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LARGE BOLTED JOINTS

STATIC TENSION TESTS OF LONG BOLTED JOINTS

UNPUBLISHED DISCUSSION

DESIGN AGAINST SLIP

by

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lower than above.

b. **Design Against Slip**

Where joint slip cannot be tolerated an ultimate strength type design can no longer be used. The prime objective of a reasonable design would then be to provide an adequate factor of safety against slip. Again there are two possible avenues of approach to the problem; design strictly on the basis of friction, or specify some allowable shear stress which will accomplish the same goal. Since there is no evident relationship between joint slip and joint length, the effect of joint length can be disregarded in a slip design.

The most direct method for obtaining a reasonable slip design is to adopt a procedure which reflects the true load carrying mechanism—friction. Based on the classic theory of static friction, a general design formula can be developed as follows:

\[
P_s = \mu N
\]

(8.1)

where

- \(P_s\) = the load required to produce slip; this is also equal to the working load (\(P_w\)) times the factor of safety (F.S.)
- \(\mu\) = the slip coefficient; depends on surface condition
- \(N\) = the normal (clamping) force; for a bolted
joint N also equals the number of slip planes (m) times the number of bolts (n) times the average internal bolt tension ($T_i$)

Making these substitutions into Eq. (8.1) will result in the following expression which would serve as the working design formula

$$P_w = \frac{\mu mnT_i}{F.S.} \quad (8.2)$$

or

$$n = \frac{P_w F.S.}{\mu m T_i} \quad (8.2a)$$

The working load ($P_w$) and the number of slip planes ($m$), dictated by the joint configuration, would be known. Values of F.S. would be set down by specifications. Reasonable values of slip coefficient ($\mu$) have been documented in the literature for different surface conditions.

Choosing one of these, it would be the design engineers' responsibility to see that this was achieved by proper job specifications governing faying surface preparation.

Internal bolt tension ($T_i$) would depend on the bolt size being used and the degree of tightening achieved by the different tightening methods.

Realizing the practical significance of utilizing as much of the potential bolt tension as possible, the following table is offered as a guide to reasonable values of internal bolt tension. Note that, in this table, the
minimum allowable bolt tension is assumed to be proof load rather than the 0.9 proof load specified by the 1954 specification. Test results\(^{(6)}\) show that this tension (PL) is exceeded by the present turn-of-nut methods and can be maintained by the calibrated impact wrench also.

**Internal Bolt Tension, \(T_i\)**

<table>
<thead>
<tr>
<th>Bolt Sizes</th>
<th>Minimum</th>
<th>Calibrated Impact Wrench*</th>
<th>(1/2) Turn-of-nut** (from snug)</th>
</tr>
</thead>
<tbody>
<tr>
<td>3/4</td>
<td>28,400 lb</td>
<td>32,600 lb</td>
<td>36,900 lb</td>
</tr>
<tr>
<td>7/8</td>
<td>36,050</td>
<td>41,500</td>
<td>46,800</td>
</tr>
<tr>
<td>1</td>
<td>47,250</td>
<td>54,400</td>
<td>60,400</td>
</tr>
<tr>
<td>1 1/8</td>
<td>56,450</td>
<td>64,500</td>
<td>73,400</td>
</tr>
<tr>
<td>1 1/4</td>
<td>71,700</td>
<td>82,500</td>
<td>93,000</td>
</tr>
</tbody>
</table>

* Approximately \(1.15 \times \text{PL}\)  
** Approximately \(1.30 \times \text{PL}\)  

Before slip of a bolted joint takes place the entire load transfer is accomplished by friction and there are virtually no shear stresses present in the bolts themselves. A slip design procedure based on allowable shear stresses would therefore be an artificial means for accomplishing the design objectives. It is possible, however, to accomplish a slip design based on allowable shear stress. The problem is to determine an allowable shear stress that
will furnish a reasonable factor of safety for various values of the variables $\mu$ and $T_i$. Allowable shear stress $(\tau)$ can be related to slip coefficient $(\mu)$ and bolt tension $(T_i)$ in the following manner:

$$P_w = \tau mn A_{nom} = \frac{\mu mn T_i}{F.S.}$$

or

$$F.S. = \frac{\mu T_i}{\tau A_{nom}}$$

Figure 27 is a plot showing the relationships which exist between these variables for a 7/8" A325 bolt. The relation for other bolt sizes is very similar. For example, if 15 ksi were assumed to be the allowable shear stress under static loading conditions, does a reasonable factor of safety exist for all the various combinations of surface condition and bolt tension? Three values of internal bolt tension are considered:

1. Proof Load.
2. $1.15 \times$ Proof Load - the average bolt tension which would result in a joint tightened by a calibrated impact wrench set to this value and having $\pm$ 15% scatter.
3. $1.30 \times$ Proof Load - approximate bolt tension presently obtained by the 1/2 turn-of-nut from snug method as measured in actual test specimens by the direct tension calibration procedure.

For purposes of illustration two values of slip coefficient, 0.3 and 0.4, have been shown. The lower value has
been obtained by some investigators\(^3\) in tests of small joints with mill scale faying surfaces; tests of large joints show that a higher value is more realistic. In fact, the average slip coefficient of nine full scale joints with tight mill scale faying surface tested at Lehigh University was 0.45.

Figure 28 is a similar plot for the case of static load plus wind \((τ = 1.33 \times 15 \text{ ksi})\). The ranges of possible safety factors are shown in each case. In the case of static load alone (Fig. 27) the least factor of safety is 1.2 while in the case of static load plus wind the least factor of safety is less than unity (0.9).

Consider a fictitious joint subjected to static load plus wind; the slip coefficient is 0.3 and the average bolt tension is the minimum tension allowable (Proof Load). According to Fig. 28 this joint will fail by slipping into bearing. On the other hand, if the turn-of-nut method were used to tighten the bolts in this connection, the average bolt tension \((1.30 \text{ PL})\) would be great enough to prevent slip. Similarly, if a calibrated impact wrench having \(± 15\%\) scatter in bolt tensions were used, the resulting average bolt tension \((1.15 \text{ PL})\) would also be sufficient to prevent slip. Finally, if the slip coefficient of the fictitious joint were equal to 0.4, slip would not occur under any conditions of clamping greater than proof load.
This preceding discussion of a fictitious bolted connection points up several important factors concerning minimum bolt tension, surface condition and joint slip. First, due to the tension scatter produced by present day tightening tools, it would be virtually impossible to assemble a satisfactory joint having an average bolt tension equal to the minimum allowable. Secondly, if care is exercised in protecting contact surfaces against detrimental treatments, slip coefficients higher than 0.3 can be expected.

On the basis of the previous discussion of a slip design based on allowable stresses, it seems reasonable to assume that a basic allowable shear stress of 15 ksi would provide an adequate factor of safety against slip under most conditions likely to be encountered in the field.

One joint designed using an allowable stress of 15 ksi has been tested at Lehigh.(1) This particular joint slipped at a nominal bolt shear stress of 34 ksi, thus providing a factor of safety against slip of more than 2. Another joint was also tested which was designed using an allowable shear stress of approximately 20 ksi (T/S = 1.00/0.96). This joint slipped at a nominal shear stress of 30.7 ksi providing a factor of safety against slip for the static load plus wind design of approximately 1.54. Both of these limited design examples indicate that a basic allowable
stress of 15 ksi for slip joints will provide adequate factors of safety against slip and that a 33 1/3% stress increase for the case of static load plus wind is warranted. This is an important observation since it demonstrates the soundness of the present design procedure (according to the 1954 specification) which allows this 33 1/3% stress increase for combinations of static and wind loads.

8.5 Long Grips

Current rivet specifications carry special provisions for the proportioning of long grip rivets. For example, the AISC Specification states that "Rivets which carry calculated stress, and the grip of which exceeds five diameters, shall have their number increased 1 percent for each additional 1/16 inch in the rivet grip". This stipulation presumably arises because of the increased bending stresses in long rivets.

The bolts of the D-Series - Part b which would fall into this grip classification were proportioned without any regard to this provision because at working load the bolts are transferring load by friction and not by shear, bearing and bending. Even at ultimate load where the bolt is in bearing and subject to bending this effect is of no importance for the grips studied.
FIG. 27 Factor of Safety Against Slip, Static Load

FIG. 28 Factor of Safety Against Slip, Static and Wind Load