Chapter 14: Screws and Fasteners

Fastening (more complex shapes = better function)

- Non-permanent
  - Bolted

- Permanent
  - Bolted
  - Welded
  - Bonded

Outline

- General Thread Nomenclature & Types
- Power Screws
- Stresses in Threads
- Preloading Fasteners/Joints
- Fasteners in Shear

Screws

with several figures from:

Table 14.2: Screw threads (ISO/ metric)
Example: M4 x 0.7, implies 4 mm diameter and 0.7 mm pitch

Table 14-2
Basic Dimensions of ISO Metric Screw Th...

<table>
<thead>
<tr>
<th>Nominal Diameter d (mm)</th>
<th>Pitch p (mm)</th>
<th>Minor Diameter d_f (mm)</th>
<th>Stress Area A_s (mm²)</th>
</tr>
</thead>
<tbody>
<tr>
<td>3</td>
<td>0.5</td>
<td>2.39</td>
<td>5.08</td>
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<tr>
<td>4</td>
<td>0.7</td>
<td>2.77</td>
<td>7.67</td>
</tr>
<tr>
<td>5</td>
<td>0.8</td>
<td>4.12</td>
<td>14.3</td>
</tr>
<tr>
<td>6</td>
<td>1</td>
<td>4.77</td>
<td>21.1</td>
</tr>
<tr>
<td>7</td>
<td>1</td>
<td>5.77</td>
<td>26.9</td>
</tr>
</tbody>
</table>

**see Tables 14-1 and 14-2 for standard sizes**
Types of Screw Fasteners

Classification by Intended Use

The same fastener may take on a different name when used in a particular manner. For example, a **bolt** is a fastener with a head and straight threaded shank intended to be used with a **nut** to clamp an assembly together. However, the same fastener is called **machine screw** or **cap screw** when it is threaded into a tapped hole rather than used with a nut.

A **stud** is a headless fastener, threaded on both ends and intended to semipermanently threaded into one-half of an assembly. A hole in the mating part then drops over the protruding stud and is secured with a nut.

Classification by Thread Type

All fasteners intended to make their own hole or make their own threads are called **tapping screws**.

**A) Slotted Screws**: Head shape can be flat, round, oval, etc. The head style can slotted or Phillips grooves. We thread them into a hole using a screw driver.

**B) Socket-Head Cap Screws**: These are extensively used in machinery. The hex socket allows sufficient torque to be applied with hexagonal Allen wrenches.

Nuts and Washers

- **Lock Nuts**: eliminates loosening of nuts due to vibration.
- **Lock Washers**: eliminates loosening of standard nuts due to vibration.

- **Standard Nuts**
Power Screw

Power screws, also called lead screws, are used to convert rotation to linear motion in actuators, machine tools, clamps, and jacks. They can lift or move large loads.

Power Screw Applications

Where have you seen power screws?
- jacks for cars
- C-clamps
- vises

Power Screw Types

Square thread
- \( d_1 = d - \frac{p}{2} \)
- \( d_2 = d - p \)
- strongest
- no radial load
- hard to manufacture

Acme
- 29° included angle
- easier to manufacture
- common choice for loading in both directions

Buttress
- great strength
- only unidirectional loading

Load Analysis

A screw thread is essentially an inclined plane that has been wrapped around a cylinder to create a helix.

\[
\tan \lambda = \frac{L}{ndp}
\]
Load Analysis

1) Sum of horizontal forces = 0
\[ F - f \cos \lambda - N \sin \lambda = 0 \]

2) Sum of vertical forces = 0
\[ N \cos \lambda - f \sin \lambda - P = 0 \]

From Eqs 1 and 2:
\[ F = P \left( \frac{\mu \cos \lambda + \sin \lambda}{\cos \lambda - \mu \sin \lambda} \right) \]

Load Analysis

The screw torque \( (T_{su}) \) required to lift load is
\[ T_{su} = \frac{P d_l}{2} \left( \frac{\mu m_l \alpha + L}{m_l + \mu d_l} \right) \]
\[ L = (\tan \lambda)(m_l') \]
\[ T_{su} = \frac{P d_l}{2} \left( \frac{\mu m_l \alpha + L}{m_l + \mu d_l} \right) \]

Collar Torque

LIFTING
\[ T_d = T_{collar} + \left( T_{collar} \right) = \frac{P}{2} \left[ \mu \left( \frac{\mu m_l \alpha + L}{m_l + \mu d_l} \right) \right] \]

LOWERINg
\[ T_d = \frac{P}{2} \left[ \mu \left( \frac{\mu m_l - L}{m_l + \mu d_l} \right) \right] \]

For Acme Threads

The radial angle introduces an additional factor in the torque equations

Self-Locking

Self-locking screw cannot be turned from applied load \( P \). In other words, self-locking screw will hold the load in place without any application of torque \( (T_u=0) \).

