ME303 Introduction to Mechanical Design

Lecture 12
Permanent Joints
(Welding, Bonding)

Song Chaoyang

Assistant Professor

Department of Mechanical and Energy Engineering

songcy@sustech.edu.cn

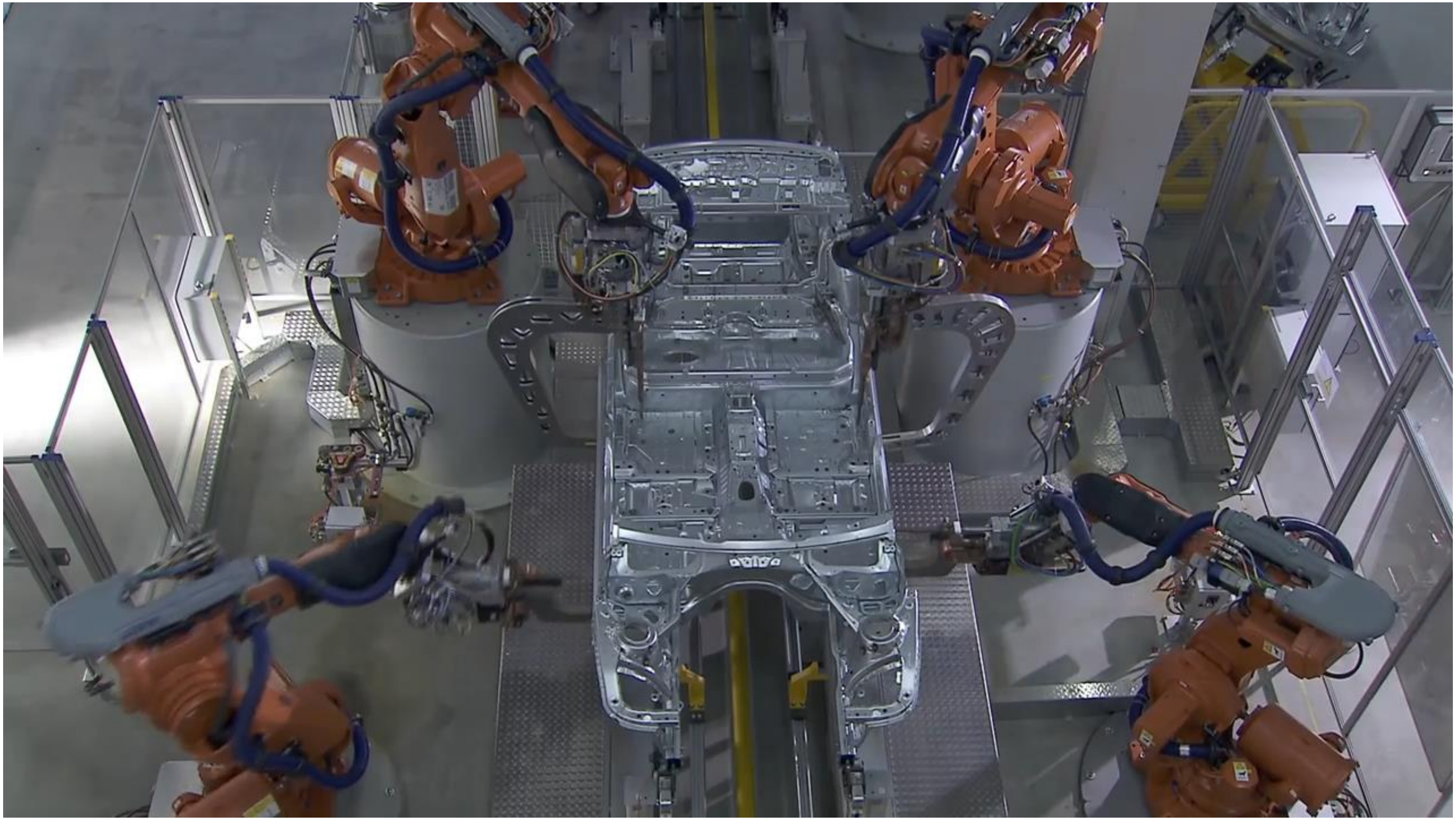
Agenda

Week 11, Wednesday, Nov 20, 2019

- Welding Symbols
- Butt and Fillet Welds
- Stresses in Welded Joints in Torsion | Bending
- The Strength of Welded Joints
- Static Loading
- Fatigue Loading
- Resistant Welding
- Adhesive Welding
- A Gentle Reminder of the Design Projects

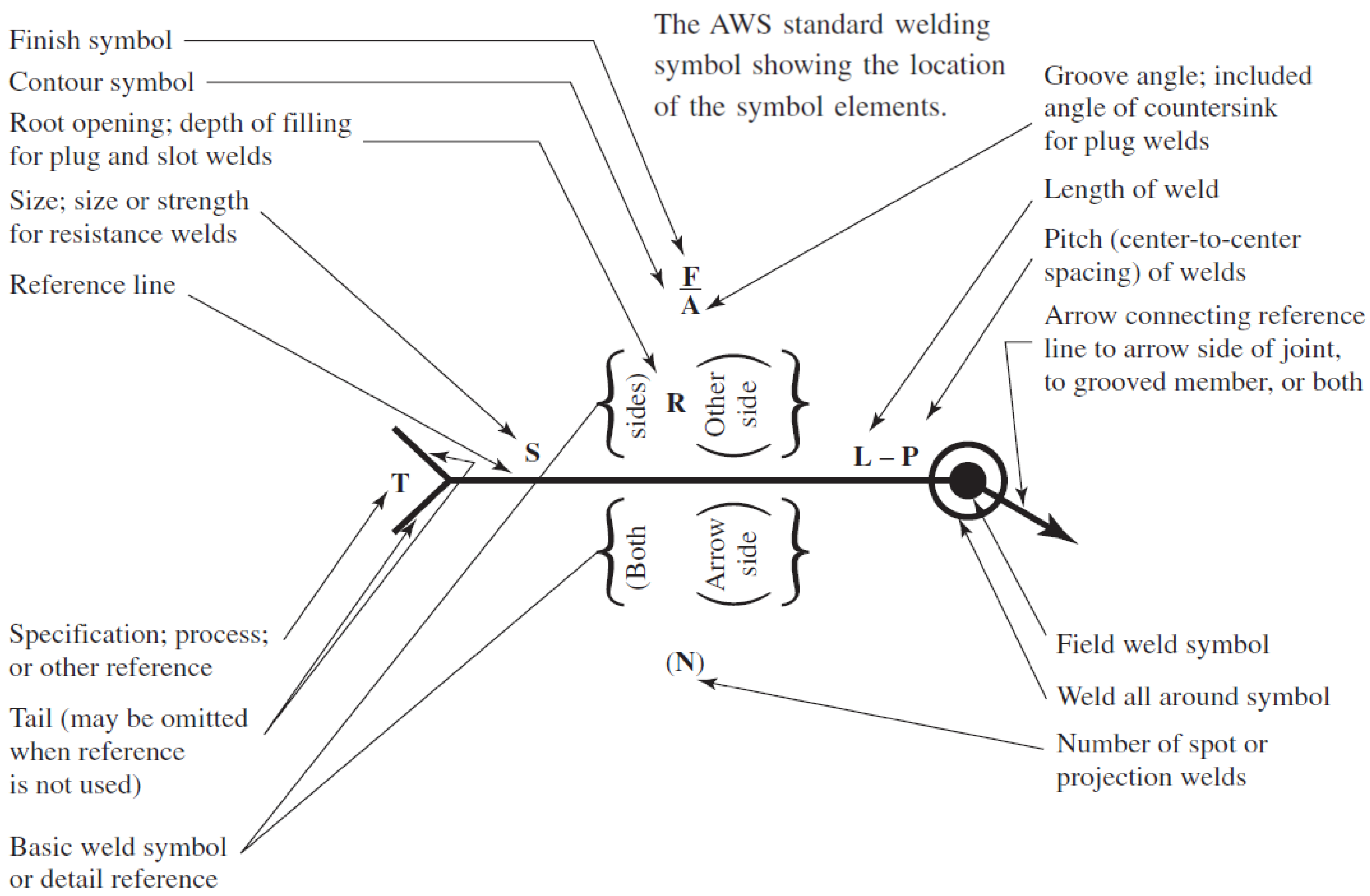
BMW F30 3 Series Production Process - Body Shop

