Designing of a Hand Operated Letter Press for Sheet Metal

Nicholas Smith, Kyle McNeely

Due: April 24, 2015

1. ABSTRACT

The problem being solved is to design two substantially different hand operated toggle presses that will mark lettering on sheet metal. The press must have an output force of 500 lbs. and it should be designed to prevent failure in static loading due to buckling in link CD and yielding due to bending in lever AB. A force analysis is using statics is done to determine the loading of each member of the press. The stresses from the loads are found and used to determine the geometry according to the specified design factor of five. A buckling analysis is done using Johnson's model for buckling for short columns. The cross-section areas are found for both designs and for both link CD and lever AB, standard sizes are found that meet the minimum requirements of the design. The designs are then compared and evaluated based on size, weight, performance, and cost. Final recommendations are made on how to truly optimize the design of the toggle press.

2. PROBLEM DESCRIPTION

The hand-operated toggle press seen in Figure 1 will be used to mark 1/16 in. lettering on 1018 steel sheets a quarter inch thick at a depth of .004 in. The required force, P, to indent the steel to the optimum depth is 500 pounds using the selected marking stamp (Figure 2) that will produce a character size of 1/16 inch. The operator will need to apply a force, F, by hand to the end of the lever AB at a reasonable load of 15-25 lbs. The press will need to withstand these forces without failing due to buckling in link CD which is a rectangular cross section, as well as prevent failure in the AB circular cross section bar due to bending. This hand-operated toggle press will be designed with a design factor n=5 to ensure a reasonable margin of safety in operation. The toggle press design is restricted so that the minimum operating angle is when angle θ=10° the height H is equal to 0 inches, with a maximum operating angle of 100°. There will be two substantially different designs made, one with cold drawn 1040 steel and the other made with a titanium alloy (Ti-6%Al-4%V). The dimensions of the two presses will be different to maintain the required load and safety considerations.

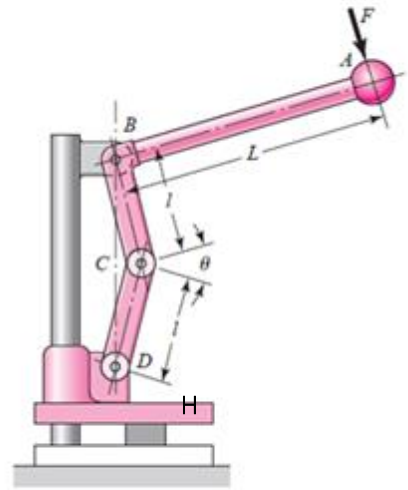


Figure 1 Hand-Operated Toggle Press

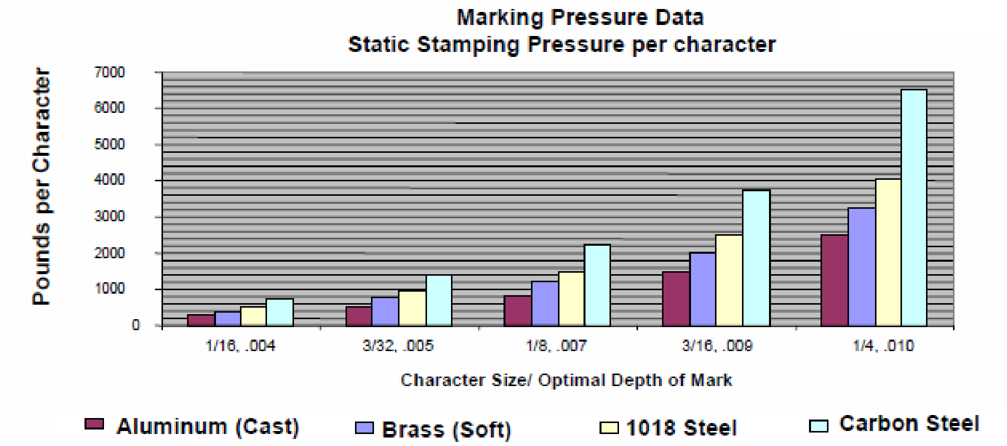


Figure 2 Marking Pressure Data

3. ANALYSIS AND RESULTS

The first step in determining the physical dimensions of the two models is to determine the basic geometry of the angles in the press so a free body diagram can be drawn and used to solve for the static loads in each member, see the Figure 3 for a diagram labeling the angles and members. The link length, *l*, was selected initially to be a size of 3 in. for the steel model and 2.5 in. for the titanium model.