For self-locking:
\[ \mu \geq \frac{L}{m_l \cos \alpha} \quad \text{or} \quad \mu \geq \tan \lambda \cos \alpha \]

If it is a square thread \( (\cos \alpha = 1) \): 
\[ \mu \geq \frac{1}{m_l} \quad \text{or} \quad \mu \geq \tan \lambda \]
Efficiency

Efficiency is work delivered by the screw in one revolution/ work done on a power screw.

\[ \eta = \frac{W_{out}}{W_{in}} = \frac{PL}{2 \pi T} \cos \alpha - \mu \tan \lambda \]

\[ \cos \alpha + \mu \cot \lambda \]

Outline

- General Thread Nomenclature & Types
- Power Screws
- Stresses in Threads
  - Body Stresses
    - Axial
    - Torsion
  - Thread Stresses
    - Bearing
    - Bending
  - Buckling
- Preloading Fasteners/Joints
- Fasteners in Shear

Fasteners: Static and Fatigue Analysis

Static problem: Fixed pressure
- What size and material bolt to use?
- How much to tighten?

Fatigue problem: Varying pressure
- What size and material bolt to use?
- How much to tighten?
- Predict life in cycles

Possible failure locations:
- Threads
- Body
- Neck

Axial Tensile Stress

\[ \sigma_t = \frac{F}{A_f} \]

\[ A_f = \frac{\pi}{4} \left( \frac{d_i}{2} + \frac{d_r}{2} \right)^2 \]

Torsional Stress

Depends on friction at screw-nut interface

For screw and nut:
- If totally locked (stapled together), the screw experiences all of torque
- If frictionless, the screw experiences none of the torque

\[ \tau_{J} = \frac{T_{J}}{J} = \frac{16T}{\pi d_r^3} \]

For power screw:
- If low collar friction, the screw experiences nearly all of torque
- If high collar friction, the nut experiences most of the torque

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Preloading & Proof Strength

- Bolts and screws are typically preloaded and the proof strength is taken as the reference for preloading (taking yield strength as reference for preloading may cause a damage on the material). $S_p$ is the stress at which bolt begins to take a permanent set and it is close to, but lower than yield strength of the material.

### Preloading
- **static loading:** preload at roughly 90% of $S_p$
- **dynamic loading:** preload at roughly 75% of $S_p$

Spring Behavior

Both material being clamped and bolt behave as springs

$$k = \frac{AE}{l}$$

For the bolt, threaded vs unthreaded have different spring constants and are modeled as springs in series:

$$k = \frac{AE_1}{l_1} + \frac{AE_2}{l_2}$$

Affected Area of Material

For material, basic model is as follows:

$$A_{m} = \frac{2}{d} \left( \frac{d^2 + d_2^2}{2} - d_2^2 \right)$$

Loading & Deflection

A preload $F_i$ is applied up to 90 percent of the proof strength.

$$F_i = 0.9 \times S_p \times A_t$$

Proof strength depends on material. See Table 14.7

### Loading & Deflection

- Slope of the bolt line is positive because its length increases with increased force.
- Slope of the material is negative as its length decreases with increasing force.
- Bolt stretches more than the material compresses.
- Material is typically stiffer than the bolt ($k_m > k_b$ since $A_m > A_b$)

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<table>
<thead>
<tr>
<th>Class Number</th>
<th>Material</th>
<th>Field Strength (MPa)</th>
<th>Ultimate Field Strength (MPa)</th>
<th>Tensile Strength (MPa)</th>
<th>Material</th>
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<tbody>
<tr>
<td>4A</td>
<td>M15-M18</td>
<td>750</td>
<td>1600</td>
<td>1400</td>
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<td>800</td>
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<tr>
<td>7D</td>
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<td>alloy special tempered</td>
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</tbody>
</table>
Loading & Deflection

\[ \Delta \delta = \Delta \delta_m + \Delta \delta_b \]
\[ P_m = P_b + P_m \]
\[ F_{m} = F_i + P(C-1) \]
\[ F_{b} = F_i + C P \]
\[ \sigma_b = \frac{F_b}{A_i} \]
\[ N_{separation} = \frac{F_i}{P(1-C)} \]

Distribution of Applied Load

\[ \Delta \delta = \frac{P}{k_s} - \frac{P_m}{k_m} \]
\[ P_m = \frac{k_m}{k_m+k_b} P \]
\[ P_b = C P, \quad \text{where} \]
\[ C = \frac{k_b}{k_m+k_b} \]
\[ P_m = P - C P = (1-C) \]

Yielding Safety Factor

\[ F_{m} = F_i + P_m \]
\[ F_{b} = F_i + P_b \]
\[ \sigma_b = \frac{F_b}{A_i} \]
\[ N_{separation} = \frac{P}{P(1-C)} \]

