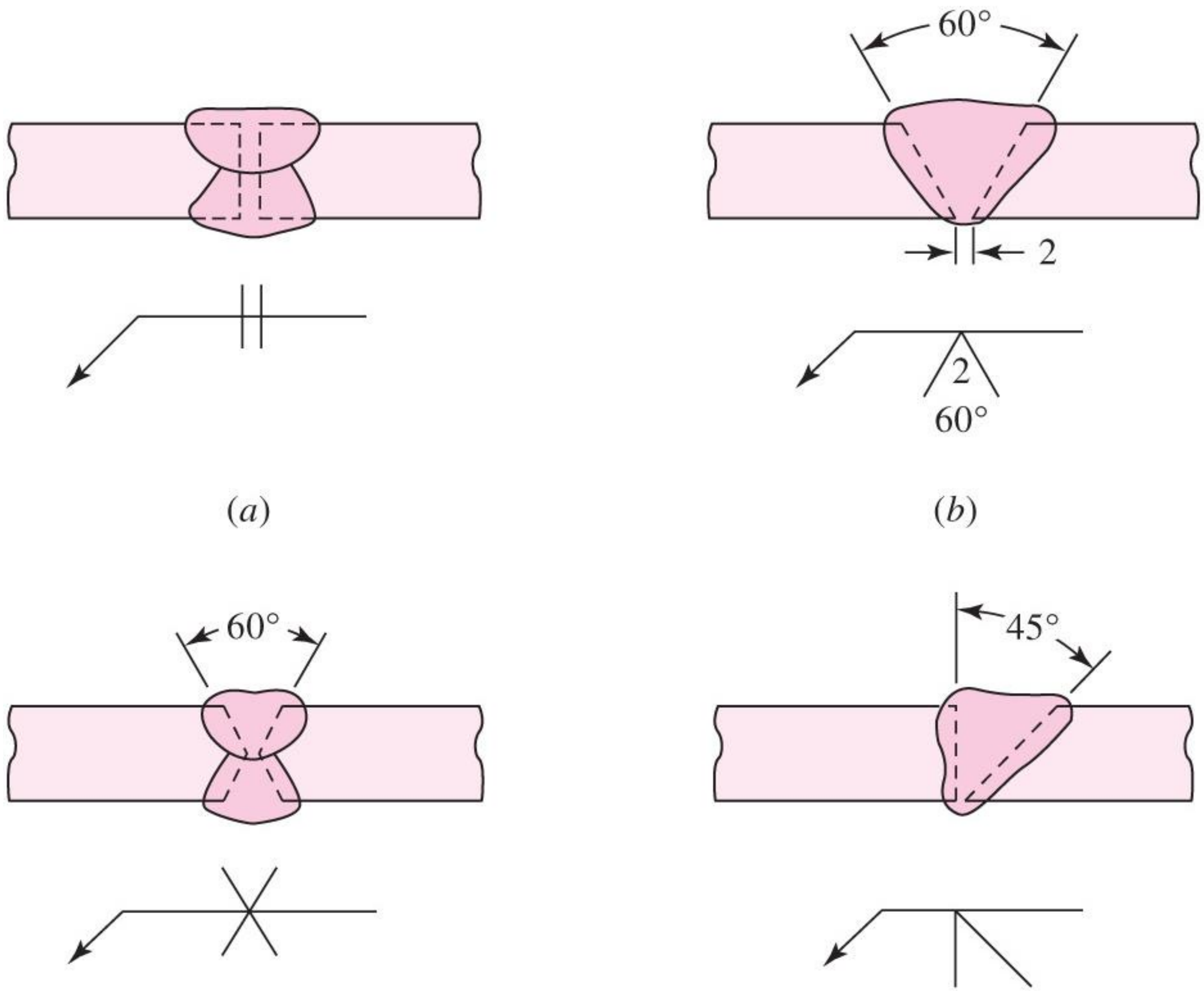
- Intermittent and staggered 60 mm along on 200 mm centers



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

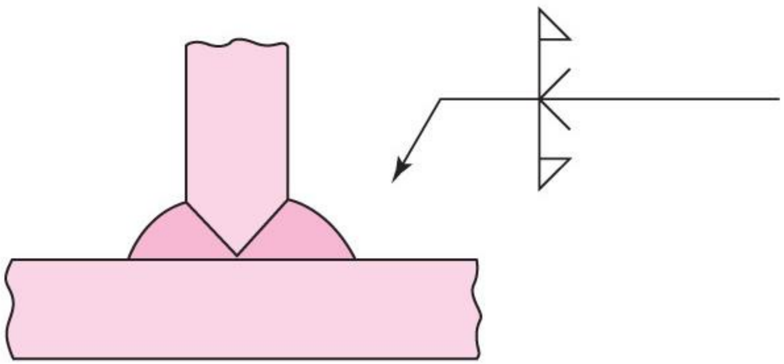


Symbols

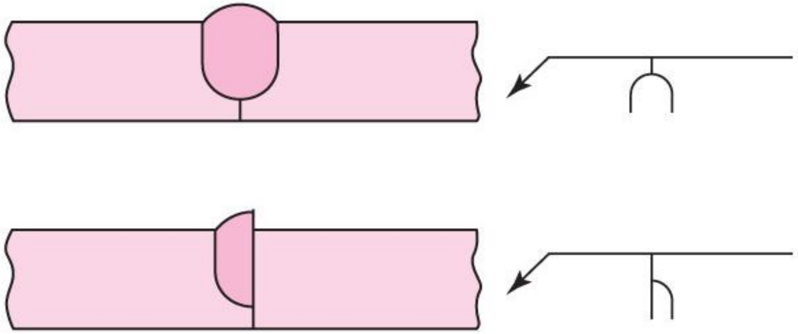


(a) (b) (c) (d)

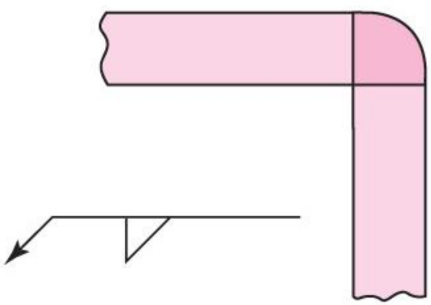
Symbols



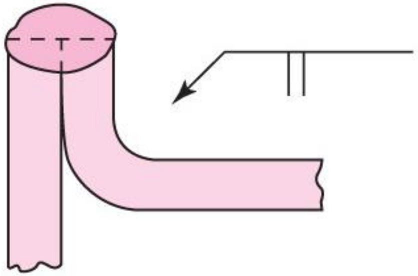
(a)



(b)



(c)

