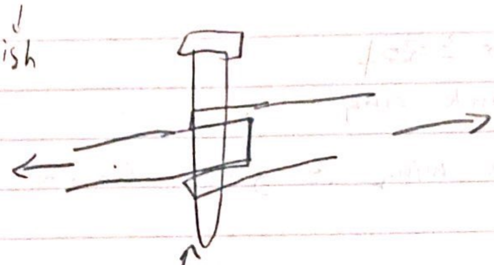
Physics

Tensile strength : pulling
stronger than shear

tensile · 0.6 = shear
ish

$Pa = \frac{N}{m^2}$
Pascals



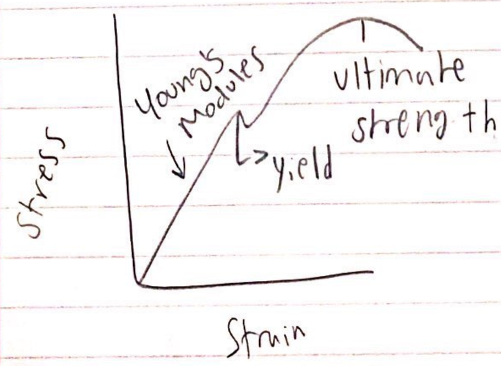
so AN4 - 26A : 140000 tensile

$140000 \cdot 0.6 = 84{,}000$

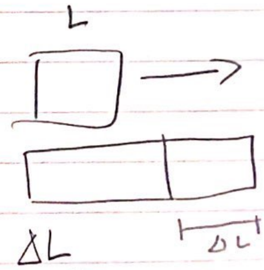
$84000 \cdot \left(\frac{1}{8}\right)^2 \cdot \pi = 4{,}123$

$\boxed{\text{length doesn't matter}}$

Ultimate: how much it can take
Yield : how much it can take w/o diforming



Strain :



$\frac{\Delta L}{L}$

Stress: $\frac{F}{A}$ ← force
← area

allowable bearing stress on the spars

load on each bolt (if have 2) → $\frac{50 \text{ psi}}{\text{bearing area}}$ → $\left(\frac{1}{4}\right)(0.058)$ $= \frac{50}{0.0145}$ $= 3448.3$

bolt diameter, wall thickness

how much each spar hole will take? → 3448.3 each

Aluminum T-6 Yield Tensile Strength: 40000 psi $\cdot \frac{1}{4}(0.058) \cdot 0.6$

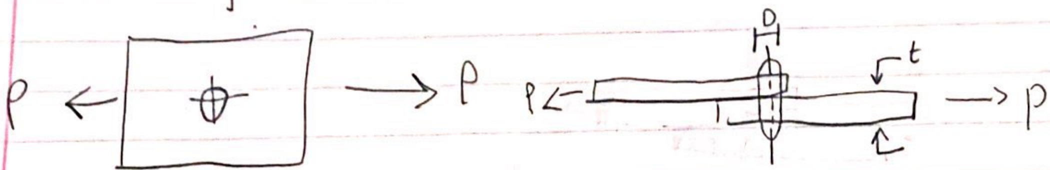
bolt diameter, spar wall thickness, to shear

$= 348$ = how much each spar hole can take

Bearing Stress

When 2 elastic bodies are forced together
- localized on the surface of the material
  - may be very high bc of small areas in contact

Ex bearing stresses in Riveted Connections



P = axial load
D = diameter
t = thickness / width of rectangular area

$$f_{br} = \frac{P}{Dt}$$

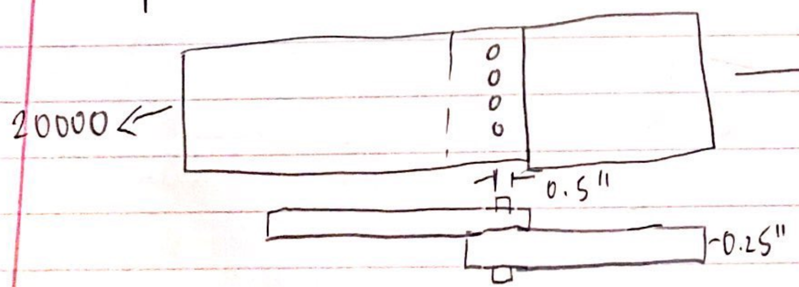
calculated bearing stress (for f_{br})

allowable bearing stress : $P_a = F_{br} Dt$

allowable axial load (for P_a)

allowable bearing stress (for F_{br})