https://www.youtube.com/watch?v=muLmD_7DANw



Welding Symbols

A weldment is fabricated by welding together a collection of metal shapes, cut to particular configurations.



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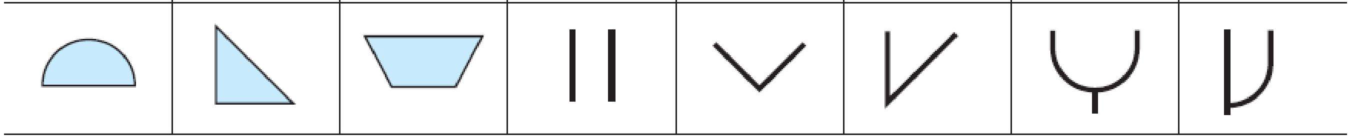
Commonly Welding Types and Symbols

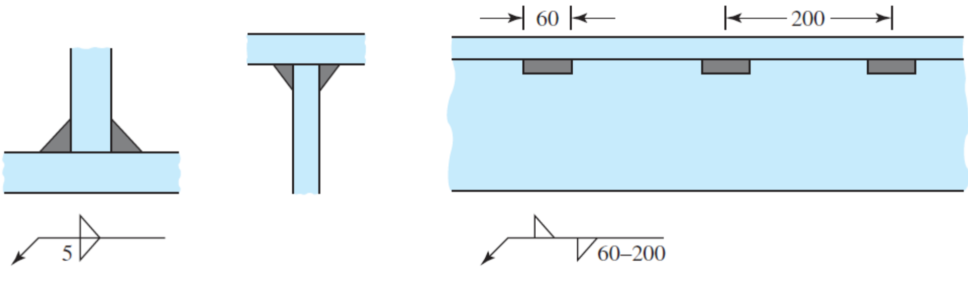
For general machine elements most welds are fillet welds, though butt welds are used a great deal in designing pressure vessels.

Arc- and gas-weld symbols.

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

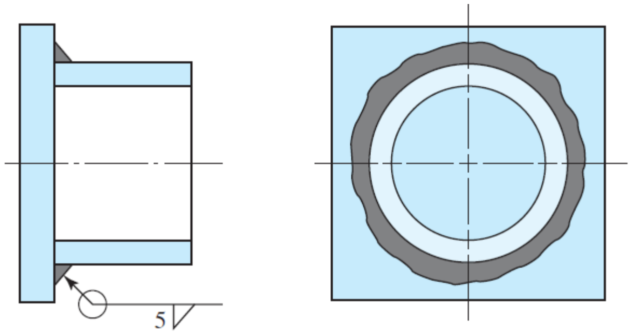


Fillet welds. (*a*) The number indicates the leg size; the arrow should point only to one weld when both sides are the same. (*b*) The symbol indicates that the welds are intermittent and staggered 60 mm along on 200-mm centers.



(*a*) (*b*)

The circle on the weld symbol indicates that the welding is to go all around.



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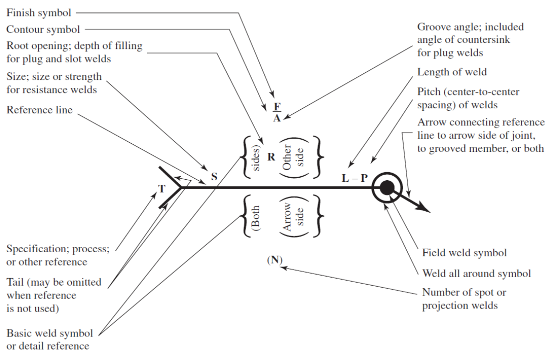
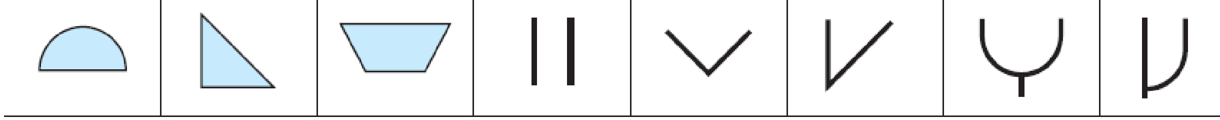
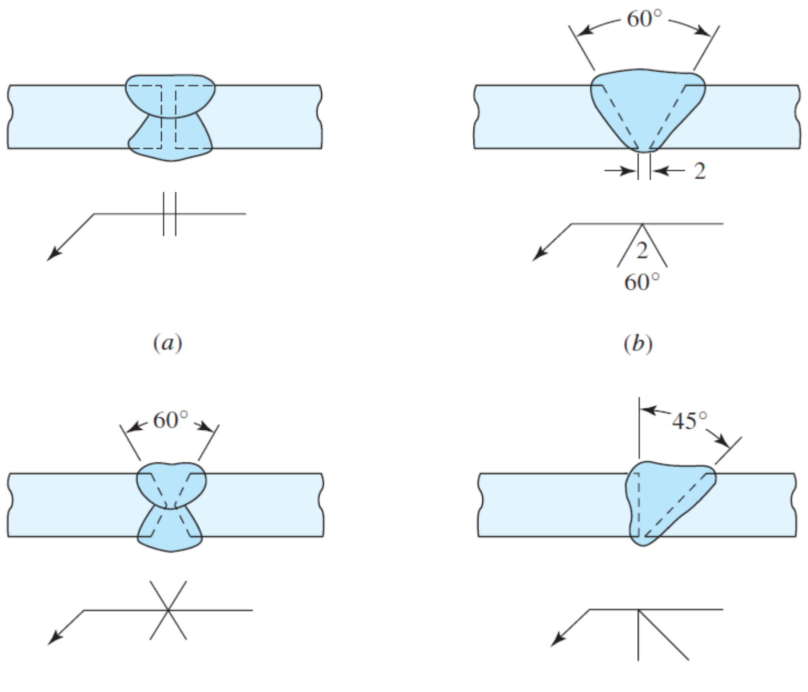