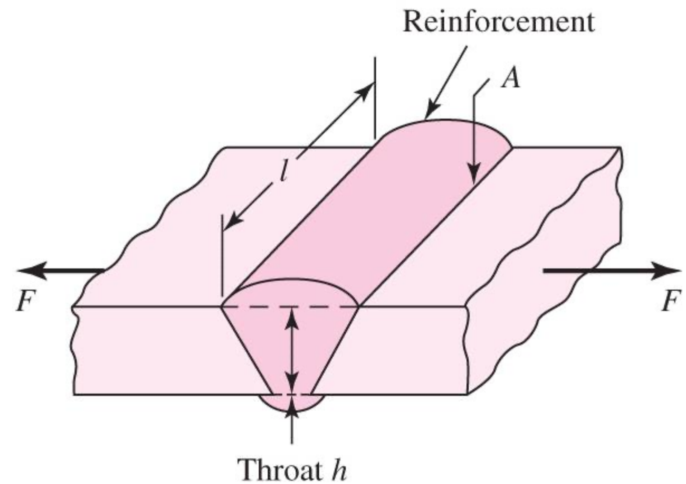


(d)

Tensile Butt Joint

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

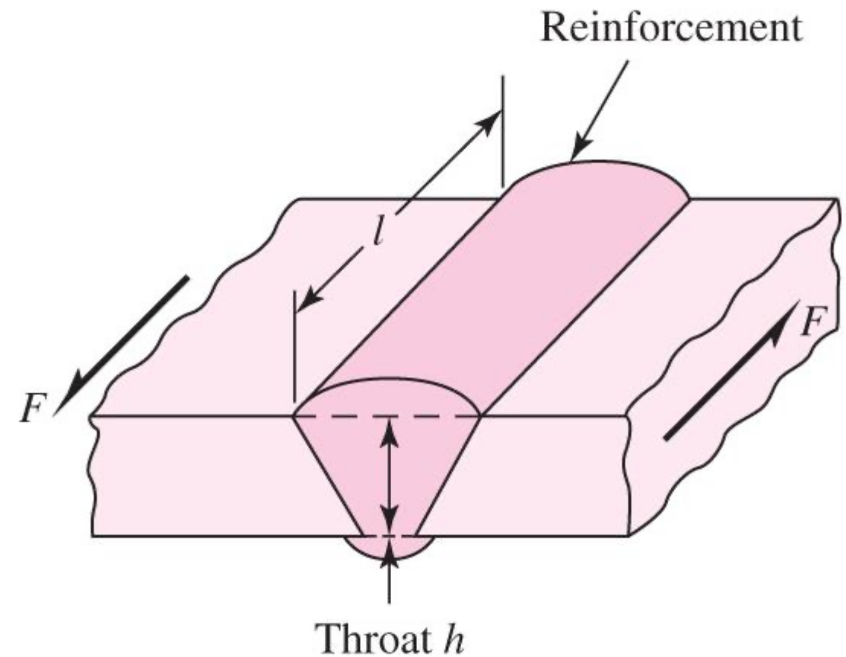
$$\sigma = \frac{F}{hl}$$



Shear Butt Joint

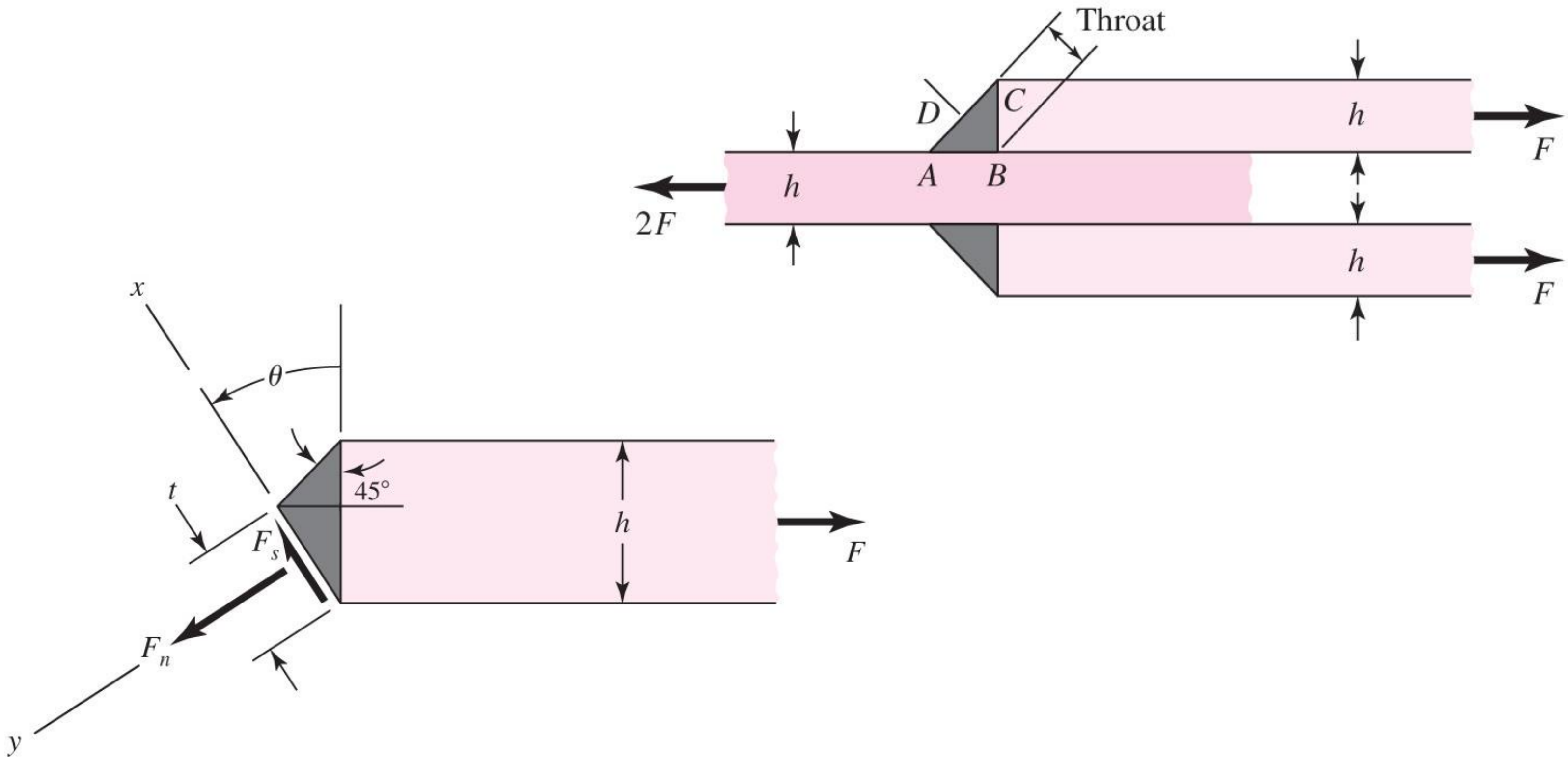
- Simple butt joint loaded in shear
- Average shear stress