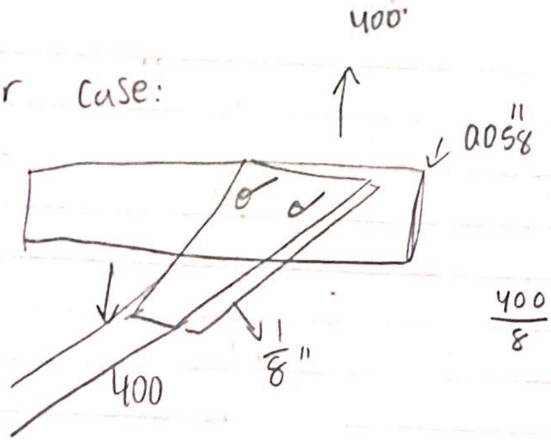
Sample Problem:



load per rivet $\frac{20000}{4}$

So you would do: $f_{br} = \frac{P}{Dt} = \frac{5,000}{(0.5)(0.25)} = 40,000$ psi

So in our case:



$$\frac{400}{8} = 50$$

$$f_{br} = \frac{P}{Dt} \qquad \frac{50}{\left(\frac{1}{4}\right)(0.058)} = 3{,}450 \text{ psi}$$ on each bolt

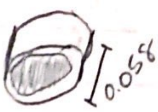
If have 2

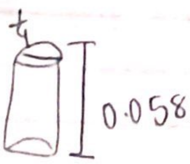
Which is A ok

Spar double shear:

aluminum tensile: 40,000 psi · 0.6 = 24,000

$$24000 \cdot \pi \cdot .25 \cdot 0.058 \cdot \underbrace{2}_{\text{double shear}} = 2186.55$$





$C = \pi d$

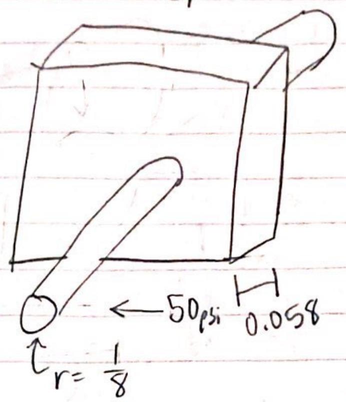
$\frac{\pi}{4} \cdot 0.058$

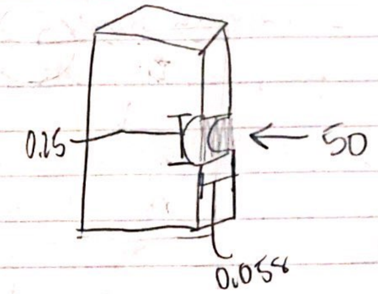
Jk

Note: bearing is compression → contact pressure between 2 seperate bodies
Shear is shearing
(obv)

Sphere & spherial socket

$P = 50$



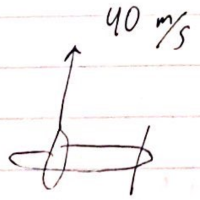


$$\sigma_b = \frac{P}{A} = \frac{P}{td} = \frac{50}{(0.058)(0.25)} = 3448.3 \text{ psi}$$

$A \to 0.058 \cdot 0.25$; $td \to 0.058 \cdot 0.25$

this will be how much if it is 2 bots

4g to psi:



how many force pounds would it

$m \cdot a = \text{force}$

$m \to$ 400 pounds / 181.43 kg

$a \to 40 \frac{m}{s^2}$

$181.43 \cdot 40 \frac{m}{s^2} = 7257.2$ Newtons $= 16315$ lbf

total pound force on plane:

16315 ~~p~~

of spar-strut attachment

Bearing strength @ 4g

but much more on spar so

$424\ lb \cdot 4 = \frac{1700\ lb\ total}{20\ bolts} = 85 \times 2$ just to be safe