Type of weld							
Bead	Fillet	Plug or slot	Groove				
			Square	V	Bevel	U	J

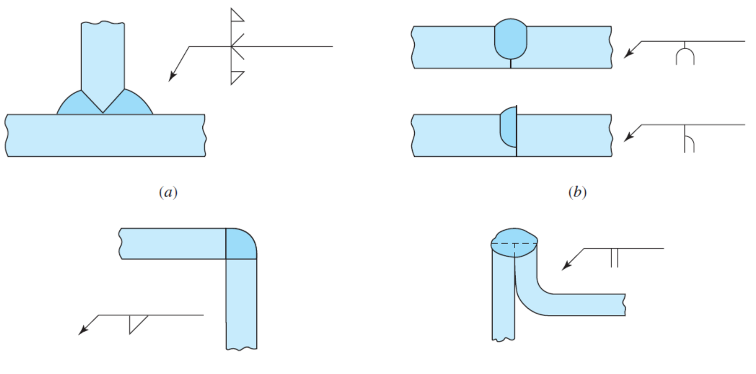




(*a*) (*b*) (*c*) (*d*)

Butt or groove welds: (*a*) square butt-welded on both sides; (*b*) single V with 60° bevel and root opening of 2 mm; (*c*) double V; (*d*) single bevel.

Special groove welds: (*a*) T joint for thick plates; (*b*) U and J welds for thick plates; (*c*) corner weld (may also have a bead weld on inside for greater strength but should not be used for heavy loads); (*d*) edge weld for sheet metal and light loads.



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

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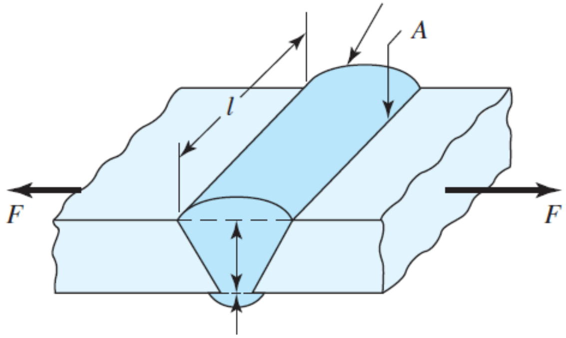
Butt Welds

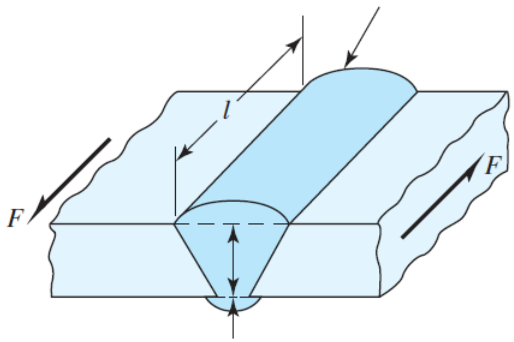
A single V-groove weld loaded by the tensile force F.

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



(a) Tensile loading

the average normal/shear stress



(b) Shear loading

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

- The reinforcement can be desirable, but it varies somewhat and does produce stress concentration at point A in the figure.
- If fatigue loads exist, it is good practice to grind or machine off the reinforcement.

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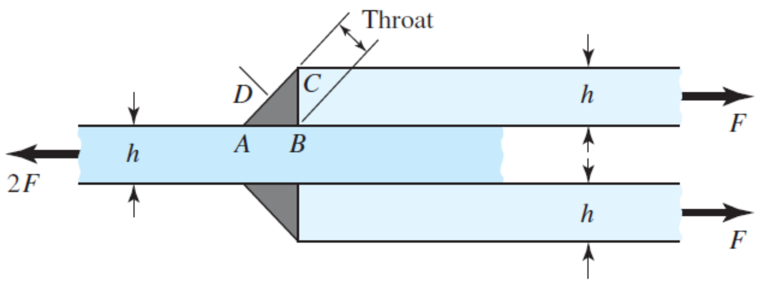
Transverse Fillet Weld



The nominal stresses at the angle θ in the weldment, τ and σ, are

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

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

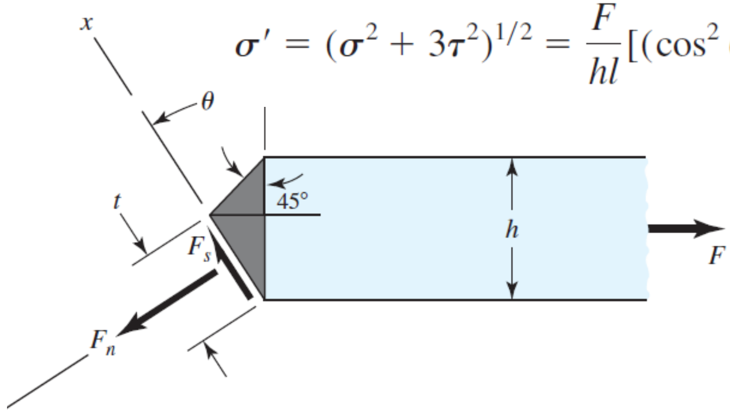
The von Mises stress σ' at angle θ is

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

$$F_s = F\sin\theta$$

$$F_n = F\cos\theta$$

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



The largest von Mises stress occurs at $\theta = 62.5°$ with a value of $\sigma' = 2.16F/(hl)$. The corresponding values of τ and σ are $\tau = 1.196F/(hl)$ and $\sigma = 0.623F/(hl)$.

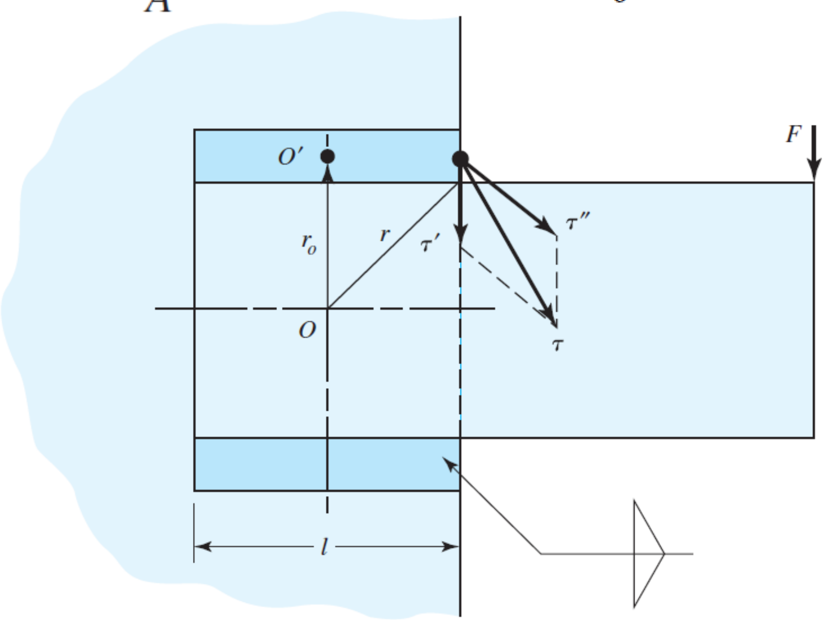
The maximum shear stress can be found by differentiating Eq. (d) with respect to θ and equating to zero. The stationary point occurs at $\theta = 67.5°$ with a corresponding $\tau_{max} = 1.207F/(hl)$ and $\sigma = 0.5F/(hl)$.