$$\tau = \frac{F}{hl}$$



Transverse Fillet Weld

- Joint loaded in tension
- Weld loading is complex



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

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

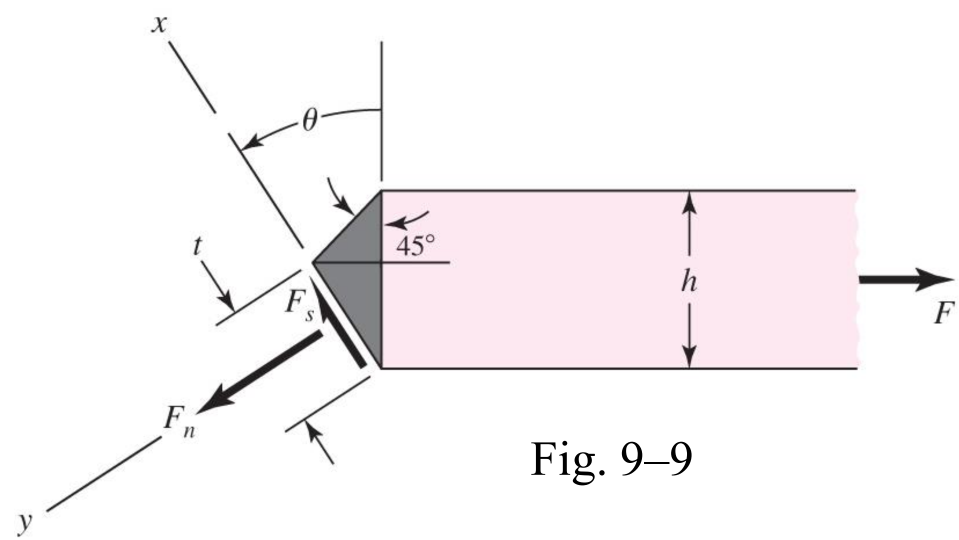


Fig. 9–9

- 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

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

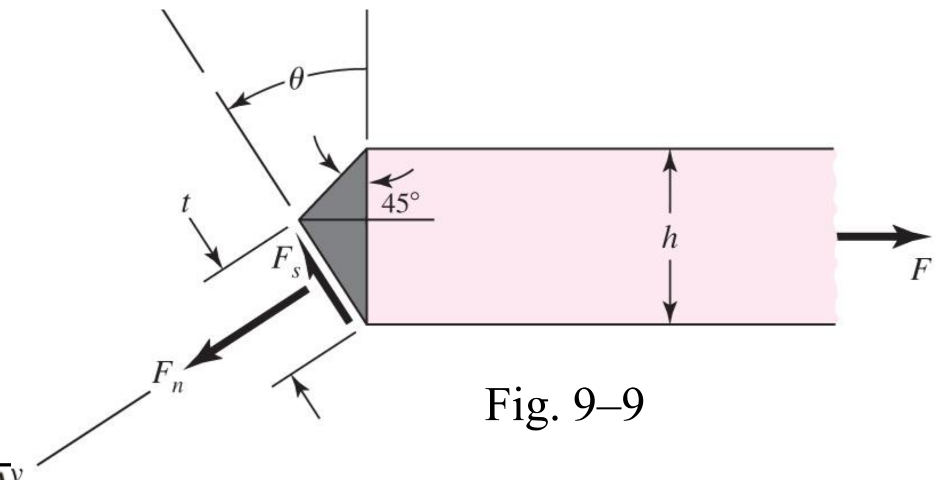


Fig. 9–9

Shigley's Mechanical Engineering Design

Fillet Weld Analysis

- 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

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

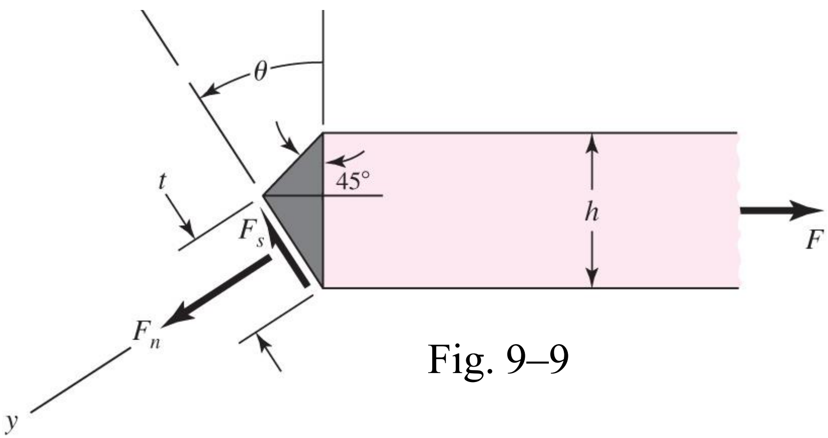


Fig. 9–9

Shigley's Mechanical Engineering Design

Fillet Weld Analysis

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

