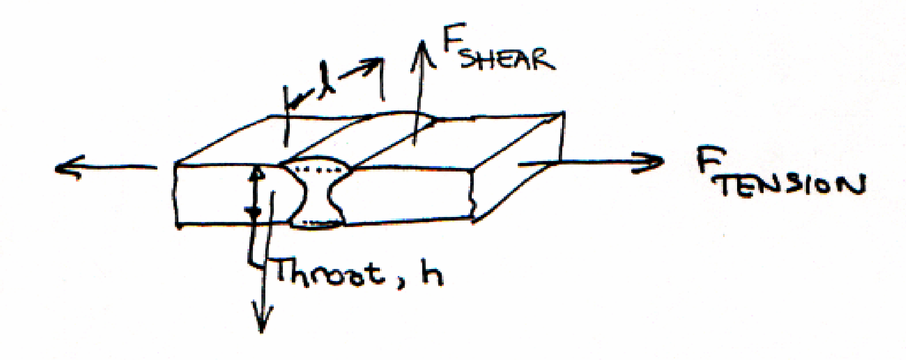
WELDED JOINTS

Butt Welds



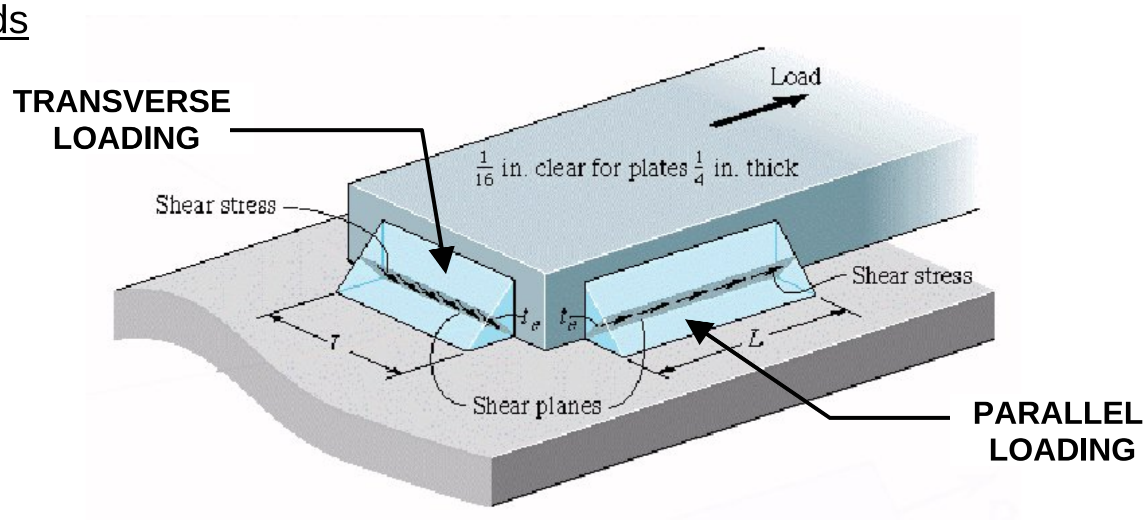
$$A = h \cdot l$$

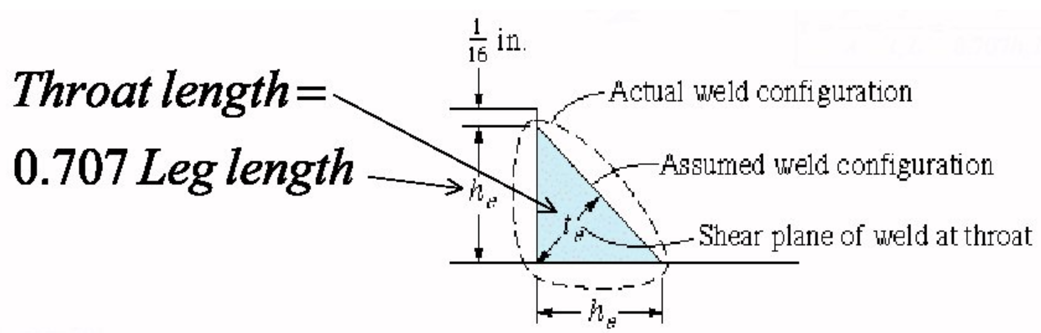
$$\sigma = \frac{F_{TENSION}}{A} = \frac{F}{hl}$$

$$\tau = \frac{F_{SHEAR}}{A} = \frac{F}{hl}$$

- Strong
- Inspectable
- No "built cracks"