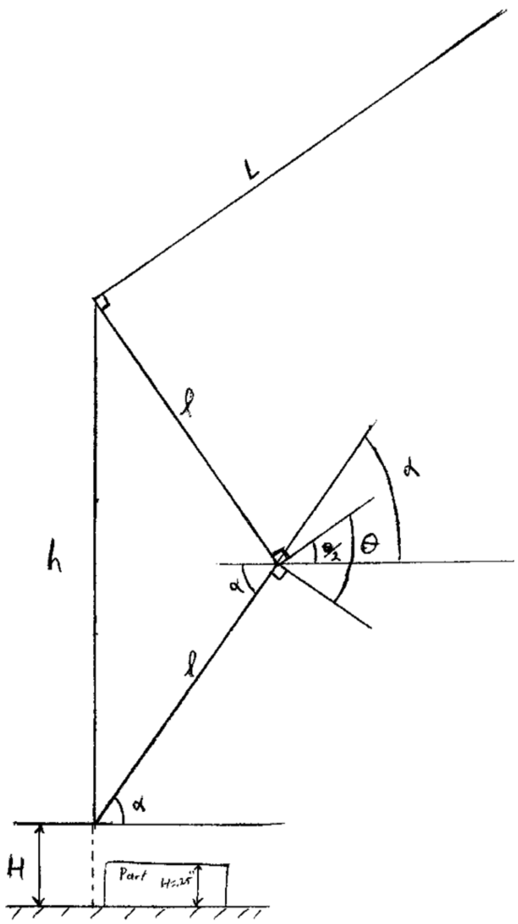


Figure 3 Basic Geometry of Angles

Next, two free body diagrams were drawn, see the Appendix, one for link CD and one for member ABC. From the free body diagram for link CD the reaction force in the direction of the link, C_R is found in Equation 1.

$$C_R = P/\sin(\alpha) \tag{1}$$

Where C_R is the force that will govern the buckling of link CD and α = 90°- $\theta/2$. The critical load for buckling in CD, P_{Cr}, is found by applying the design factor to C_R. The end conditions for link CD in plane is rounded-rounded (C=1) and for out of plane it was fixed-fixed (C=1.2). It was first assumed that buckling would be of the Euler type but after doing the calculations seen in the

Appendix for the limiting slenderness ratio, Equation 2, it was determined that the buckling follows that of Johnson's equation according to the slenderness ratio.

$$(l/k)_1 = \sqrt{\frac{2\pi CE}{Sy}} \tag{2}$$

Where E is the modulus of elasticity and S_y is the yield strength of the material. Putting the geometry found from Euler's equation into the slenderness ratio, l/k Equation 3, it was found to be lower than the limiting slenderness ratio which indicates that it is a short column and that Johnson's model for buckling will take place. Johnson's equation is seen in Equation 4:

$$\frac{l}{k} = \frac{l\sqrt{12}}{t} \tag{3}$$

$$\frac{P_{Cr}}{A} = S_y - \frac{S_y^2}{4\pi^2 CE}\left(\frac{l}{k}\right)^2 \tag{4}$$

Where A is the cross-sectional area of link CD which is a square with dimensions t x t, and k is the radius of gyration which is $k = \sqrt{(I/A)}$ where I is the moment of inertia. After solving Equation 4 for the thickness, t, it was put back into the slenderness ratio for geometry and check again and it was indeed lower than the limiting slenderness ratio meaning Johnson's equation is still valid for this analysis. Using MatLab it was found that both the steel and titanium models followed Johnson's buckling model. The solving Equation 4 for t gives Equation 5, see hand calculations in the Appendix for a detailed solution:

$$t = \sqrt{\frac{C_R * n}{S_y} + \frac{S_y 12 l^2}{4CE\pi^2}} \tag{5}$$

The next concern is the yielding in lever AB due to bending. The lever length *L* is found from statics, particularly summing the moments about point B, given a reasonable force *F* and then *L* is backed out, *L* is given by Equation 6, see Appendix for detailed calculations.

$$L = \frac{\boldsymbol{C_R l cos(\alpha) cos\left(\frac{\theta}{2}\right)}}{\boldsymbol{F}} \tag{6}$$