Separation

\[ \Delta \delta = \Delta \delta_m + \Delta \delta_b \]
\[ P_m = P_b + P_m \]
\[ F_{m} = F_i + P(C-1) \]
\[ F_{b} = F_i + C P \]
\[ \sigma_b = \frac{F_b}{A_i} \]
\[ N_{separation} = \frac{F_i}{P(1-C)} \]

Dynamic Loading of Fasteners

\[ P \text{ is a function of time, varying some } P_{min} \text{ and maximum } P_{max} \text{ values, both positive. A very common situation is that of a fluctuating load such as in a bolted pressure vessel that is cycled from zero (} P_{min}=0 \text{) to maximum pressure.} \]

For the general case \( P_{min} > 0, P_{max} > 0 \)

\[ F_{term} = P_{term} \times F_i \]
\[ F_{term} = P_{term} \times F_i \]
\[ F_{term} = (F_{term} \times F_{term})/2 \]
\[ F_{term} = (F_{term} \times F_{term})/2 \]

\[ \text{where,} \]
\[ P_{term} = C P_{max} \]
\[ P_{term} = C P_{min} \]
Dynamic Loading of Fasteners

For the special case (Pmin = 0, Pmax > 0)

\[ P_{b\text{min}} = 0 \text{ (since } P_{\text{min}} = 0) \]
\[ F_{\text{bmin}} = F_{\text{i}} \]
\[ F_{\text{bmax}} = F_{\text{i}} + \frac{(F_{\text{bmax}} - F_{\text{i}})}{2} = \frac{(F_{\text{bmax}} + F_{\text{i}})}{2} \]

\[ \sigma_b = K_{f} \cdot \frac{F_{\text{bmin}}}{A} \]
\[ \sigma_b = K_{f} \cdot \frac{F_{\text{bmax}}}{A} \]

Take \( K_{f} = 1 \) for preloaded fasteners

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- General Thread Nomenclature & Types
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- Fasteners in Shear
  - What is Shear?
  - Straight Direct Shear
  - Eccentric Shear
- Doweled Joints

Doweled Joints

"It is not considered good practice to use bolts or screws in shear to locate and support precision machine parts under shear loads."

Shear can be handled by friction caused by bolts… but, better practice is to use dowels

- Bolts need clearances… at best 2 out of a 4 bolt pattern will bear all of load
Eccentrically Loaded Shear

**Strategy**
- Find Centroid
- Find primary shear $F_1$
- Find secondary shear $F_2$
- Find Moment about centroid
- Find distances from centroid
- Find secondary shear $F_2$ and angles
- Combine $F_1$ and $F_2$
- Decompose to x and y
- Add $F_1$ and $F_2$
- Decompose into $F_{net}$
- Identify Max $F_{net}$, find $\tau$, safety factor

Primary Shear

- $F_{1a}$
- $F_{1b}$
- $F_{1c}$

$|F_1| = \frac{P}{W}$

$n$ – number of dowels

Moment/Secondary Shear

- The moment will add further shear to the dowels, $F_2$
- The moment is centered around the center of gravity of the dowels

$$\tau = \frac{A_1 y_1 + A_2 y_2 + A_3 y_3}{A_1 + A_2 + A_3} \sum_{i=1}^{n} \frac{r_i}{A_i}$$

Finding Forces from Moment

$$M = F_{2A} r_A + F_{2B} r_B + F_{2C} r_C$$

$$F_{2i} = \frac{M r_i}{r_A^2 + r_B^2 + r_C^2 + \cdots} = \frac{M r_i}{\sum_{j=1}^{n} r_j^2}$$

Angles and Vectors

- Force is perpendicular to radial lines
- From known dimensions and trig, calculate $\alpha$

$$y - \tau = \tan(90 - \alpha_A)$$

Remaining Steps

- Decompose $F_{2i}$ into x and y components
- Add x and y components of $F_{2i}$ to $F_{ii}$
- Recompose x and y components into $F_{net,i}$ and determine angle of $F_{net}$
Safety Factor

Calculate safety factor for most heavily loaded dowel

\[ \tau = \frac{F}{A_{\text{shear}}} \]

\[ N = \frac{S_{\text{ty}}}{\tau} = 0.5775 S_{\text{ty}} \]

Strategy Review

- Find Centroid
- Find primary shear \( F_1 \)
- Find secondary shear \( F_2 \)
- Find Moment about centroid
- Find distances from centroid
- Find secondary shear \( F_2 \) and angles
- Combine \( F_1 \) and \( F_2 \)
- Decompose to \( x \) and \( y \)
- Add \( F_1 \) and \( F_2 \)
- Recompose into \( F_{\text{net}} \)
- Identify Max \( F_{\text{net}} \), find \( \tau \), safety factor

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- General Thread Nomenclature & Types
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