Fillet Welds





Fillet welds fail by shearing at the minimum section – at the throat of the weld.

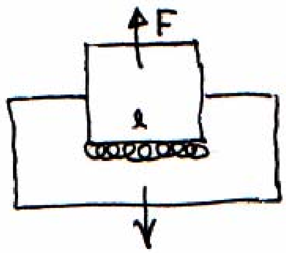
$$\tau = \frac{Load}{t_e L} = \frac{P}{0.707\,hL} = \frac{1.414\,P}{hL} \qquad \text{(Eqn. 16.49)}$$

Keep $\tau < S_{sy} = 0.4\ S_y$ (From Eqn, 3.14, P. 109)

See Table 16.13 for AISC weld electrode strengths.

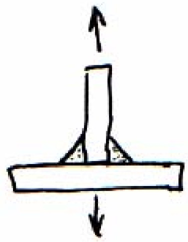
Four Basic Loadings on Fillet Welds

1. TRANSVERSE



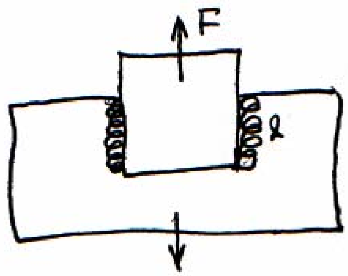
FATIGUE FACTOR

$$\tau = \frac{1.414F}{h\ell} \qquad K_f = 1.5$$



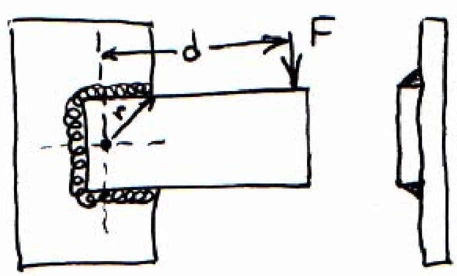
$$\tau = \frac{1.414\,F/2}{h\ell} = \frac{F}{1.414h\ell} \qquad K_f = 2$$

2. PARALLEL



$$\tau = \frac{F}{1.414h\ell} \qquad K_f = 2.7$$

3. TORSION

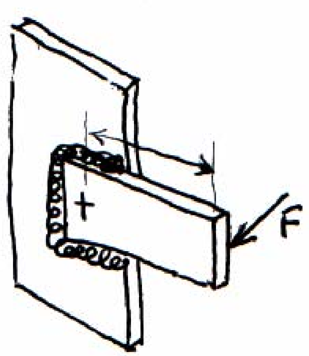


PRIMARY SHEAR $= \dfrac{F}{A_{TOT}}$

SECONDARY SHEAR (TORSIONAL) $= \dfrac{Mr}{J}$

SECOND POLAR MOMENT OF WELD AREA ABOUT CENTROID

4. BENDING

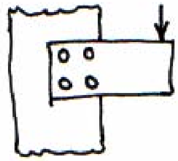


PRIMARY SHEAR $= \dfrac{F}{A_{TOT}}$

NORMAL STRESS $= \dfrac{MC}{I}$

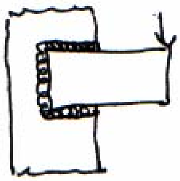
To Understand Torsional Loading, First Consider Bolted Joint