For the first design using steel the force, *F*, was given and *L* was found and for the second design using titanium *L* was given and *F* was found by simply rearranging Equation 6. For bending the lever is treated as a cylindrical beam fixed at one end with a tip load, *F*, applied. The Appendix has a diagram, hand calculations, and a stress element drawn for the stress due to bending neglecting transverse shear. Equation 7 is the stress due to bending:

$$\sigma_x = \frac{Mc}{I} = \frac{4FL}{\pi r^3} \tag{7}$$

Where *M* is the moment about B and *c* is the distance from the neutral axis of the beam to the surface with the maximum stress. With the factor of safety applied the value of *r*, the radius of the beam AB, is found in Equation 8.

$$r = \frac{4nFL}{\pi S_y} \tag{8}$$

This stress analysis is done for both designs and the diameters were found from doubling the radius. Below in Table 1 is a comparison between the two designs, the thickness *t* and the diameter *d* have been rounded to the nearest standard size available. The table includes both the slenderness ratio and the limiting slenderness ratio for each design and for in plane and for out of plane.

	Design 1 1040 CD Steel	Design 2 Titanium Alloy Ti-6%Al-4%V
E(psi)	30000000	16500000
Sy(psi)	71000	128000
l	3	2.5
H	0.25	0.25
L(in)	30	16
F(lbs)	15	25
(l/k)1 in plane	91.3264	50.44311
t (in) in plane (C=1)	0.208238	0.187615
l/k in plane	49.90588	46.15976
(l/k)1 out of plane	100.0431	55.25766
t (in) out of plane (C=1.2)	0.205631	0.18095
l/k out of plane	50.53866	47.85983
d (in)	0.68454	0.562447
Standard thickness link CD ,t, (in)	.25	.2
Standard Diameter for Lever AB (in)	.8	.6

Table 1 Results

Table 2 shows the breakdown of the total cost. The density of the 1040 steel being used is 0.2834 lb/in^3. With the price of it being about 30 cents/pound. The following data was calculated with Table 2, with AB, twice of CD for the two symmetric links being used as the volume to find the cost of the material: For the basic design, the price of the material actually used will be $1.31, with the required being 95 cents. The use of standard sized bars and rods increased the cost by 35 cents. The factor of safety was increased to 6.9 as the lowest for in plane failure

For the advance design: the density for titanium (Ti-6%Al-4%V) is 0.159683 lb/in^3 (7). With the price of it being $8.62/pound. With the required material cost being $5.33, and the actual being $6.50, a 0.79 price change to add standard sizing. The factor of safety is also increased to 6.9 using the new sizes.

With the increase in the factor of safety further changes could be made to have standard sizing of the lengths changed to either increase or decrease the force on the arm. For the length of CD in the titanium design we did change it to be 2.5 inches .5 inches smaller than the steel model. The titanium design comes out to be more expensive but decreases the overall size of the press. Reducing the length of the arm from 30 inches to 16 inches, but also requires 10 more pounds of force.

CD

		b, in	h, in	Area, in^2	Volume, in	Cost, $
Basic Design	Required	0.2082	0.2082	0.04334724	0.13004172	0.022112
	Actual used	0.24	0.24	0.0576	0.1728	0.029383
Advance Design	Required	0.1876	0.1876	0.03519376	0.0879844	0.242215
	Actual used	0.2	0.2	0.04	0.1	0.275293

AB

		d=2r in	Area, in^2	Volume, in^3	Cost, $	Total, $
Basic Design	Required	0.6846	0.368098181	10.97377977	0.93299076	0.955103
	Actual used	0.8	0.502654825	15.07964474	1.2820714	1.311454
Advance Design	Required	0.5424	0.23106238	3.696998086	5.08879757	5.331013
	Actual used	0.6	0.282743339	4.523893421	6.22699209	6.502286

Cross section area sizes and price/lb Table 2.

4. CONCLUSIONS

Two presses were designed, made of two different materials to accomplish the same goal, to mark 1/16 in. lettering on a .25 in. thick piece of 1018 steel. The first design is a basic design that is cheaply made and the second design is an advanced design that makes the press smaller. A comparison of standard sizing was done as well to see if it would be recommended to use off the shelf parts for the press and be reasonable with pricing of the material. Standard sizes are used since the cost of machining to the exact dimensions is high. The required design for the press and the actual specifications add an even greater factor of safety and cost to build the press. The titanium design while it would be able to produce smaller presses, the price of the material vs. what it offers is too high. For a truly optimal design it is suggested to change the cross sectional area of the link CD to be a rectangle rather than a square to cut down on expenses. The shorter the column the less likely it is to fail due to buckling and since the range of motion needed is small, the length of the link can be reduced. The longer the lever is the easier it is to operate the machine but at a higher cost so it is recommended to start with a desired operating force and back out the geometry. To optimize the design further a cheaper material of slightly less strength could be used and still be within the factor of safety specified.