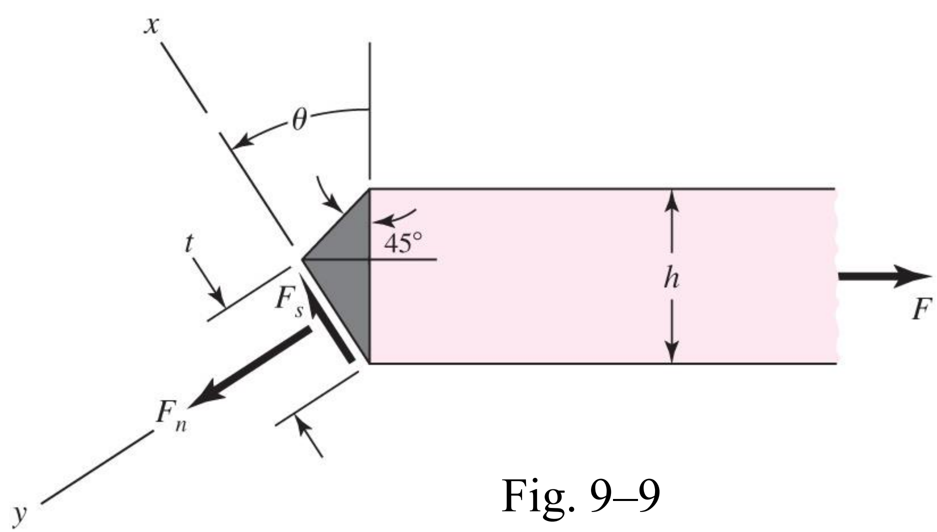


Fig. 9–9

Experimental Stress Distribution

- Experimental results are more complex

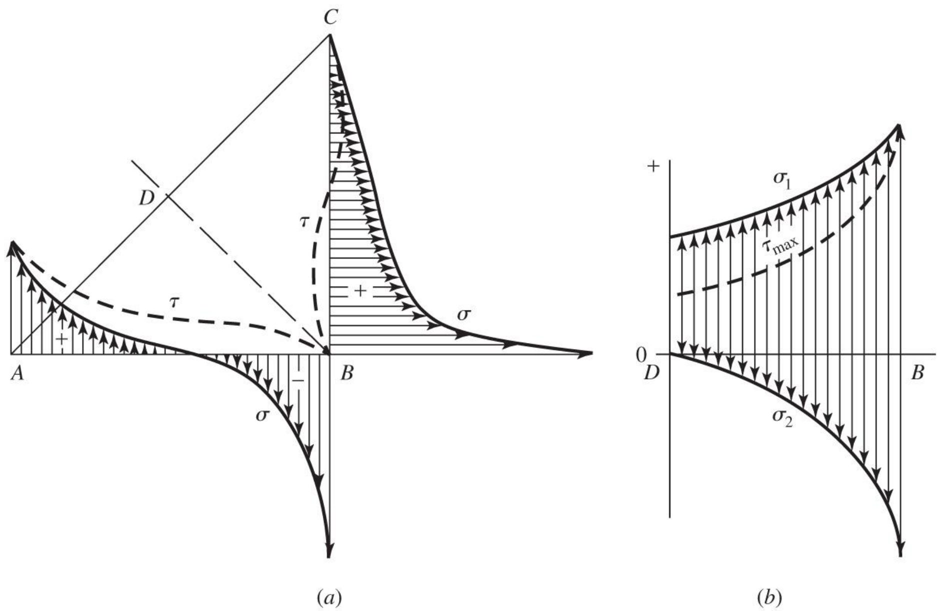


Fig. 9–10

Shigley's Mechanical Engineering Design

Simplified Analysis

- 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.

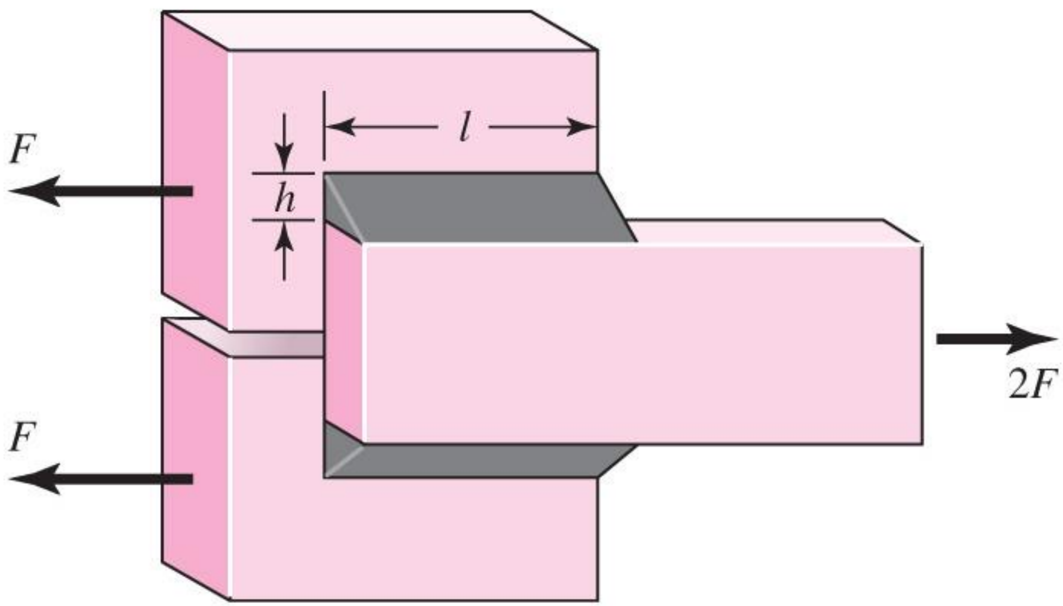
$$\tau = \frac{F}{0.707hl} = \frac{1.414F}{hl} \qquad (9\text{–}3)$$

- 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.

Simplified Analysis

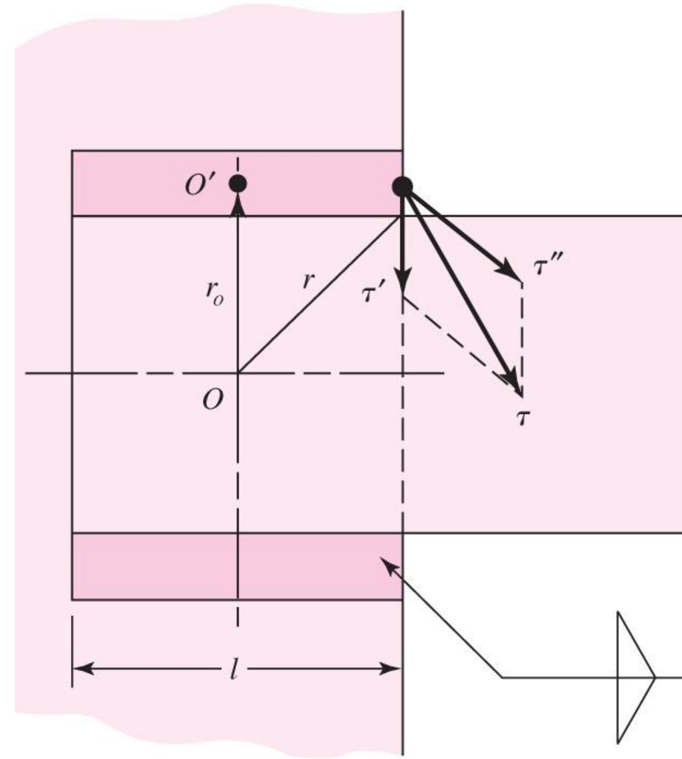
- Same equation also applies for simpler case of simple shear loading in fillet weld

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

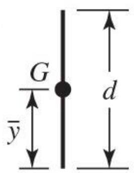
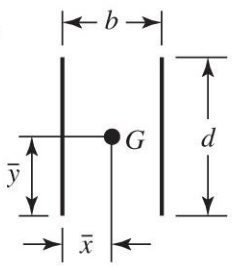
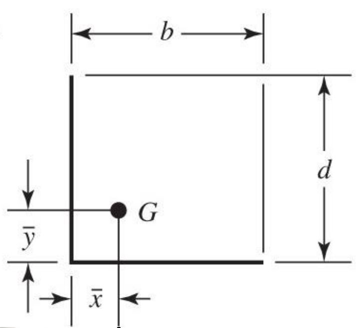