Stresses in Welded Joints in Torsion

A cantilever welded to a column by two fillet welds each of length *l*

- The reaction at the support of a cantilever always consists of a shear force *V* and a moment *M*
  - *Primary shear* *Secondary shear*

$$\tau' = \frac{V}{A} \qquad \tau'' = \frac{Mr}{J}$$



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

- Depends on whether the size of the welds are known for the equations to be solved.

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Example of Two Welds in a Group

Throat thickness

$$t_1 = 0.707h_1 \qquad t_2 = 0.707h_2$$

The throat area of both welds together

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

Second moment of area

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

Second polar moment of area of welds 1 & 2 about its own centroid

$$J_{G1} = I_x + I_y = \frac{t_1 d^3}{12} + \frac{d t_1^3}{12} \qquad J_{G2} = \frac{b t_2^3}{12} + \frac{t_2 b^3}{12}$$

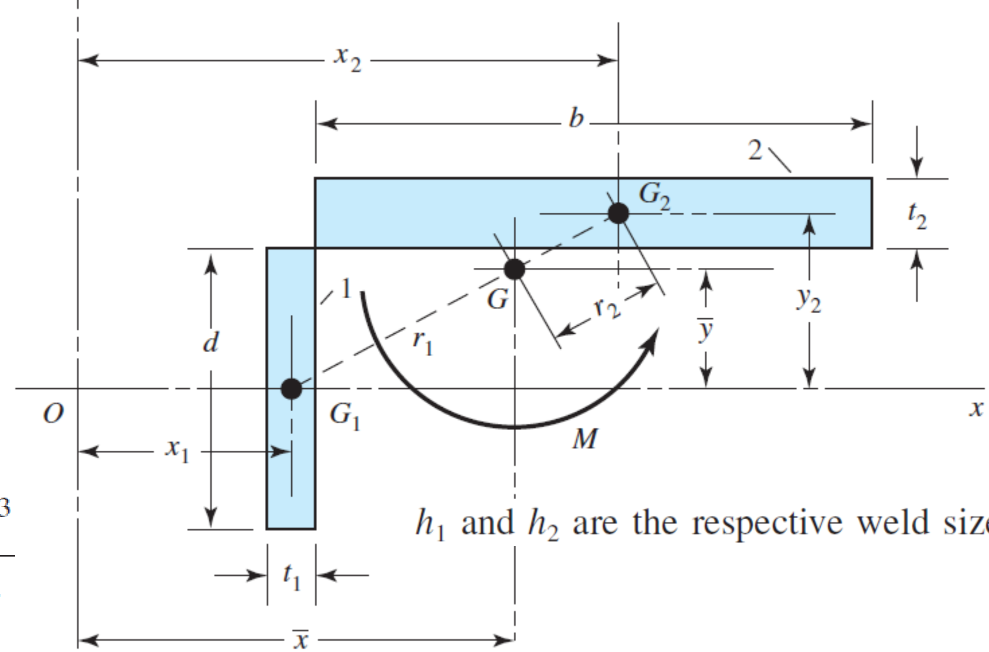
Coordinate of the centroid G of weld group

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

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

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

The rectangles represent the throat areas of the welds



h_1 and h_2 are the respective weld sizes.

Second moment of area of the weld group

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

J_u is found by conventional methods for an area having unit width.

$$J = 0.707 h J_u$$

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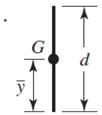
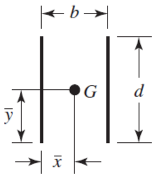
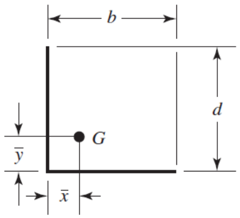
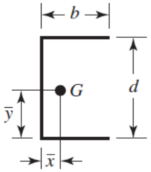
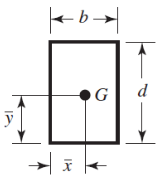
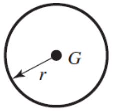
Torsional Properties of Fillet Welds

The throat areas and the unit second polar moments of area for the most common fillet welds encountered

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

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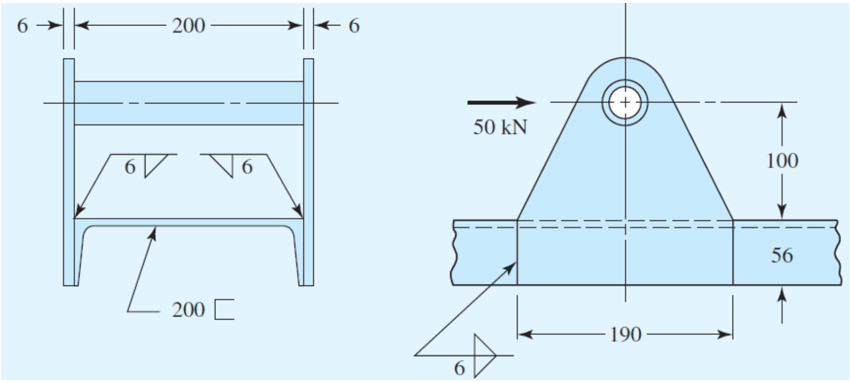
Example: A 50-kN load is transferred from a welded fitting into a 200-mm steel channel.

Estimate the maximum stress in the weld.

(*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.



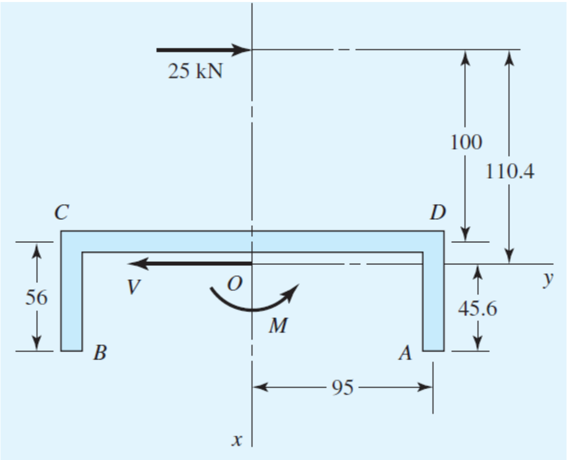
(*b*) Estimate the primary shear stress τ'. As shown in Fig. 9–14, each plate is welded to the channel by means of three 6-mm fillet welds. Figure 9–15 shows that we have divided the load in half and are considering only a single plate. From case 4 of Table 9–1 we find the throat area as

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

Then the primary shear stress is

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

(*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

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

(*e*) Find the distances r_i (see Fig. 9–16):

$$r_A = r_B = [(190/2)^2 + (56 - 10.4)^2]^{1/2} = 105 \text{ mm}$$
$$r_C = r_D = [(190/2)^2 + (10.4)^2]^{1/2} = 95.6 \text{ mm}$$

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(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:

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

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

(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.