EX. 1: BOLTED JOINT



- GET DIRECT SHEAR
- FIND CENTROID OF BOLT GROUP (EQ. 4.5)
- GET TORSIONAL SHEAR
- VECTORIALLY SUM - FIND BIGGEST

EX. 2: WELDED JOINT



- GET DIRECT SHEAR = LOAD / TOTAL WELD AREA
- FIND CENTROID OF WELD GROUP
  - SEE TABLE 16.12
- GET TORSIONAL SHEAR
  - USE <u>UNIT</u> POLAR AREA MOMENT OF INERTIA, J_u, BY SETTING THE WELD THROAT = 1.
  - SEE TABLE 16.12 FOR J_u FORMULAS FOR NINE COMMON WELD JOINTS.
  - ONCE YOU GET J_u,

$$J = t_e J_u = 0.707\, h_e J_u \qquad (16.53)$$

  - THEN TORSIONAL SHEAR IS

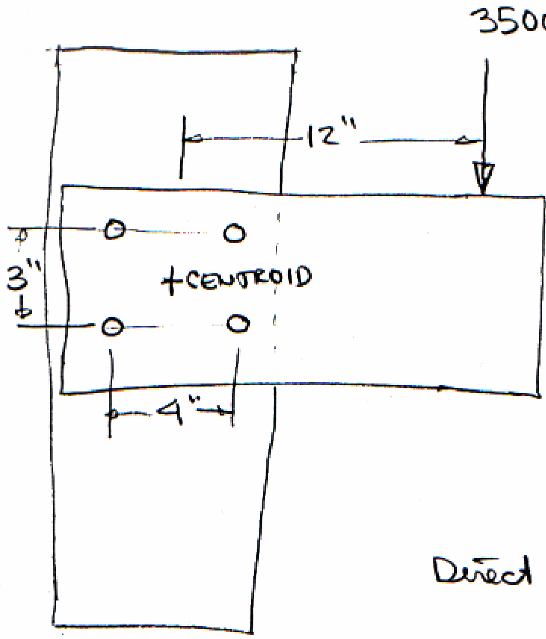
$$\tau = \frac{T r}{J}$$

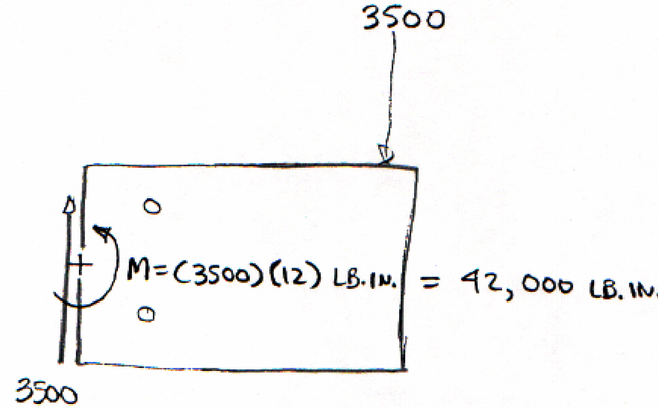
PICK POINTS ON WELD AT LARGEST RADIUS r FROM CENTROID.

- <u>VECTORIALLY</u> SUM DIRECT AND TORSIONAL SHEAR.

→ WATCH OUT FOR EQ. 16.54 THAT SHOWS $\tau = \tau_d + \tau_t$

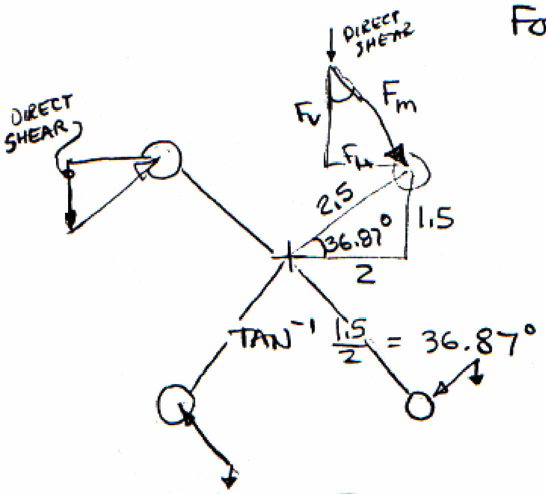
Example 1: Bolted Joint Loaded in Torsion





