# A Rational Approach to Design of Tee Shear Connections

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A tee shear connection (Figure 1) is similar to a shear tab in that there is a perceived lack of ductility in the connection which could cause premature fracture in the shop weld or shop bolts, i.e., fracture before the connection is able to absorb the self-limited beam end rotation in a ductile manner. This problem is resolved with a shear tab by two ductility requirements, e.g.,

1. the bolt diameter to tab thickness ratio must be greater than or equal to 2 ($d/t \geq 2$). This criterion guarantees that the tab experiences ductile bearing failure before the bolts fracture in shear.
2. The fillet weld of the tab to the support must have a leg size greater than or equal to 75 percent of the tab thickness ($w/t \geq 0.75$). This criterion guarantees that the tab plate yields before the weld yields.

A tee shear connection differs from a shear tab connection in that there is another element of the connection that can be used as a ductility provider. This is the flange of the tee.

## DUCTILITY DEMAND OF TEE FLANGE

As the beam end rotates under gravity load, the tee will induce forces in the weld at the tee flange tips or the shop bolts caused by the stiffness and strength of the flange. It is the purpose of this section to estimate the maximum possible force that can be induced in the welds or bolts by the tee flange. Consider the tee of Figure 1. Assume that under load, the tee rotates about its lower edge and induces the distribution of force $V$ in the welds or shop bolts shown in Figure 2. Because the effect of the shear $V$ on the plastic moment capacity $m_p$ of the flange, and the effect of the vertical shear force on $m_p$ are both neglected, this is a worst case which will give a maximum value to the induced couple $M$.

From the geometry of Figure 1

$$\text{Ø} = \theta\sqrt{1+\left(\frac{b}{L}\right)^2},\ \delta = \theta g,\ g = \frac{b}{L}y$$

where

$\theta$ = the rotation at the vertical yield lines and
Ø = the rotation at the inclined yield lines.

Also, $\delta$ is the deflection at the center of the flange of the WT, i.e., the amount by which the center of the WT separates from the supporting member.

The virtual work equation can be written

$$m_p\text{Ø}\sqrt{L^2+b^2} + m_p\theta L = \int_o^L (Vdy)g\theta = \frac{1}{2}VbL\theta$$

With $m_p = \frac{1}{4}F_y t^2$ and $\eta = b/L$, the force per unit length of weld $V$ can be written as

$$V = \frac{1}{2}F_y\frac{t^2}{b}(\eta^2+2) \tag{1}$$

By taking moments about the bottom of the connection of Figure 2, Equation 1 produces a connection couple

$$M = VL^2 \tag{2}$$

which is compared with the physical tests of Astaneh and Nader (1988, 1989) in Table 1. This comparison is made to determine if Equation 2 provides a reasonable upper bound for tee shear connections. The theoretical values of $M$ are calculated for $F_y$ = 44.77 ksi, which is the average yield value of the Astaneh and Nader test specimens.

From Table 1, it can be seen that the theoretical values of $M$ from Equation 2 exceed the experimental values at the very high rotation of 0.07 radian except for Test 1 where the

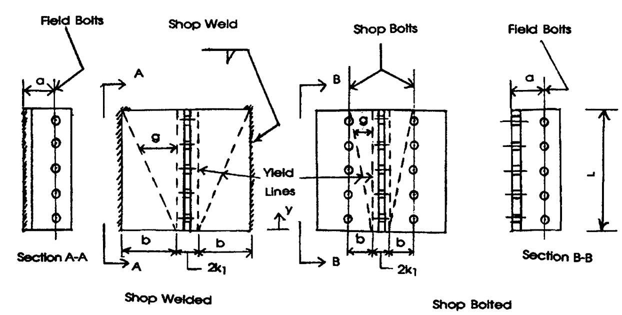


*Fig. 1. Tee shear connections.*

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**Table 1.**
**Theoretical and Experimental Values of Connection Couple (Shop Welded WT)**