(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

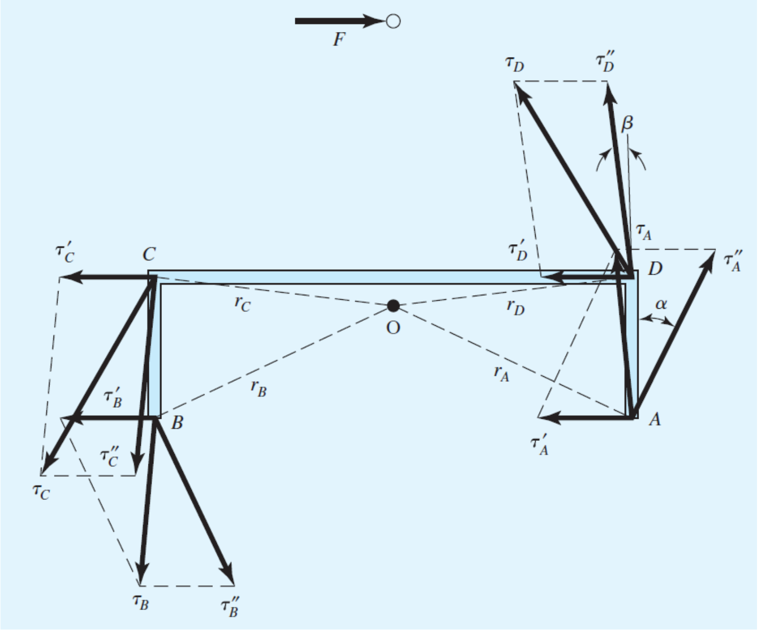
$$\tau_A = \tau_B = \sqrt{(19.5 - 41.0 \sin 25.64°)^2 + (41.0 \cos 25.64°)^2} = 37.0 \text{ MPa}$$

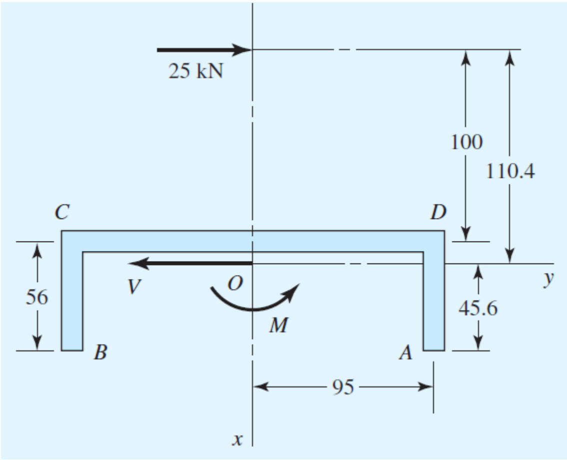
Similarly, for C and D, $\beta = \tan^{-1}(10.4/95) = 6.25°$. Thus

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

(k) Identify the most highly stressed point:

$$\tau_{max} = \tau_C = \tau_D = 43.9 \text{ MPa}$$





Stresses in Welded Joints in Bending

Similar Process as in Torsion

Vertical (primary) shear

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

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

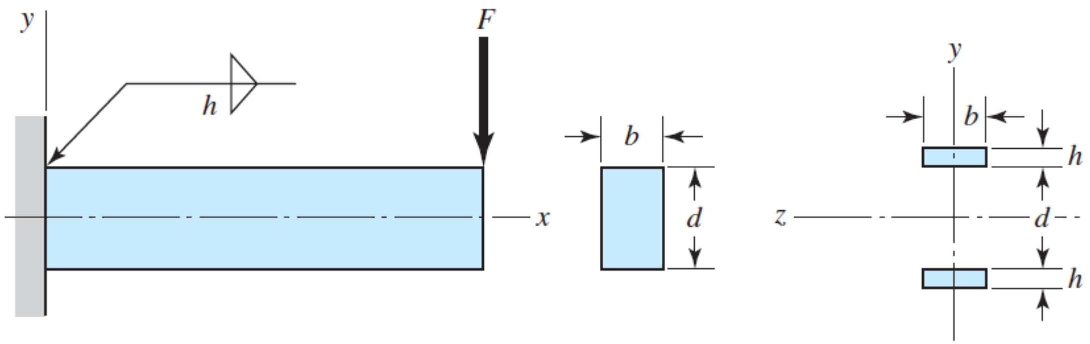
Horizontal (secondary) shear

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

$$I_u = \frac{bd^2}{2}$$

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

A cantilever welded to a support by fillet welds at top and bottom

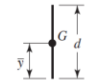
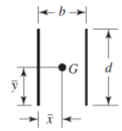
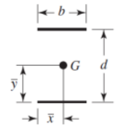
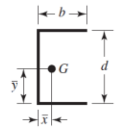
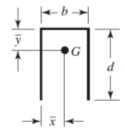
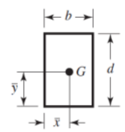
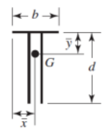
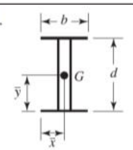
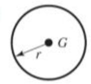


(a) (b) Weld pattern

- Treating welds as a line
- Produce a slightly larger second moment of area, and result in a smaller level of stress.
- Convenient for estimation

Bending Properties of Fillet Welds

Similar process for convenient calculated estimation.

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. 	$A = 0.707h(2b + d)$	$\bar{x} = \dfrac{b^2}{2b + d}$ $\bar{y} = d/2$	$I_u = \dfrac{d^2}{12}(6b + d)$
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. 	$A = 1.414\pi hr$		$I_u = \pi r^3$

*I_u, unit second moment of area, is taken about a horizontal axis through G, the centroid of the weld group, h is weld size; the plane of the bending couple is normal to the plane of the paper and parallel to the y-axis; all welds are of the same size.

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The Strength of Welded Joints

Under the proper conditions, all steels can be welded.

- The matching of the electrode properties with those of the parent metal is usually not so important as speed, operator appeal, and the appearance of the completed joint.
- It is preferable, in designing welded components, to select a steel that will result in
  - **a fast, economical weld** even though this may require a sacrifice of other qualities such as machinability.

Fatigue Stress-Concentration Factors

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

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.

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Schedule A: Allowable Load for Various Sizes of Fillet Welds

	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.

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

Schedule B: Minimum Fillet Weld Size, h

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.

Source: From Omer W. Blodgett (ed.), *Stress Allowables Affect Weldment Design,* D412, The James F. Lincoln Arc Welding Foundation, Cleveland, May 1991, p. 3. Reprinted by permission of Lincoln Electric Company.

Static Loading Example

A $\frac{1}{2}$-in by 2-in rectangular-cross-section UNS G10150 HR 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?