Direct shear on each bolt $= \frac{3500}{4} = 875$ LB/BOLT

Force to resist rotating moment



$$(4)(2.5)\,F_m = 42{,}000 \text{ LB.IN}$$

$$F_m = \frac{42{,}000}{10} = 4200 \text{ LB/BOLT}$$

$$F_{V_m} = F_m \cos 36.87° = 4200 \cdot \frac{2}{2.5} = 3360 \text{ LB}$$

$$F_{H_m} = F_m \sin 36.87° = 4200 \cdot \frac{1.5}{2.5} = 2520 \text{ LB.}$$

$$F_{TOT} = F_m + F_{shear} = \sqrt{(F_{V_m} + F_s)^2 + F_{H_m}^2} = \sqrt{(3360 + 875)^2 + 2520^2}$$

$$F_{TOT/BOLT} = 4928 \text{ LB}$$

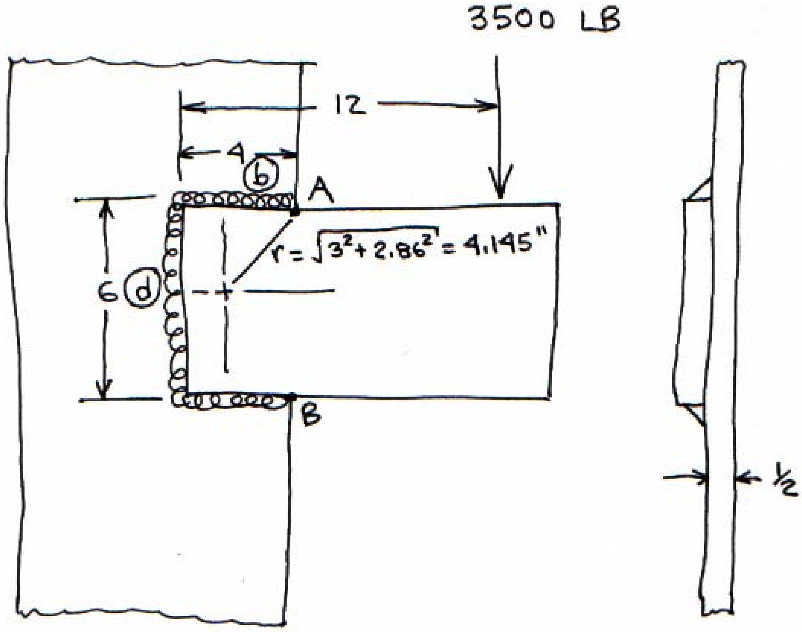
IF BOLT HAD AN ALLOWABLE SHEAR STRESS OF 18 KSI

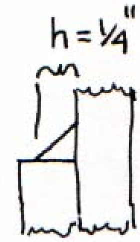
$$\text{Area} = \frac{F_{TOT}}{\tau_{allow}} = \frac{4928}{18{,}000} = 0.274 \text{ IN}^2$$

$$D = \sqrt{\frac{4A}{\pi}} = \sqrt{\frac{1.095}{\pi}} = 0.59'' \qquad A = \pi\frac{d^2}{4}$$

$\rightarrow$ PICK $\frac{5}{8}$" DIA = 0.625 IN.