Fillet Welds Loaded in Torsion

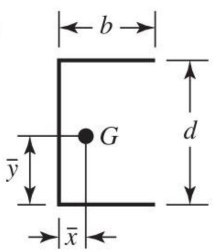
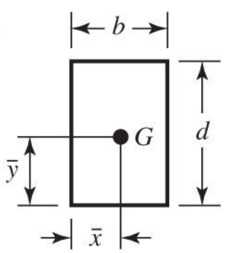
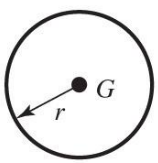
- Fillet welds carrying both direct shear V and moment M
- Primary shear $\tau' = \frac{V}{A}$
- Secondary shear $\tau'' = \frac{Mr}{J}$
- A is the throat area of all welds
- r is distance from centroid of weld group to point of interest
- J is second polar moment of area of weld group about centroid of group, G



Common Torsional Properties of Fillet Welds

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

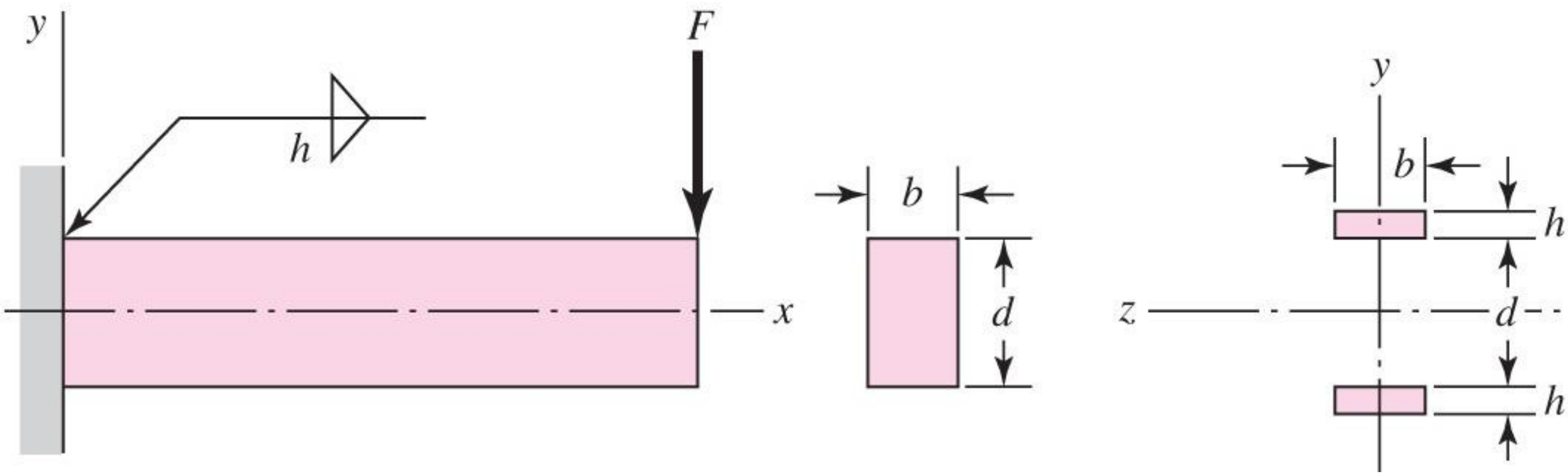
4.		$A = 0.707h(2b + d)$	$\bar{x} = \dfrac{b^2}{2b + d}$ $\bar{y} = d/2$	$J_u = \dfrac{8b^3 + 6bd^2 + d^3}{12} - \dfrac{b^4}{2b + d}$
5.		$A = 1.414h(b + d)$	$\bar{x} = b/2$ $\bar{y} = d/2$	$J_u = \dfrac{(b + d)^3}{6}$
6.		$A = 1.414\,\pi hr$		$J_u = 2\pi r^3$

*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.

These weld configurations can be analyzed much accurately by Finite Element Analysis

Fillet Welds Loaded in Bending

- Fillet welds carry both shear V and moment M

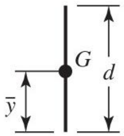
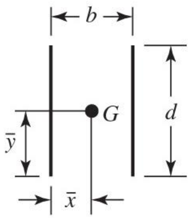
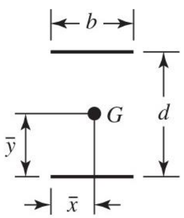
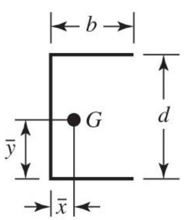


$$\tau' = \frac{V}{A} \qquad I_u = \frac{bd^2}{2} \qquad I = 0.707hI_u = 0.707h\frac{bd^2}{2}$$

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

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

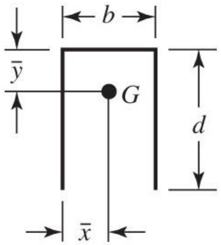
Bending Properties of Fillet Welds (Table 9–2)

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. 	$A = 0.707h(2b + d)$	$\bar{x} = \dfrac{b^2}{2b+d}$ $\bar{y} = d/2$	$I_u = \dfrac{d^2}{12}(6b + d)$

Shigley's Mechanical Engineering Design

Common Bending Properties of Fillet Welds

5\.

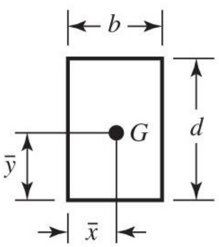


$$A = 0.707h(b + 2d)$$

$$\bar{x} = b/2 \qquad \bar{y} = \frac{d^2}{b + 2d}$$

$$I_u = \frac{2d^3}{3} - 2d^2\bar{y} + (b + 2d)\bar{y}^2$$

6\.

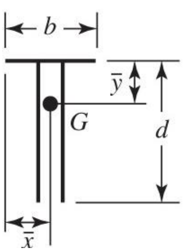


$$A = 1.414h(b + d)$$

$$\bar{x} = b/2 \qquad \bar{y} = d/2$$

$$I_u = \frac{d^2}{6}(3b + d)$$

7\.

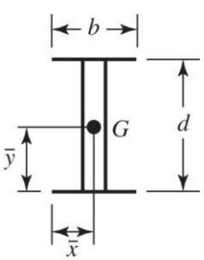


$$A = 0.707h(b + 2d)$$

$$\bar{x} = b/2 \qquad \bar{y} = \frac{d^2}{b + 2d}$$

$$I_u = \frac{2d^3}{3} - 2d^2\bar{y} + (b + 2d)\bar{y}^2$$

8\.