Schedule A: Allowable Load for Various Sizes of Fillet Welds

	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						
τ =	18.0	21.0	24.0	27.0	30.0	33.0	36.0
	Allowable Unit Force on Fillet Weld, kip/linear in						
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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
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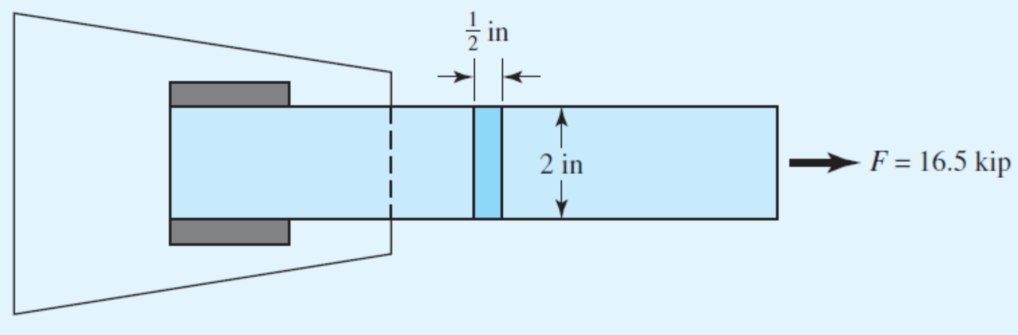
5/8	7.96	9.28	10.61	11.93	13.27	14.58	15.91
1/2	6.37	7.42	8.48	9.54	10.61	11.67	12.73
7/16	5.57	6.50	7.42	8.35	9.28	10.21	11.14
3/8	4.77	5.57	6.36	7.16	7.95	8.75	9.54
5/16	3.98	4.64	5.30	5.97	6.63	7.29	7.95
1/4	3.18	3.71	4.24	4.77	5.30	5.83	6.36
3/16	2.39	2.78	3.18	3.58	3.98	4.38	4.77
1/8	1.59	1.86	2.12	2.39	2.65	2.92	3.18
1/16	0.795	0.930	1.06	1.19	1.33	1.46	1.59

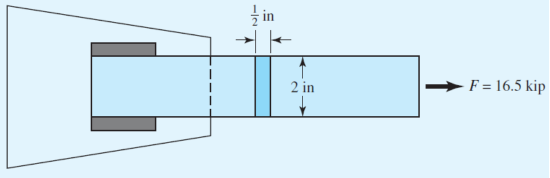


$$F = 5.57l = 5.57(4) = 22.28 \text{ kip}$$

Since $22.28 > 16.5$ kip,

weld metal strength is satisfactory.

Static Loading Example



A $\frac{1}{2}$-in by 2-in rectangular-cross-section UNS G10150 HR 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. (*b*) Is the attachment strength satisfactory?

Table A-20

1 UNS No.	2 SAE and/or AISI No.	3 Processing	4 Tensile Strength, MPa (kpsi)	5 Yield Strength, MPa (kpsi)	6 Elongation in 2 in, %	7 Reduction in Area, %	8 Brinell Hardness
G10060	1006	HR	300 (43)	170 (24)	30	55	86
		CD	330 (48)	280 (41)	20	45	95
G10100	1010	HR	320 (47)	180 (26)	28	50	95
		CD	370 (53)	300 (44)	20	40	105
G10150	1015	HR	340 (50)	190 (27.5)	28	50	101
		CD	390 (56)	320 (47)	18	40	111
G10180	1018	HR	400 (58)	220 (32)	25	50	116
		CD	440 (64)	370 (54)	15	40	126

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.

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

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

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

Since $\tau_{\text{all}} \geq \tau$, the attachment is satisfactory near the weld beads.

The tensile stress in the shank of the attachment σ is

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

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

Since $\sigma \leq \sigma_{\text{all}}$, the shank tensile stress is satisfactory.

Fatigue Loading Example

The AISI 1018 HR 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.

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. 296, $k_a = 39.9(58)^{-0.995} = 0.702$. 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. 298, for torsion (shear),

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

From Eqs. (6–8), p. 290, and (6–18), p. 295,

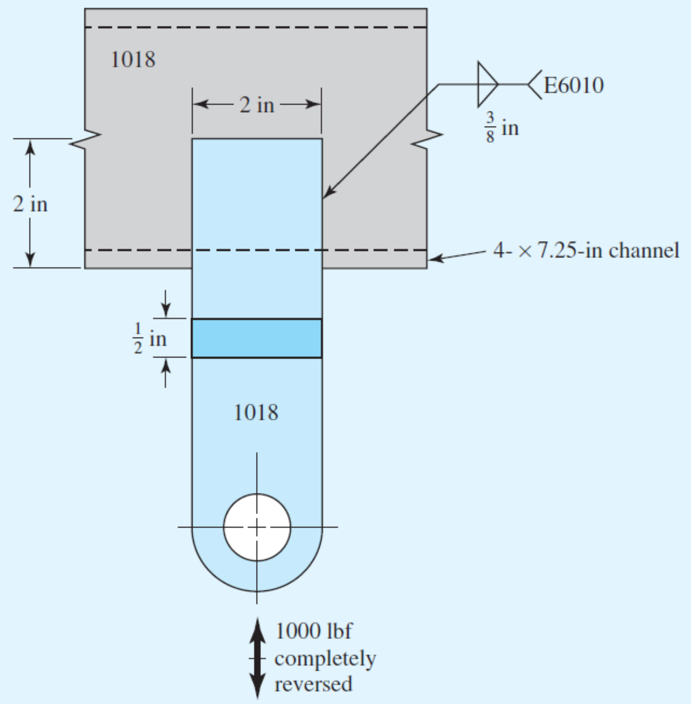
$$S_{se} = 0.702(1)0.59(1)(1)(1)0.5(58) = 12.0\ \text{kpsi}$$

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

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



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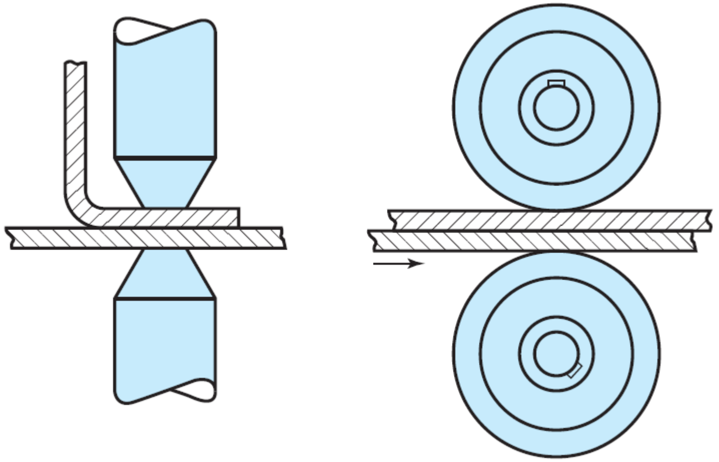
Resistance Welding

The heating and consequent welding that occur when an electric current is passed through several parts that are pressed together.