Example 2: Weld Joint Loaded in Torsion





USING TABLE 15.9

TORSION

$$\bar{x} = \frac{b^2}{2b+d} = \frac{4^2}{8+6} = 1.14''$$

$$\bar{y} = \frac{d}{2} = 3''$$

$$J_u = \frac{(2b+d)^3}{12} - \frac{b^2(b+d)^2}{(2b+d)}$$

$$= \frac{14^3}{12} - \frac{16(10^2)}{14}$$

$$J_u = \frac{2744}{12} - \frac{1600}{14} = 114.38 \text{ IN}^3$$

$$J = 0.707\, h\, J_u$$

$$= (0.707)(0.25)(114.38)$$

$$J = 20.22 \text{ IN}^4$$

WELD AREA

$$A = (0.707)(0.25)(4+6+4)$$

$$A = 2.475 \text{ IN}^2$$

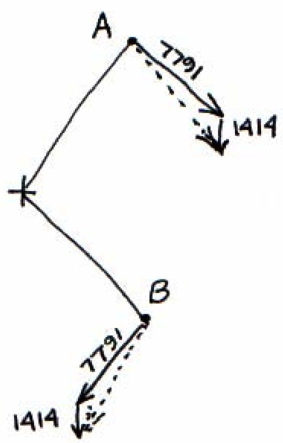
DIRECT SHEAR

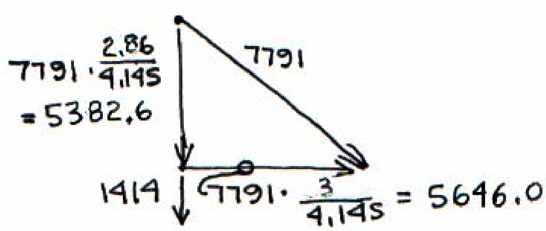
$$\tau_d = \frac{3500}{A} = \frac{3500}{2.475} = 1414 \text{ PSI}$$

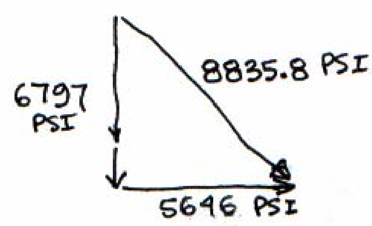
TORSIONAL SHEAR

$$\tau_t = \frac{Mr}{J} = \frac{(3500)(12-1.14)(4.145)}{20.22}$$

$$\tau_t = 7791.5 \text{ LB/IN}^2$$







FOR E60XX ELECTRODE,

$$S_y = 50 \text{ KSI}$$

$$S_{sy} = 0.4 S_y = 20 \text{ KSI}$$

$$n_s = \frac{S_{sy}}{\tau_{TOT}} = \frac{20000}{8836}$$

$$n_s = 2.26$$

Bending Load on Weld

$\boxed{\text{HAMROCK CORRECTIONS}}$

A) TABLE 16.12 GIVES UNIT SECTION MODULUS, Z_u, NOT I_u.

$$Z_u = \frac{I_u}{c}$$

$$Z = t_e Z_u = 0.707\, h_e Z_u$$

$$\sigma = \frac{M}{Z}$$ GIVES TENSILE STRESS

B) CROSS OUT EQUATION (16.55)

- DIRECT SHEAR IS STILL = LOAD/TOTAL WELD AREA
- COMBINE TENSILE AND SHEAR STRESSES

$$\sigma_e = \sqrt{\sigma^2 + 3\tau^2}$$ (VON MISES)

- COMPARE TO ELECTRODE YIELD STRENGTH, NOT SHEAR STRENGTH.