weight of plane - wings bc they're generating lift

4g

2 on each strut tube = 8

gusset

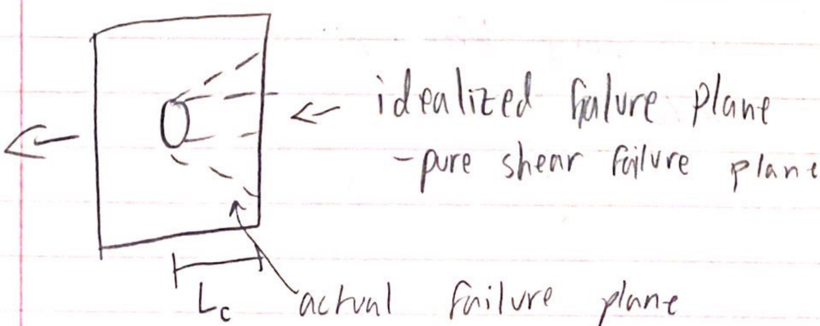
2 on each attachment = 8

1 on each fuselage = 4

$\frac{170\ lb}{d\ t} = 11{,}724$

$d = \frac{1}{4}$, $t = 0.058$

Aluminum Bearing Yield Strength: 56000 psi



L_c = clear distance

t = thickness of part/plane

$R_n = 2(0.6F_u)(L_c)(t)$

↳ fracture shear strength

Salutations comrades
Oink oink
long live proletariate revolution

Column Buckling

- many different ways a beam can fail
  - too much bending moment
  - pushed together & bends & buckled
    ↓
    column buckling -> one of the weakest
    ↓

kip: kilopounds $\frac{(\pi \cdot E \cdot I)}{(k \cdot L)^2}$ → gives you strength of beam - in force (kips)

psi ← E: Young's Modulus of the material we are using
ratio or ending point ← - important bc when the deformation of the
- beam becomes plastic & that's when the beam fails

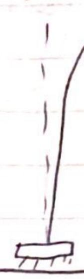
I: area moment of inertia ← smallest
- their area · distance away from center of inertia
- important bc when further away the more it stretches
- closer to center of inertia = easier to deform bc have to stretch less
- ours is: $\frac{\pi(D_o^4 - D_i^4)}{64}$
- units in inches^4

L: length
- longer it is more seseptible it is to bending/buckling
  - inches or whatever everything cancels out in

K = effective length coefficient
- how is the beam attached

Physics sense → - pin-pin is ideal so it makes it 1
- all the length is used in this situation

Fixed - fixed = less rotation

	Pinned-Pinned	Fixed-Fixed	Fixed-Pinned	Fixed-Free
				
Theoretical Effective Length Factor, K:	1	0.5	0.699	2
Recommended Effective Length Factor, K:	1	0.9	0.9	2.1

Axial, Bearing, Shear, Tearout Stress

note: this video uses F but it's interchangable w/ P

Axial - force applied along a longitudinal axis

Stress - intensity of internal force

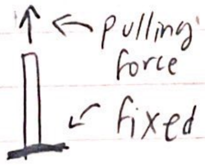
Normal stress - stress is on a plane "normal" to the force; "normal" = perpendicular

$\frac{\text{Stress}}{\text{yeild strength}}$; tells you safety factor ; if ≥ 1 you're j chilin
<1 bruh

axial stress → $\sigma_{ave} = \frac{F}{A}$ ← internal force
sigma ↵ ; A ← cross-sectional Area

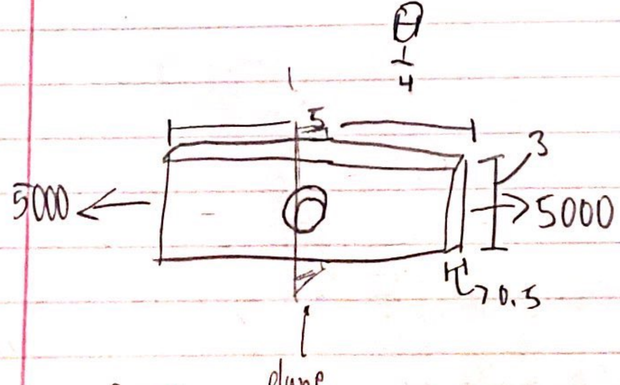
and I quote "put it in solid works & simulate it"

Tensile :



ex.

$$\sigma = \frac{F}{A} = \frac{10\,lb}{\pi\left(\frac{1}{8}\right)^2} = \boxed{203.7\ psi}$$



cross section :

forces ↘

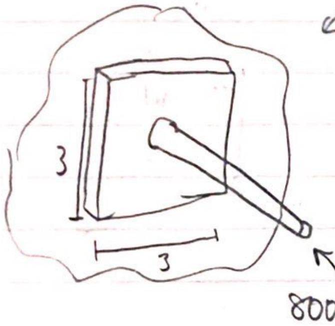
$$\sigma = \frac{2\ (2500)}{2\ (1in \times 0.5in)} = 5000\ psi$$