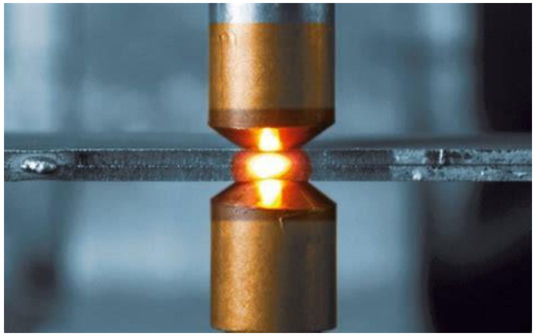
- Advantages
  - Speed,
  - the accurate regulation of time and heat,
  - the uniformity of the weld,
  - the mechanical properties that result,
  - The process is easy to automate,
  - Filler metal and fluxes are not needed.
- Failure of a resistance weld occurs either by
  - Shearing of the weld or by tearing of the metal around the weld.
  - Because of the possibility of tearing, it is good practice to avoid loading a resistance-welded joint in tension.
- Thus, for the most part, design so that the spot or seam is loaded in pure shear.
  - The shear stress is then simply the load divided by the area of the spot.
  - Because the thinner sheet of the pair being welded may tear, the strength of spot welds is often specified by stating **the load per spot based on the thickness of the thinnest sheet.**
  - **Such strengths are best obtained by experiment.**



Spot welding

Seam Welding

actually a series of overlapping spot welds



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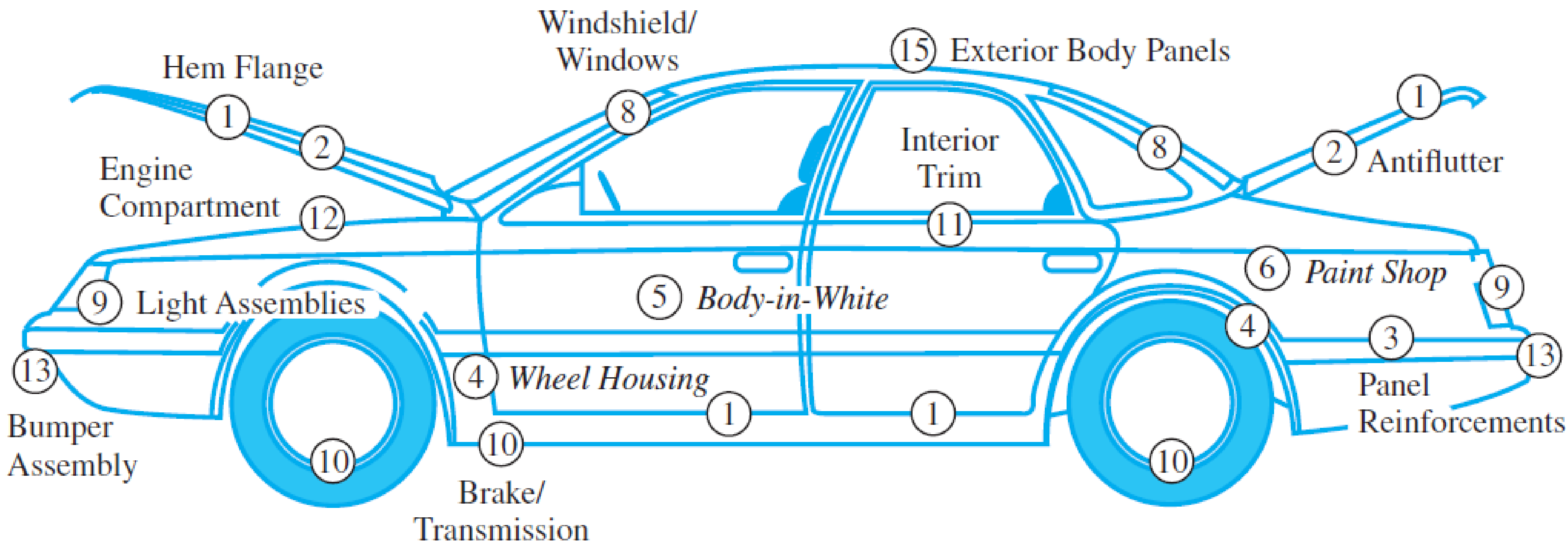
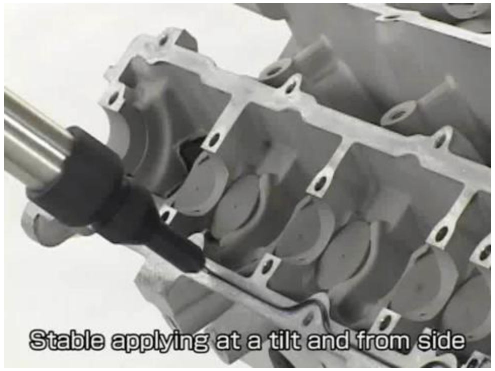
Adhesive Bonding

The use of polymeric adhesives to join components for structural, semi-structural, and nonstructural applications



- Significant reduction in weight
  - Less fasteners
- Permit the use of thinner-gauge materials
  - Stress concentration associated with the holes are eliminated
- Significant reduction in noise, vibration, and harshness
  - Energy dissipation by polymeric adhesives
- To assemble heat-sensitive materials or components that might be damaged by drilling holes for mechanical fasteners
- Join dissimilar materials or thin gauge stock that cannot be joined through other means.



Application of adhesive using robotics and NETZSCH Dispensing Pump to engine block

https://www.youtube.com/watch?v=J8MpHQr7jE8

Automated Adhesive Dispensing System – Automotive Robotics

https://www.youtube.com/watch?v=WJ2bt10FyO8

Automotive Sealing System with FANUC M-20iA Robots - Courtesy of Inovision

https://www.youtube.com/watch?v=Wc2DvmLtZMI

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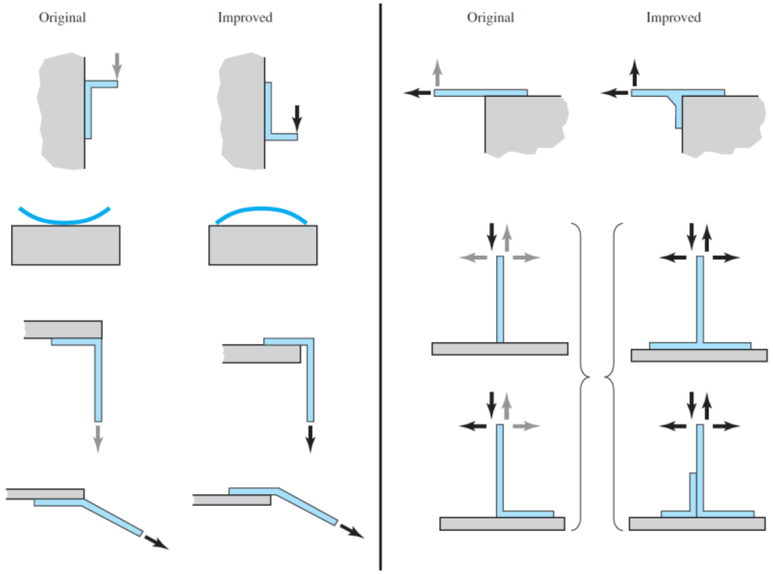
Mechanical Performance of Various Types of Adhesives

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)

