Torque tube calculations

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

Thickness of extension tube to withstand shear

$\frac{\tau_{max}}{X} \geq \sqrt{\left(\frac{16PL_3D_2}{\pi\left(D_2^4-D_1^4\right)}\right)^2 + \left(\frac{32PLD_2}{\pi\left(D_2^4-D_1^4\right)}\right)^2}$ (rankine criterion)

$\tau_{max} = 20,000\frac{lb}{in^2}$ (maximum shear stress)
$P = 100lb$ (load)
$L = 9.5in$ (extension shaft length)
$L_3 = 10in$ (handle length)
$D_2 = 1.25in$ (extension shaft outer diameter)
$X = 2.0$ (safety factor)
$D_1 < 1.018in$ (extension shaft inner diameter)

Pin radius

At handle interface :

$\frac{\tau_{max}}{X} \geq \frac{2T}{D_2N\pi r^2}$ (Rankine criterion)
$D_2 = 1.25in$ (extension shaft outer diameter)
$T = 1000in \cdot lbs$ (torque on extension shaft)
$N = 2$ (number of pinned junctions)
$\tau_{max} = 20,000\frac{lb}{in^2}$ (maximum shear stress)
$X = 2.0$ (safety factor)
$r > 0.162in$ (minimum radius of aluminum pin to bear shear)

At socket interface :

$\frac{\tau_{max}}{X} \geq \frac{2T}{D_1N\pi r^2}$ (Rankine criterion)
$D_1 < 1.018in$ (extension shaft inner diameter)
$T = 1000in \cdot lbs$ (torque on extension shaft)
$N = 2$ (number of pinned junctions)
$\tau_{max} = 20,000\frac{lb}{in^2}$ (maximum shear stress)
$X = 2.0$ (safety factor)
$r > 0.185in$ (minimum radius of aluminum pin to bear shear)

Extension tube & socket tube thicknesses to withstand pin load

$\frac{\Upsilon_{max}}{X} = \frac{T}{Nr(D_2-D_1)}$ (Rankine criterion)
$\Upsilon_{max} = 35000\frac{lb}{in^2}$ (yeild stress)
$T = 1000in \cdot lbs$ (torque on extension shaft)
$X = 2.0$ (safety factor)

$N = 2$ (number of pinned junctions)

Extension tube @ handle interface

$r = 0.162in$
$D_2 = 1.25in$
$D_1 < 1.074in$

Extension tube @ socket joint

$r = 0.185in$
$D_2 = 1.25in$
$D_1 < 1.096in$

Socket tube

$r = 0.185in$
$D_2 = 1.00in$ (outer diameter of socket tube)
$D_1 < 0.846in$ (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

$\frac{\Upsilon_{max}}{X} = \frac{32PL_2D_2}{\pi(D_2^4 - D_1^4)}$ (rankine criterion)
$\Upsilon_{max} = 35000ksi$ (yield strength)
$X = 2.0$ (safety factor)
$P = 100lb$ (load)
$L_2 = 10in$ (handle length)
$D_1 = 0.5in$ (inner diameter (stock))
$D_2 > 0.868in$ (minimal outer diameter)

Cross section @ grip end

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