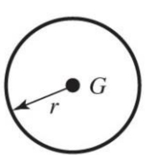


$$A = 1.414h(b + d)$$

$$\bar{x} = b/2 \qquad \bar{y} = d/2$$

$$I_u = \frac{d^2}{6}(3b + d)$$

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.

tresses Permitted by the AISC Code for Weld Metal

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	$0.60\text{–}0.66S_y$	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

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		$\frac{5}{8}$

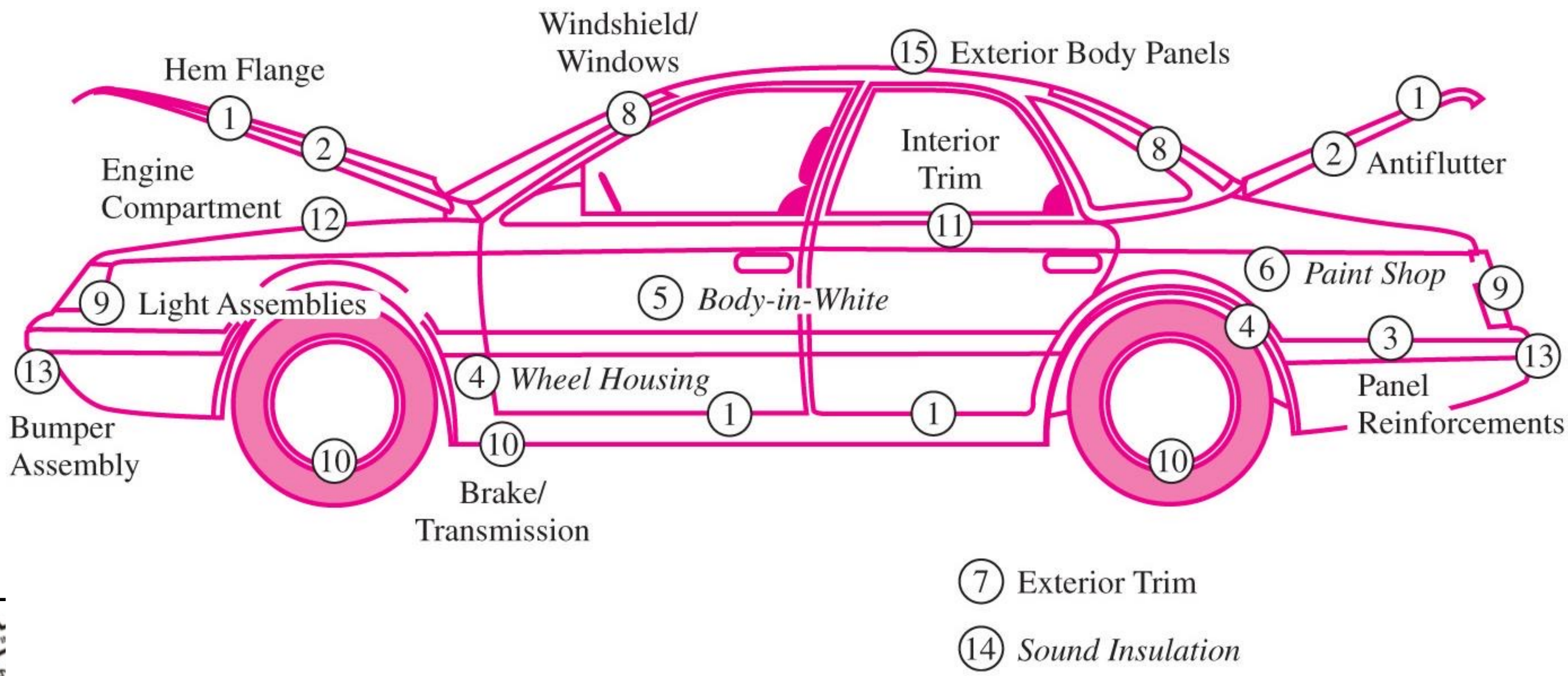
Not to exceed the thickness of the thinner part.

*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.

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



Types of Adhesives

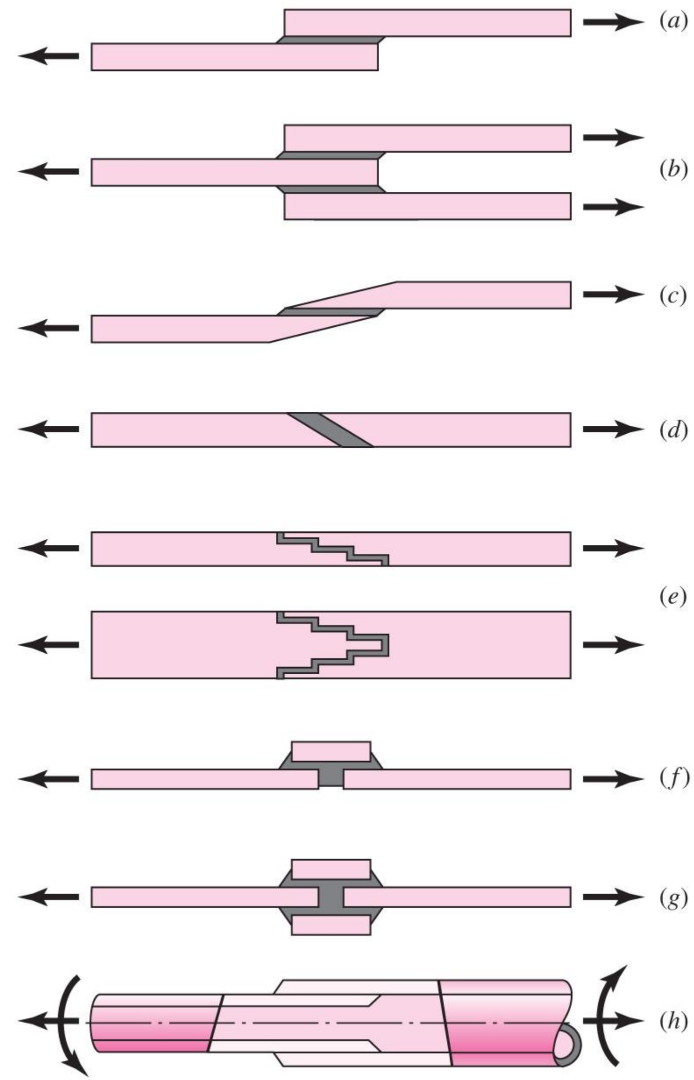
- May be classified by
  - Chemistry
    - Epoxies, polyurethanes, polyimides
  - Form
    - Paste, liquid, film, pellets, tape
  - Type
    - Hot melt, reactive hot melt, thermosetting, pressure sensitive, contact
  - Load-carrying capability
    - Structural, semi-structural, non-structural

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)