| Test No. | Section | $L$ in. | $t$ in. | $b_f$ in. | $k_1$ in. | $b$[1] in. | Theoretical $M$ (Eq. 2) | Experimental $M$ (Astaneh & Nader) | |
|---|---|---|---|---|---|---|---|---|---|
| | | | | | | | $F_y$ = 44.77 k-in. | @Ø = .03 k-in. | @Ø = .07 k-in. |
| 1 | WT4×7.5 | 8.5 | .315 | 4.00 | ½ | 1.5 | 217 | 180[4] | 223 |
| 2 | WT7×19 | 14.5 | .515 | 6.77 | ⅝ | 2.76 | 921 | 300[4] | 533 |
| 3 | WT7×19 | 8.5 | .515 | 6.77 | ⅝ | 2.76 | 327 | 110[4] | 218 |
| 4 | WT4×7.5 | 14.5 | .315 | 4.00 | ½ | 1.5 | 626 | 370[4] | 413[6] |
| 5 | WT4×20 | 14.5 | .56 | 8.07 | ⅝ | 3.41 | 890 | 250[4] | 479[5] |
| 6 | WT4×20 | 8.5 | .56 | 8.07 | ⅝ | 3.41 | 321 | 50[4] | 227 |
| 7 | WT7×19[2] | 14.5 | .515 | 6.77 | ½[3] | 2.89 | 881 | 290[4] | 683 |
| 8 | WT4×20[2] | 8.5 | .56 | 8.07 | ½[3] | 3.53 | 312 | 30[4] | 228 |
| 9 | WT4×20[2] | 14.5 | .56 | 8.07 | ½[3] | 3.53 | 861 | 400[4] | 748 |

[1] $b = b_f/2 - k_1$
[2] with stem replaced with ½ A36 plate, ¼ fillet welds
[3] $k_1$ = ¼ + ¼ = ½
[4] Estimated from Figure 4.5 of Astaneh & Nader (1988) report to AISC.
[5] 47.9 in original. Data assumed corrupt.
[6] Value is for .06 radian.

theoretical and experimental values are essentially equal. The experimental values of $M$ for beam end rotation Ø = .03 radian are reasonable maximums for $M$ because Ø = .03 exceeds the beam end rotation of most beams when a plastic hinge forms at the center.

## SHOP WELDED CONFIGURATION

From these results, it is conservative to take the theoretical $M$ of Equation 2 as the maximum possible couple able to be generated by the T flange, even in the case of the very high rotation of 0.07 radians. Then, the maximum possible force per unit length of weld is $V$ from Equation 1, and this is the ductility demand of the T flange. In order that the weld be able to absorb this demand without fracture, we require that

$$\frac{1}{\sqrt{2}} w y F_{exx} \geq V \qquad (3)$$

where

$w$ = fillet weld size
$y$ = weld strength factor for transverse fracture of fillet weld

With $y = \sqrt{2/3} = .8165 \approx .8$, $F_{exx} = 70$, Equation 3 becomes

$$w \geq .0253V \qquad (4)$$

Adding 25 percent margin to Equation 4 to account for the actual yield strength exceeding the min. specified yield strength, gives

$$w_{min} \geq .0316V \qquad (5)$$

Table 2 gives the minimum weld sizes for the nine tests of Astaneh and Nader.

Included in Table 2 is the minimum weld size that would have been required if the tee stem were cut from the tee and welded direct to the support as a shear tab. As expected, the minimum weld size required by Equation 5 is always less than that required by a shear tab.

### Design Algorithm

1. Select a WT and determine the minimum weld size as $W_{min} = \min\{0.0316V, 0.75t_s\}$.
2. Check that $d / t_s \geq 2$.
3. Rigid support—design shop welds for applied shear $R$ and couple $Ra$; design field bolts for $R$.
4. Flexible Support—design shop welds for applied shear $R$; design field bolts for $R$ and couple $Ra$.
5. Make all the usual additional design checks, i.e., gross

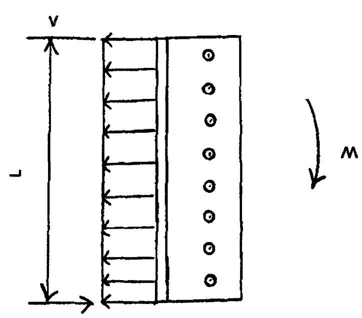


*Fig. 2. Induced force in shop welds or bolts (equal to shear at vertical yield lines).*

shear, net shear, edge distance, bearing, and stem buckling.