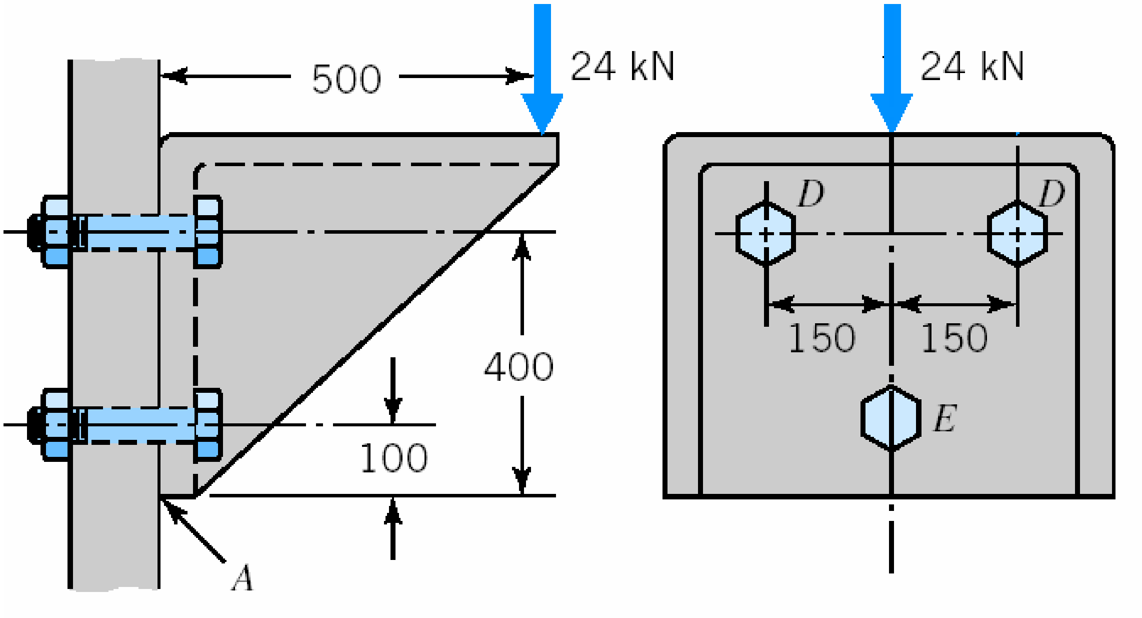
C) EQUATION (16.54) IS VECTOR ADDITION

$$\vec{\tau} = \vec{\tau}_d + \vec{\tau}_t < (S_{sy})_{weld}$$

D) TABLE 16.12 (P.747) LOWER LEFT CORNER:

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

Example 3: Bolted Bracket (Shear Carried by Friction)



Select the proper size of SAE class 9.8 steel bolts for a minimum factor of safety of 6 (based on proof strength) for this loaded bracket.

Note: Assumption for this type of arrangement is that the load tends to pivot the bracket about an axis through point A.

Because of this assumption of "hinging" about A, the strain (and therefore the tension) on the bolts is proportional to their distance from A. The tension load F_D on the two D bolts is four times the load F_E on the E bolt: $F_E = 0.25\, F_D$.

We do a summation of moment about A for the design overload of 6 x 24kN = 144kN:

$$(500)(144) = (100)F_E + (400)F_D + (400)F_D$$

$$72000 = (100)(0.25)F_D + (800)F_D = 825F_D$$

$$F_D = 87.27\,kN$$

Class 9.8 steel bolts have a proof strength of 650 MPa (Table 16.8), so the required tensile area is: $A_t = \dfrac{87{,}270\,N}{650\,MPa} = 134\,mm^2$

Table 16.10 shows that an M16 bolt is required.

Example 4: Bolted Bracket (Shear Carried by Bolts)

If the bolts in the bracket were to be loose enough so that the 24kN was not carried by friction but by the bolt shanks, then we would assume that there would be 24kN vertical shear load equally distributed among the three bolts.

Considering the FOS, this would put an average shear on each bolt of F/A = (144kN / 3) / A = 48kN / A , where A is the shank area. This combines with the tension = 87.27kN / A at the upper bolts, worst case. We use the von Mises equation to combine the stresses:

$$\sigma_e = \sqrt{\sigma \quad + 3\tau^2} = \frac{1}{A}\sqrt{(87{,}270)^2 + 3(48{,}000)^2} = \frac{120{,}532}{A}$$

Equating this to the bolt proof strength gives

$$\frac{120{,}532}{A} = S_p = 650\,MPa$$

And A = 185 mm^2 .

Then

$$d = \sqrt{\frac{4A}{\pi}} = \sqrt{\frac{(4)(185)}{\pi}} = 15.36\,mm$$

So a bolt with a shank diameter of 16mm is needed. (This happens to be the same size as before, but it is not always the case.)

This example is from *Fundamentals of Machine Component Design*, 3rd Edition, Juvinall & Marshek, John Wiley, New York, 2000.