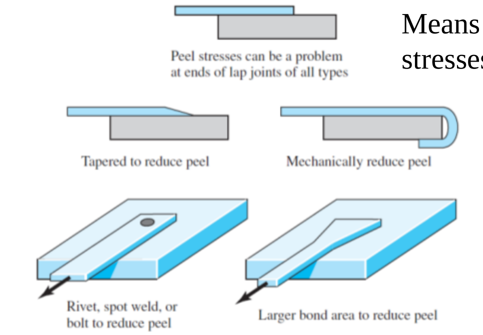
Joint Design

Some basic guidelines that should be used in adhesive joint design

- **Design to place bondline in shear, not peel.**
  - Beware of peel stresses focused at bond terminations. When necessary, reduce peel stresses through tapering the adherend ends, increasing bond area where peel stresses occur, or utilizing rivets at bond terminations where peel stresses can initiate failures.
- **Where possible, use adhesives with adequate ductility.**
  - The ability of an adhesive to yield reduces the stress concentrations associated with the ends of joints and increases the toughness to resist debond propagation.
- **Recognize environmental limitations of adhesives and surface preparation methods.**
  - Exposure to water, solvents, and other diluents can significantly degrade adhesive performance in some situations, through displacing the adhesive from the surface or degrading the polymer. Certain adhesives may be susceptible to environmental stress cracking in the presence of certain solvents. Exposure to ultraviolet light can also degrade adhesives.
- **Design in a way that permits or facilitates inspections of bonds where possible.**
  - A missing rivet or bolt is often easy to detect, but debonds or unsatisfactory adhesive bonds are not readily apparent.
- **Allow for sufficient bond area so that the joint can tolerate some debonding before going critical.**
  - This increases the likelihood that debonds can be detected. Having some regions of the overall bond at relatively low stress levels can significantly improve durability and reliability.
- **Where possible, bond to multiple surfaces to offer support to loads in any direction.**
  - Bonding an attachment to a single surface can place peel stresses on the bond, whereas bonding to several adjacent planes tends to permit arbitrary loads to be carried predominantly in shear.
- **Adhesives can be used in conjunction with spot welding.**
  - The process is known as weld bonding. The spot welds serve to fixture the bond until it is cured.



Gray load vectors are to be avoided as resulting strength is poor.



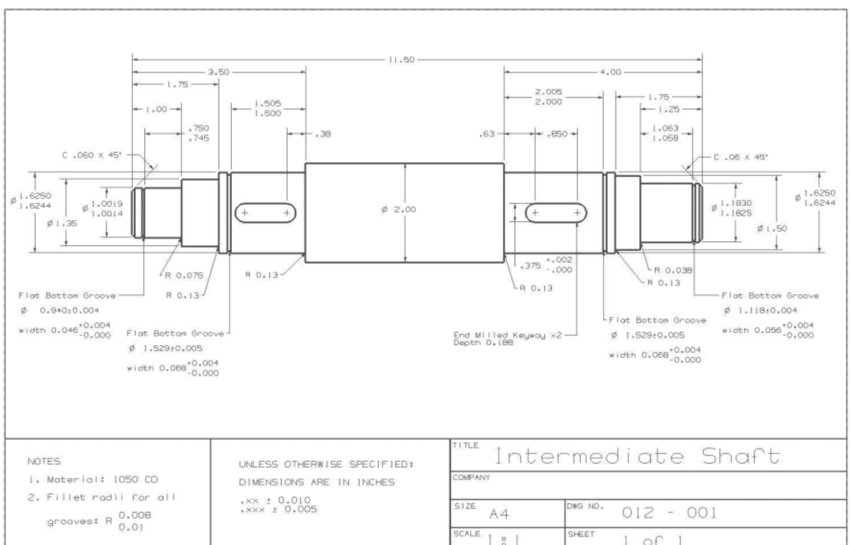
Means to reduce peel stresses in lap-type joints.

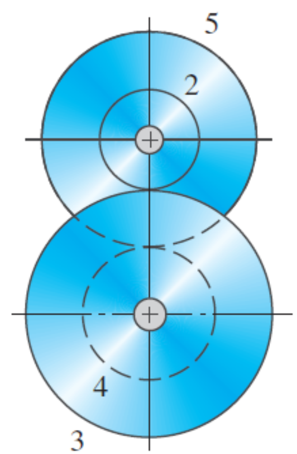
A Gentle Reminder of the 3rd Design Project

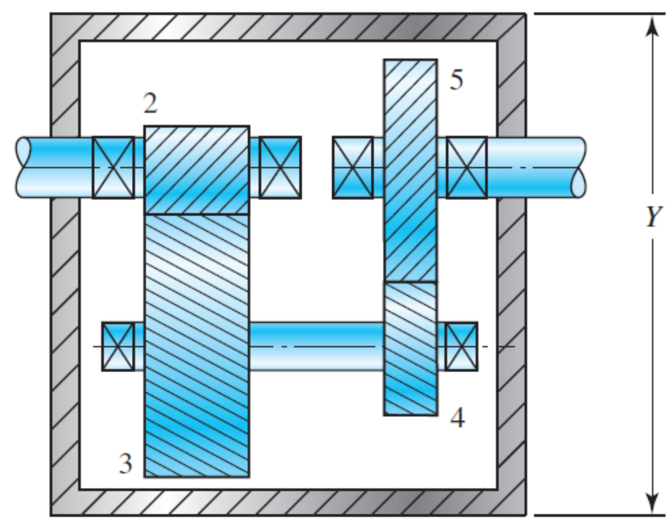
To be submitted before the end of Week 13's class.

Power to be delivered: ~~20 hp~~ **40 hp**
Input speed: 1750 rpm
Output speed: 82-88 rev/min
Usually low shock levels, occasional moderate shock
Input and output shafts extend 4 in outside gearbox
Maximum gearbox size: 14-in × 14-in base, 22-in height
Output shaft and input shaft in-line
Gear and bearing life > 12 000 hours; infinite shaft life

Design the intermediate shaft, including complete specification of the gears, bearings, keys, retaining rings, and shaft.







A Gentle Reminder of the 2nd Design Project

Scoring Rubrics

- Things to include in your report
  - Design Need Analysis
  - Existing Design Review
  - Mechanical Design Analysis
  - Fabrication & Assembly
  - Testing & Presentation

- Design consultation with the Teaching Assistants
  - Purchase request ends next Friday (29 Nov)
  - Make sure to talk with the TAs to refine your design and report
  - Make sure you have a full report and a working prototype
  - No need for control integration, but mainly on mechanical design

Design Testing Report Submission

- Online at course website

Next class

- **Lab for Group 2**: Design Consultation
- Friday 0800-1000, Nov 22
- Room 412, 5 Wisdom Valley

- **Discussion for Group 1**: Design Consultation
- Friday 0800-1000, Nov 22
- Room 202, 1 Lychee Park

Thank you!

Song Chaoyang (songcy@sustech.edu.cn)

- Xiao Xiaochuan (xiaoxc@sustech.edu.cn)
- Yu Chengming (11930324@mail.sustech.edu.cn)
- Zhu Wenpei (11930368@mail.sustech.edu.cn)
- Guo Ning (11930729@mail.sustech.edu.cn)

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