## SHOP BOLTED CONFIGURATION

It is proposed that the same formulas, Equations 1 and 2, be used for this case, with the dimension $b$ taken to the center line of the shop bolts. No experimental data to test this proposal for shop bolted WT shear connection appears to be available, but data is available for a similar type of connection, field bolted double angles as shown in Figure 3 (Lewitt, Chesson, and Munse, 1969). Using the data of this report, Table 3 shows excellent agreement between the theoretical connection couple from Equation 2 and the experimentally obtained couple at a beam end rotation of 0.03 radian.

Figure 1 shows the configuration. Equation 1 gives the estimated maximum force per inch of tee length.

Proceeding as in the shop welded configuration, the maximum force $V$ produces a maximum couple

$$M = VL^2 \qquad \text{(2 repeated)}$$

Also, the stem of the tee can produce a maximum couple

$$M = \frac{1}{4} F_y t_s L^2 \qquad (6)$$

From Equations 2 and 6

$$\frac{1}{4} F_y t_s = V \qquad (7)$$

The ultimate tensile strength of A325 bolts is 90 ksi. Assuming the usual bolt spacing of $p$ = 3 inches, for the bolts to be able to resist the maximum couple that could be developed by the tee flange

$$90 \frac{\pi}{4} d^2 \geq 3V \qquad (8)$$

Including a margin of 25 percent

$$d^2 \geq 3V \times 1.25 \times \frac{4}{\pi} \times \frac{1}{90} = 0.053V$$

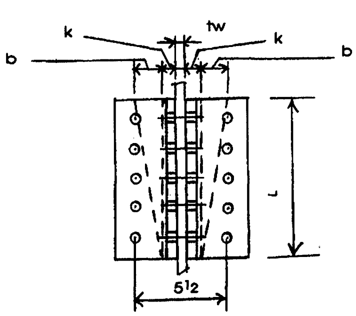


*Fig. 3. Double angle shear connection.*

**Table 2.**
**Minimum Weld Sizes, Astaneh and Nader Tests**

| Test No. | $\eta$ | $V$ @ $F_y$ = 36 Eq. 1 k/in. | $w_{min}$ Eq. 5 in. | $w_{min}$[1] $.75t_s$ in. | $w$ Test Spec. in. |
|---|---|---|---|---|---|
| 1 | .1765 | 2.42 | .076 | .184 | 3/16 |
| 2 | .1903 | 3.52 | .111 | .233 | 1/4 |
| 3 | .3247 | 3.64 | .115 | .233 | 1/4 |
| 4 | .1034 | 2.39 | .076 | .184 | 3/16 |
| 5 | .2352 | 3.40 | .107 | .270 | 1/4 |
| 6 | .4012 | 3.58 | .113 | .270 | 1/4 |
| 7 | .1993 | 3.37 | .106 | .375 | 1/4 |
| 8 | .4153 | 3.47 | .110 | .375 | 1/4 |
| 9 | .2434 | 3.29 | .104 | .375 | 1/4 |

[1] $w_{min}$= $.75t_s$ is minimum weld for shear tab of thickness $t_s$.

Thus

$$d \geq 0.23\sqrt{V} \qquad (9)$$

Considering next the tee stem, the bolts must be strong enough to resist the couple that could be produced by the stem. Then

$$90 \frac{\pi}{4} d^2 \geq 3V = \frac{3}{4} F_y t_s \qquad (10)$$

where Equation 7 has been used. With a margin of 25 percent

$$d^2 \geq \frac{3}{4} F_y t_s \times 1.25 \times \frac{4}{\pi} \times \frac{1}{90} = 0.478 t_s$$

where $F_y$ = 36 ksi has been used. Finally

$$d \geq 0.69\sqrt{t_s} \qquad (11)$$

Combining Equations 9 and 11, the minimum bolt diameter to insure ductility is

$$d_{min} = \min\{0.23\sqrt{V}, 0.69\sqrt{t_s}\} \qquad (12)$$

Table 4 shows that Equation 12 gives reasonable minimum bolt sizes, even though it is based on upper bounds for the expected maximum connection couple.

### Design Algorithm

1. Select a WT and check Equation 12 to determine minimum bolt size.
2. Check that $d / t_s \geq 2$.
3. Rigid support—Design shop bolts for applied shear $R$ and couple $Ra$; design field bolts for $R$.
4. Flexible support—Design shop bolts for applied shear $R$ and field bolts for shear $R$ and couple $Ra$.
5. Make all the usual design checks, i.e., gross shear, net shear, edge distance, bearing, and stem buckling.