↑ areas ; ↳ cross sectional area

k = kilo = x1000 = ksi
↳ kips

σ_y = yield (or elastic) strength

Bearing stress // contact stress



← "indistructable"
↳ aka our steel gussets

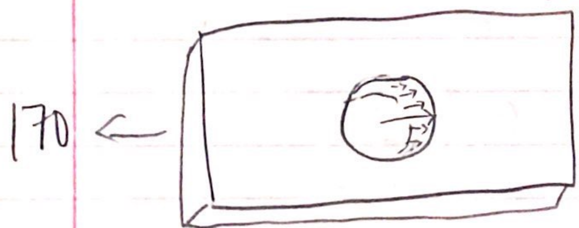
$$\sigma_B = \frac{F}{A}$$

A = bearing surface, contact patch area

must do previous calculations to determine thickness

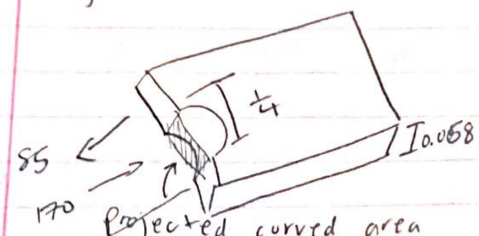
$$\sigma = \frac{8000\ lb}{3in \times 3in} = 888.9\ psi$$

on our example : $\frac{170\ lb}{0.058 \cdot \frac{1}{4}} = 11,724$



↳ because the forces aren't distributed equally, we have to use a projected area

Projected Area



Projected curved area onto a flat plane

$$\sigma_B = \frac{F}{A}$$

$$\sigma_B = \frac{170}{0.058 \cdot \frac{1}{4}} = 11\,724 \text{ psi}$$

0.058 ↓ thickness

bearing strength

So looks like we ~~the~~ just compare this to aluminum's psi which is 50,000

so we j chillin

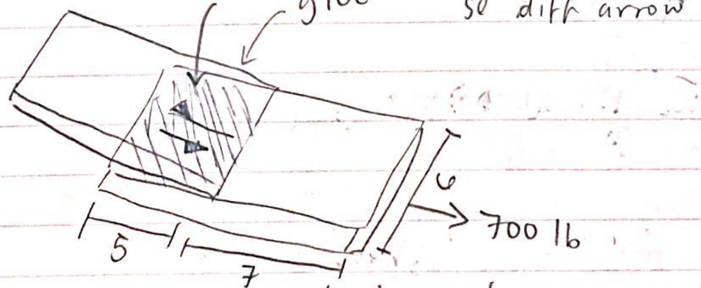
Shear stress

- not an axial stress
- a shear stress - this man is defining the word w/ the word

different type of stress

glue

so diff arrow



- Shear force is not perpendicular to area of stress
- Shear forces create shear stress

$$\tau = \frac{V}{A}$$

V ← Shear force - can be bolts or what not

A ← area that shear force acts on

τ ↑ tau for shear stress

$$\tau = \frac{700}{5\text{in} \cdot 6\text{in}} = 23.3 \text{ psi}$$

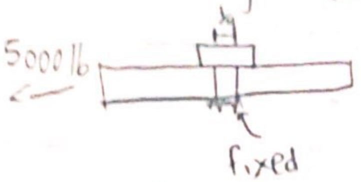
↑ compare to shear strength, not compressive or tensile

super conservative ↓

tensile $\cdot$ 6.5 ≪

Shear Strength cont.

more comonly: ex. side view of pinned bolted plate



V = 5000 lb

$\tau = \frac{V}{A}$ ← 5000 lb

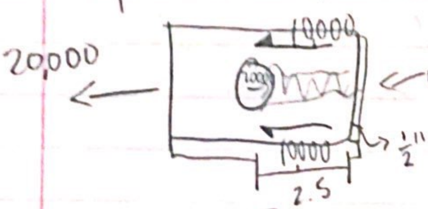
↖ $\frac{\pi d^2}{4}$ ← of bolt

$\frac{5000 \cdot 4}{\pi(0.125)}$

= shear force on bolt

- So check tensile strength of bolt & x.6

Top View of bolt: Tear out



← where this bolt can tear out

$$\tau = \frac{V}{A} = \frac{2(10000)}{2 \cdot (2.5 \cdot 0.5)} = 8000 \text{ psi}$$

- So bolt is ok

tear out is a shear force

What's the difference between shear, bearing & tear out Stress

Bolt stuff:

- according to NASA: bolts should (if designed properly) should generate a clamping force that will carry less than 20% of the external loads on the bolts
  - majority of the work is generated by the compressive energy by the flangs

Variables that effect preload:

- tensile strength - finish - head style
- lubrication → will reduce the torque required to reduce a clamping force i
  - multiplying the standard dry torque by as much as 0.45-1.70 can compensate for lubrication & coatings