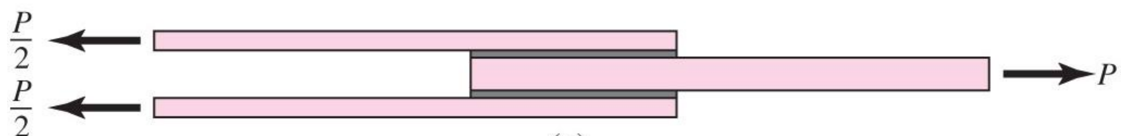
Stress Distribution

- 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

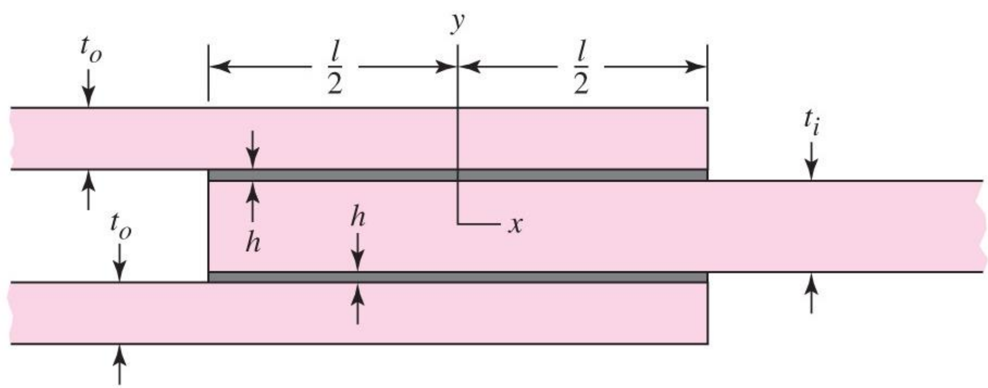


Double Lap Joint

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



(a)



(b)

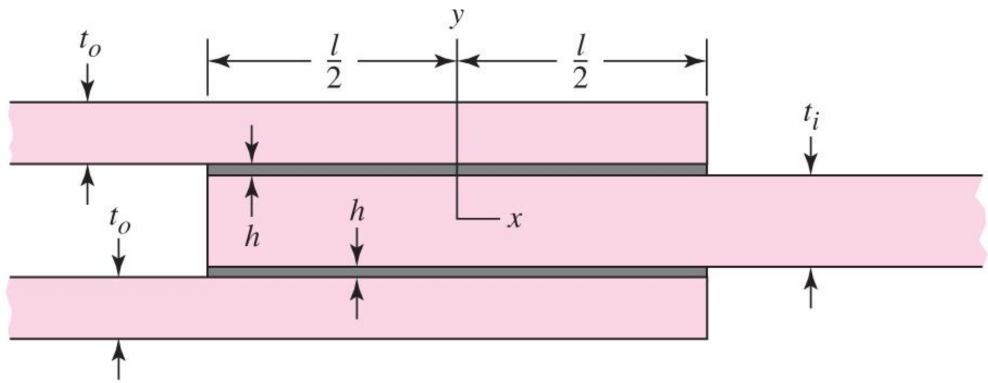
Double Lap Joint

- 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

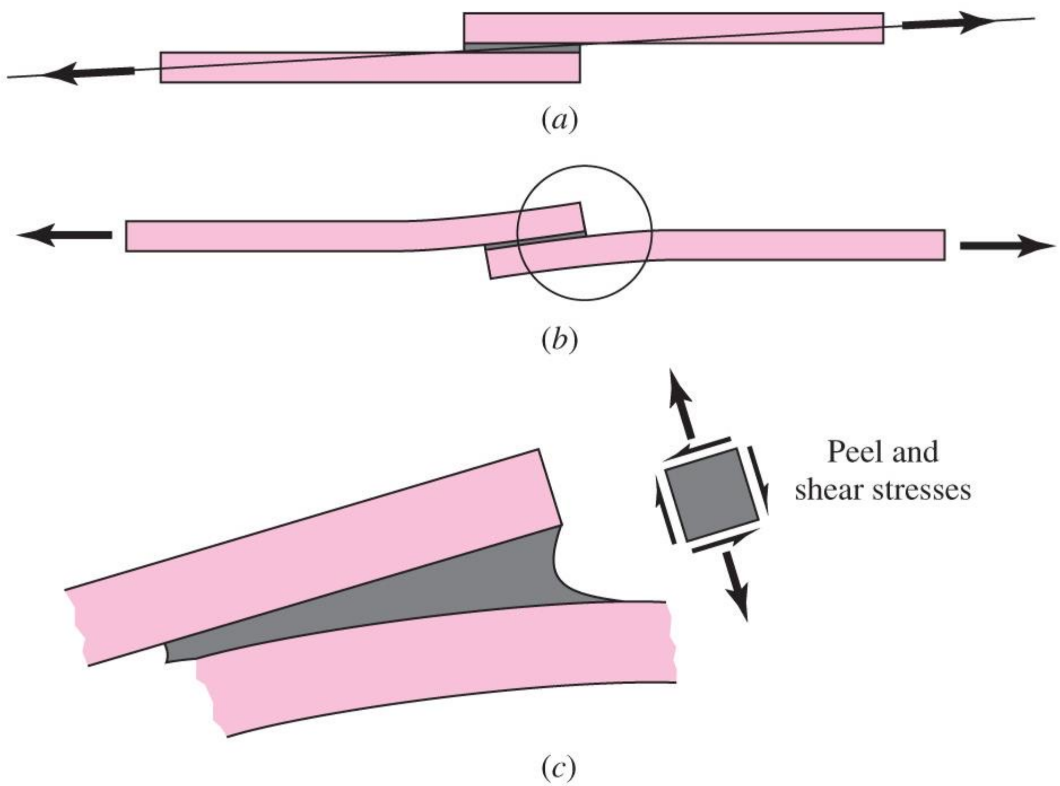
$$\omega = \sqrt{\frac{G}{h}\left(\frac{1}{E_o t_o} + \frac{2}{E_i t_i}\right)}$$



(b)

