Torque tube calculations

Maximum strength/weight incurred by using a hollow tube with maximal outer diameter

Thickness of extension tube to withstand shear

\[
\tau_{\text{max}} \geq \sqrt{\left( \frac{16PLD_2}{\pi(D_2^2-D_1^2)} \right)^2 + \left( \frac{32PLD_2}{\pi(D_2^2-D_1^2)} \right)^2} \quad (\text{rankine criterion})
\]

\[\tau_{\text{max}} = 20,000 \frac{\text{lb}}{\text{in}^2} \quad (\text{maximum shear stress})\]

\[P = 100 \text{lb} \quad (\text{load})\]

\[L = 9.5 \text{in} \quad (\text{extension shaft length})\]

\[L_3 = 10 \text{in} \quad (\text{handle length})\]

\[D_2 = 1.25 \text{in} \quad (\text{extension shaft outer diameter})\]

\[X = 2.0 \quad (\text{safety factor})\]

\[D_1 < 1.018 \text{in} \quad (\text{extension shaft inner diameter})\]

Pin radius

At handle interface:

\[
\tau_{\text{max}} \geq \frac{2T}{D_2N^2\pi} \quad (\text{Rankine criterion})
\]

\[D_2 = 1.25 \text{in} \quad (\text{extension shaft outer diameter})\]

\[T = 1000 \text{in} \cdot \text{lbs} \quad (\text{torque on extension shaft})\]

\[N = 2 \quad (\text{number of pinned junctions})\]

\[\tau_{\text{max}} = 20,000 \frac{\text{lb}}{\text{in}^2} \quad (\text{maximum shear stress})\]

\[X = 2.0 \quad (\text{safety factor})\]

\[r > 0.162 \text{in} \quad (\text{minimum radius of aluminum pin to bear shear})\]

At socket interface:

\[
\tau_{\text{max}} \geq \frac{2T}{D_1N^2\pi} \quad (\text{Rankine criterion})
\]

\[D_1 < 1.018 \text{in} \quad (\text{extension shaft inner diameter})\]

\[T = 1000 \text{in} \cdot \text{lbs} \quad (\text{torque on extension shaft})\]

\[N = 2 \quad (\text{number of pinned junctions})\]

\[\tau_{\text{max}} = 20,000 \frac{\text{lb}}{\text{in}^2} \quad (\text{maximum shear stress})\]

\[X = 2.0 \quad (\text{safety factor})\]

\[r > 0.185 \text{in} \quad (\text{minimum radius of aluminum pin to bear shear})\]

Extension tube & socket tube thicknesses to withstand pin load

\[
\gamma_{\text{max}} = \frac{35000 \frac{\text{lb}}{\text{in}^2}}{N \pi (D_2^2-D_1^2)} \quad (\text{Rankine criterion})
\]

\[\gamma_{\text{max}} = 35000 \frac{\text{lb}}{\text{in}^2} \quad (\text{yield stress})\]

\[T = 1000 \text{in} \cdot \text{lbs} \quad (\text{torque on extension shaft})\]

\[X = 2.0 \quad (\text{safety factor})\]
\[ N = 2 \text{ (number of pinned junctions)} \]

**Extension tube @ handle interface**

\[ r = 0.162\text{in} \]
\[ D_2 = 1.25\text{in} \]
\[ D_1 < 1.074\text{in} \]

**Extension tube @ socket joint**

\[ r = 0.185\text{in} \]
\[ D_2 = 1.25\text{in} \]
\[ D_1 < 1.096\text{in} \]

**Socket tube**

\[ r = 0.185\text{in} \]
\[ D_2 = 1.00\text{in} \text{ (outer diameter of socket tube)} \]
\[ D_1 < 0.846\text{in} \text{ (inner diameter of socket tube)} \]

**Handle Calculations**

Tapered design meets high moment of inertia requirements at shaft end and low shear requirements at grip end.

Handle modelled as a hollow cylinder to calculate rough dimensions.

**Cross section @ shaft end**

\[ \tau_{\text{max}} = \frac{2PL_2}{\pi(D_2^2 - D_1^2)} \] (rankine criterion)
\[ \tau_{\text{max}} = 35000\text{ksi} \text{ (yield strength)} \]
\[ X = 2.0 \text{ (safety factor)} \]
\[ P = 100\text{lb} \text{ (load)} \]
\[ L_2 = 10\text{in} \text{ (handle length)} \]
\[ D_1 = 0.5\text{in} \text{ (inner diameter (stock))} \]
\[ D_2 > 0.868\text{in} \text{ (minimal outer diameter)} \]

**Cross section @ grip end**

\[ \tau_{\text{max}} = \frac{4P}{\pi(D_2^2 - D_1^2)} \] (rankine criterion)
\[ \tau_{\text{max}} = 20,000\text{psig} \text{ (maximum shear stress)} \]
\[ X = 2.0 \text{ (safety factor)} \]
\[ P = 100\text{lb} \text{ (load)} \]
\[ D_1 = 0.5\text{in} \text{ (inner diameter (stock))} \]
\[ D_2 > 0.513\text{in} \text{ (minimal outer diameter)} \]