**Table 3.**
**Theoretical and Experimental Connection Couples**
**(Bolted and Riveted Double Angles)**

| Specimen | Angle Properties $F_y$ ksi | $t$ in. | $L$ in. | $b^1$ in. | $\eta$ $b/L$ | $V$ Eq. 1 k/in. | Theoretical $M$ (Eq. 2) k-in. | Experimental $M$ (Lewitt, Chesson, & Munse) @0.03 Radian k-in. |
|---|---|---|---|---|---|---|---|---|
| FK-3 | 39.3 | .354 | 8½ | 1.771 | .2084 | 2.841 | 205 | 170 |
| FK-4AB | 39.3 | .354 | 11½ | 1.708 | .1486 | 2.915 | 385 | 330 |
| FK-4P | 39.3 | .354 | 11½ | 1.708 | .1486 | 2.915 | 385 | 330 |
| WK-4 | 39.3 | .354 | 11½ | 1.708 | .1486 | 2.915 | 385 | 320 |
| FK-4AB-M | 41.6 | .375 | 11½ | 1.6875 | .1467 | 3.504 | 463 | 370 |
| FB-4 | 38.8 | .371 | 11½ | 1.8165 | .1580 | 2.9767 | 394 | 330 |
| FB-4A | 38.8 | .371 | 11½ | 1.8165 | .1580 | 2.9767 | 394 | 330[2] |
| FK-5 | 36.8 | .443 | 14½ | 1.6195 | .1117 | 4.4872 | 943 | 810 |
| WB-10AB | 40.1 | .440 | 29½ | 1.7475 | .0592 | 4.4503 | 3873 | 3500[2,3] |

[1] $b = (5\frac{1}{2} - t_w(beam))/2 - k$
[2] Slip between clip angles and beam web not included.
[3] Extrapolated from Figure B35 of Lewitt, Chesson, and Munse

**Table 4.**
**Minimum Bolt Diameter from Equation 12 for Selected WTs**

| WT | $t_s$ in. | $L$ in. | $t$ in. | $b_f$ in. | $k_1$ in. | Gage in. | $b$ in. | $\eta$ | $V$ k/in. | $d_{min}$ $.23\sqrt{V}$ in. | $d_{min}$ $.69\sqrt{t_s}$ in. |
|---|---|---|---|---|---|---|---|---|---|---|---|
| 1[1] | .245 | 8.5 | .315 | 4.00 | ½ | — | — | — | — | — | — |
| 2 | .310 | 14.5 | .515 | 6.77 | ⅝ | 4 | 1.375 | .0948 | 6.97 | .61 | .39 |
| 3 | .310 | 8.5 | .515 | 6.77 | ⅝ | 4 | 1.375 | .1618 | 7.03 | .61 | .39 |
| 4 | .245 | 14.5 | .315 | 4.00 | ½ | — | — | — | — | — | — |
| 5 | .360 | 14.5 | .56 | 8.07 | ⅝ | 4½ | 1.625 | .1121 | 6.99 | .61 | .41 |
| 6 | .360 | 8.5 | .56 | 8.07 | ⅝ | 4½ | 1.625 | .1912 | 7.07 | .61 | .41 |
| 7 | .500 | 14.5 | .515 | 6.77 | ½ | 4 | 1.5 | .1034 | 6.40 | .58 | .49 |
| 8 | .500 | 8.5 | .56 | 8.07 | ½ | 4½ | 1.75 | .2059 | 6.59 | .59 | .49 |
| 9 | .500 | 14.5 | .56 | 8.07 | ½ | 4½ | 1.75 | .1207 | 6.50 | .59 | .49 |
| WT9×25 | .355 | 21 | .57 | 7.495 | 13/16 | 4½ | 1.44 | .0685 | 8.14 | .66 | .41 |

[1] Numbers correspond to Astaneh and Nader Test Numbers.

## REFERENCES


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3. Lewitt, C. W., Chesson, E., Jr., and Munse, W. H., "Restraint Characteristics of Flexible Riveted and Bolted Beam to Column Connections," *University of Illinois Engineering Experiment Station Bulletin 500,* Vol. 66, No. 63, January 1969.