FAA says: the amount of clamping force exerted by properly tensioned fastener is normally stated to be 75% of the proof load

Bearing Stress:

- forces acting on the hole a bolt goes through
  - used to determine if there will be deformation of the hole
  - preloading will reduce bearing forces
    - note: not all bolts are preloaded

to calculate: divide force over the contact area between fastener & hole:

$$\sigma_b = \frac{P}{A_b}$$

P ← Axiel forces acting on the fastener / plate

A_b ← bearing area

σ_b ← bearing force

Aircraft	7075 t6	906	4' · 12' · 1/16"
	6061 t6	914	4' · 12' · 1/8"
	4130	805.2	4' · 12' · 1/16"
Local	4130	465	3' 12' · 1/16"

1 · 2 = 33.55
↓ ↓ ↓
4 · 6 = 24

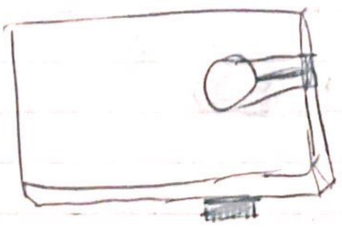
Bearing cont.

- Depending on the clearance btwn hole & fastener.
  - A_b could be length of contact area · diameter of the hole
  - if clearance hole = multiply final $\cdot \frac{\pi}{4}$

But: neither of these take into consideration that stress isn't distributed equally across the diameter of the hole

- adding more fasteners will reduce the bearing stress

Shear Stress:



$$\tau = \frac{V}{A_{shear}}$$

V ← force

τ → shear stress

A_{shear} → area in shear

Direction shear is ideal

- it reduces/eliminated extraneous stresses & material slipage
- even preloaded joints may be subject to excessive loading, impact loads, or creep that would reduce a preload & cause shear w/ bending

Transverse load: pre-derived equasion

$$\tau_{max} = \frac{4V}{3A}$$

- bending stress on a beam can increase tensile & shear stresses on a bolt

Tear Out

- type of Shear
  - instead of shearing the bolt, it's shearing the material surrounding the hole
- rule of thumb: distance from bolt hole to edge should be 12x the material thickness
  - no more than 6 in
- between bolt holes:
  - 24x the thickness of the material
    - no more than 12 in

Margins of Safety:

$$MS = \frac{t_A}{P_b} - 1$$

t_A ← tensile allowed (or shear!)

P_b ← total axial bolt load

MS ← margin of safety

must be at least 0 but more is better

For combined bending & shear:

$$Ms = \frac{1}{\sqrt{R_b^2 + R_\tau^2}} - 1$$

R_τ → $\frac{\text{Safety factor} \cdot \text{force}}{\text{Shear allowable}}$

R_b ↓ $\frac{\text{Total axial bolt load}}{\text{bending allowable}}$

- ask other's what their safety factor is & use bc we are new!!

Scanned with CamScanner

Bolt Preload

- the compression created as the nut is tightened against the bolt (or vice versa)
  - when a bolt is tightened against a material, it allows the bolt to distribute the force through the material
    - so bolts can hold more load when tension is applied

Load: the amount of force acting on a fastener assembly
Preload: the amount of tension (compression) needed to distribute a load's force throughout a fastener assembly
Working load: the load placed on the assembly once ready to perform
Bolt Preload: the tension created when a nut is screwed onto a bolt to hold two materials together

when the bolt preload tension reaches optimal preload → the working load is distributed on the installed material
- bolt doesn't take entire load

Outcomes:
- if loose: load only on bolt & ~~to~~ may be bolt failure
- if tight: load will may only cause bolt deformation

Ways to determine Bolt Preload:

1. Use a torque wrench to reach optimal Torque

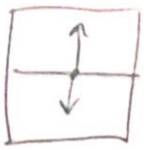
what do we even need to calculate

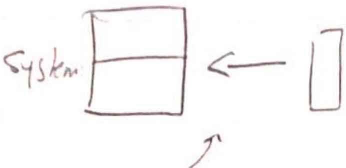
- Standard landing load
- notice during turbulence > equations?
- cross loading - tail
- rudder loading?
- we don't know what we need to know

- bring truss into CFD

External Forces vs. Internal Forces

Internal Forces: forces that are acting between a pair of objects that are both within the system:





External Forces: forces that act on the object within the system by an object outside the system:

$$\vec{F}^{\,total}_{system} = m_{system}\,\vec{a}_{system}$$

System: Block 1, Block 2

notation: $\vec{F}_{12}$ meaning: force exerted by 1 on 2

everything else is internal