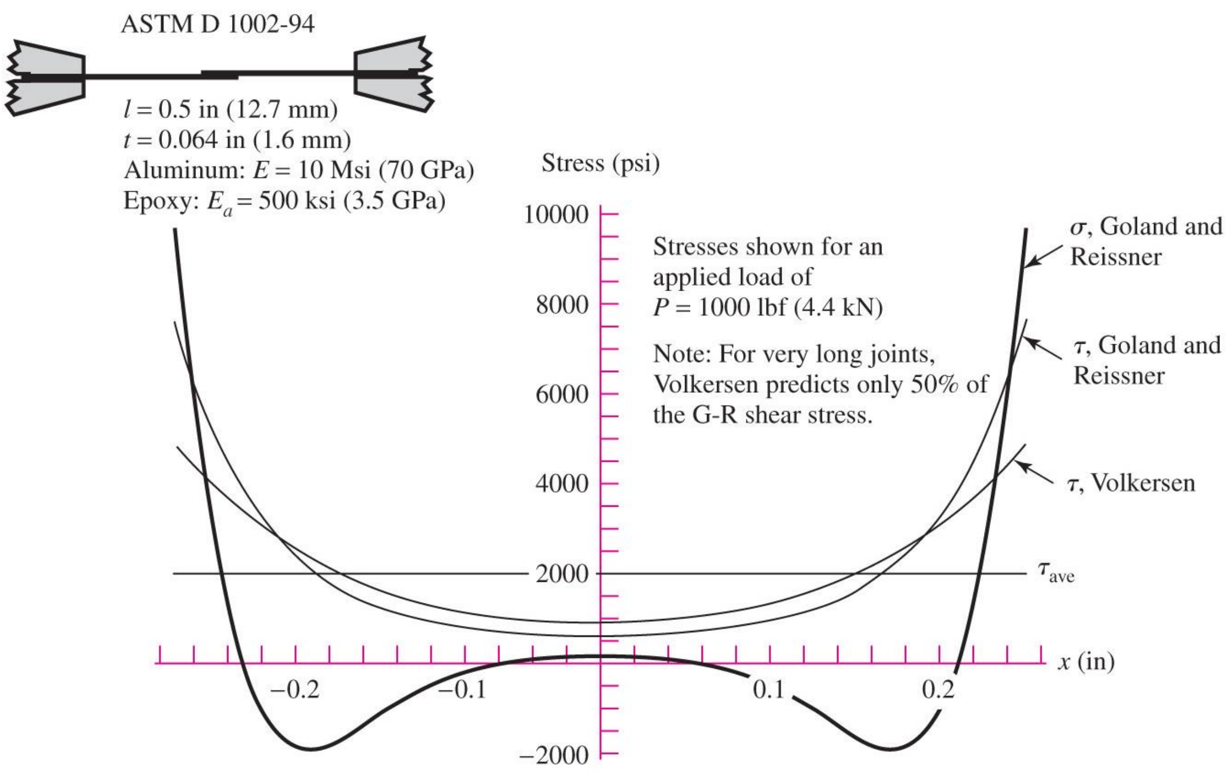
Single-lap Joint

- 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



Single-lap Joint

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

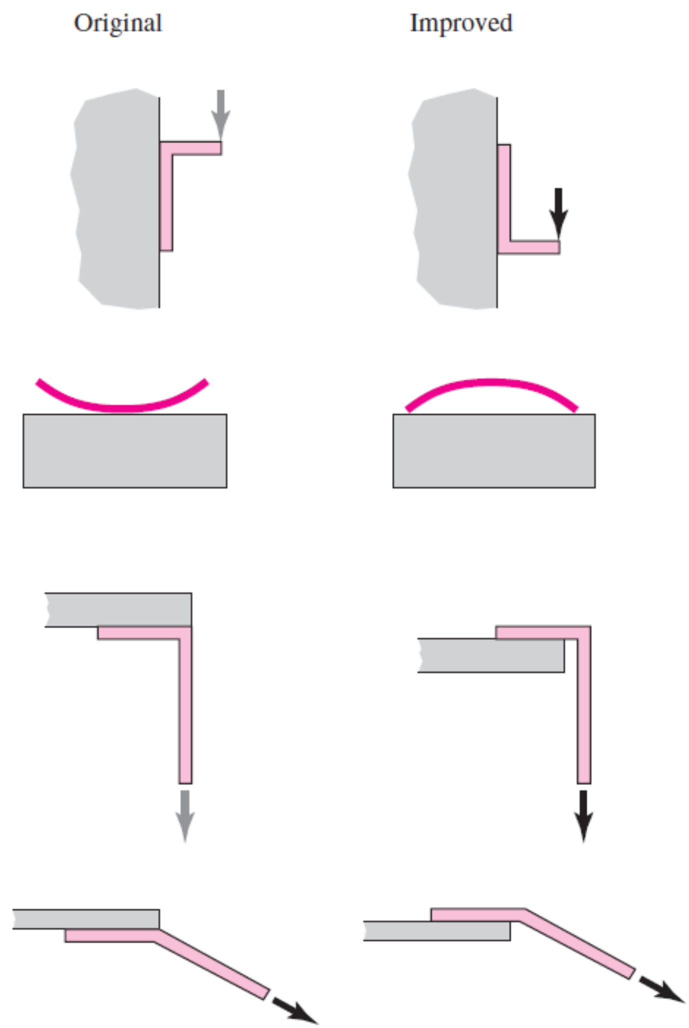


(d)

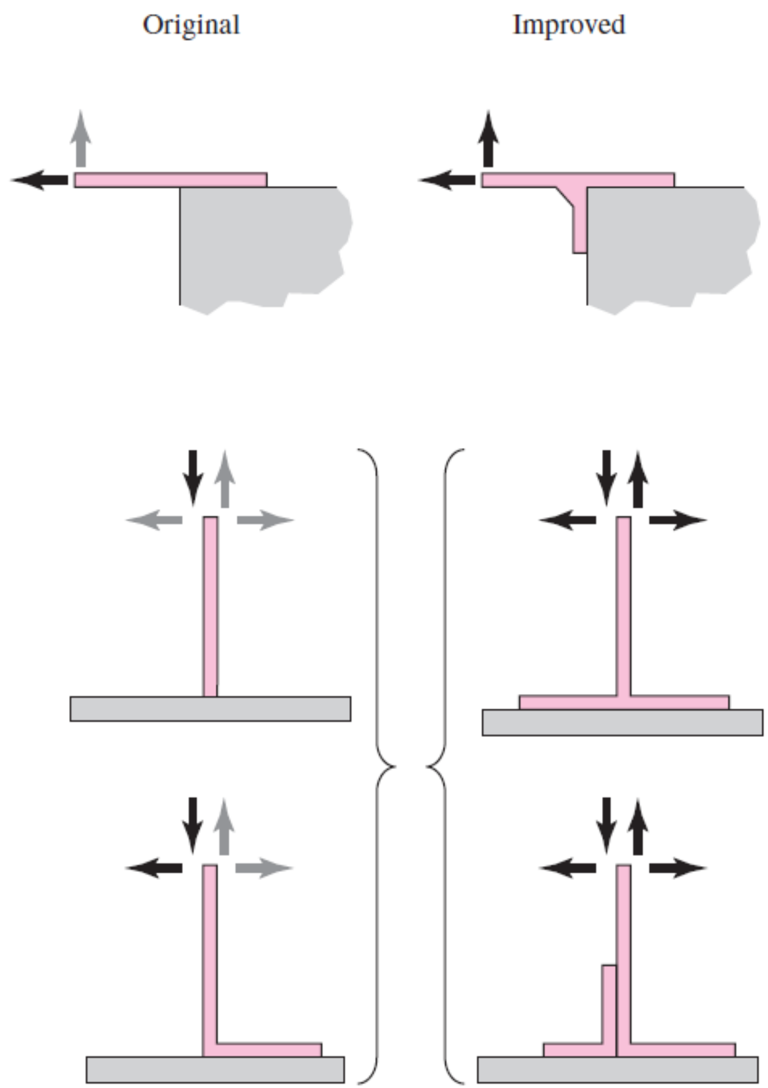
Adhesive Joint Design Guidelines

- 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.

Improved Bonding Designs



Improved Bonding Designs



ME 423: Machine Design
Instructor: Ramesh Singh

Improved Bonding Designs