REFERENCES

1. Budynas & Nisbett Shigley's Mechanical Engineering Design 9th, chapter 4 page 210
2. http://www.columbiamt.com/CMT-Main/Catalogs/stamp_and_die_complete_web_3-07.pdf, sheet specs
3. http://www.azom.com/article.aspx?ArticleID=6115, 1018 steel specs
4. http://www.azom.com/article.aspx?ArticleID=6525, 1040 steel specs
5. http://www.meps.co.uk/World%20Carbon%20Price.htm, 1040 steel price
6. http://www.metalprices.com/p/TitaniumFreeChart, titanium price
7. http://www.azom.com/article.aspx?ArticleID=1341, titanium specs

APPENDIX

Table 2 excel file can be available upon request.

Nicholas Smith, smith.1517@wright.edu

Table A–5 †Often preferred.

Physical Constants of Materials

Material	Modulus of Elasticity E		Modulus of Rigidity G		Poisson's Ratio ν	Unit Weight w		
	Mpsi	GPa	Mpsi	GPa		lbf/in^3	lbf/ft^3	kN/m^3
Aluminum (all alloys)	10.4	71.7	3.9	26.9	0.333	0.098	169	26.6
Beryllium copper	18.0	124.0	7.0	48.3	0.285	0.297	513	80.6
Brass	15.4	106.0	5.82	40.1	0.324	0.309	534	83.8
Carbon steel	30.0	207.0	11.5	79.3	0.292	0.282	487	76.5

Modulus of Elasticity, E for the steel used in the press Table 3.

Table A-17

Preferred Sizes and Renard (R-Series) Numbers
(When a choice can be made, use one of these sizes; however, not all parts or items are available in all the sizes shown in the table.)

Fraction of Inches

$\frac{1}{64}$, $\frac{1}{32}$, $\frac{1}{16}$, $\frac{3}{32}$, $\frac{1}{8}$, $\frac{5}{32}$, $\frac{3}{16}$, $\frac{1}{4}$, $\frac{5}{16}$, $\frac{3}{8}$, $\frac{7}{16}$, $\frac{1}{2}$, $\frac{9}{16}$, $\frac{5}{8}$, $\frac{11}{16}$, $\frac{3}{4}$, $\frac{7}{8}$, 1, $1\frac{1}{4}$, $1\frac{1}{2}$, $1\frac{3}{4}$, 2, $2\frac{1}{4}$, $2\frac{1}{2}$, $2\frac{3}{4}$, 3, $3\frac{1}{4}$, $3\frac{1}{2}$, $3\frac{3}{4}$, 4, $4\frac{1}{4}$, $4\frac{1}{2}$, $4\frac{3}{4}$, 5, $5\frac{1}{4}$, $5\frac{1}{2}$, $5\frac{3}{4}$, 6, $6\frac{1}{2}$, 7, $7\frac{1}{2}$, 8, $8\frac{1}{2}$, 9, $9\frac{1}{2}$, 10, $10\frac{1}{2}$, 11, $11\frac{1}{2}$, 12, $12\frac{1}{2}$, 13, $13\frac{1}{2}$, 14, $14\frac{1}{2}$, 15, $15\frac{1}{2}$, 16, $16\frac{1}{2}$, 17, $17\frac{1}{2}$, 18, $18\frac{1}{2}$, 19, $19\frac{1}{2}$, 20

Decimal Inches

0.010, 0.012, 0.016, 0.020, 0.025, 0.032, 0.040, 0.05, 0.06, 0.08, 0.10, 0.12, 0.16, 0.20, 0.24, 0.30, 0.40, 0.50, 0.60, 0.80, 1.00, 1.20, 1.40, 1.60, 1.80, 2.0, 2.4, 2.6, 2.8, 3.0, 3.2, 3.4, 3.6, 3.8, 4.0, 4.2, 4.4, 4.6, 4.8, 5.0, 5.2, 5.4, 5.6, 5.8, 6.0, 7.0, 7.5, 8.5, 9.0, 9.5, 10.0, 10.5, 11.0, 11.5, 12.0, 12.5, 13.0, 13.5, 14.0, 14.5, 15.0, 15.5, 16.0, 16.5, 17.0, 17.5, 18.0, 18.5, 19.0, 19.5, 20

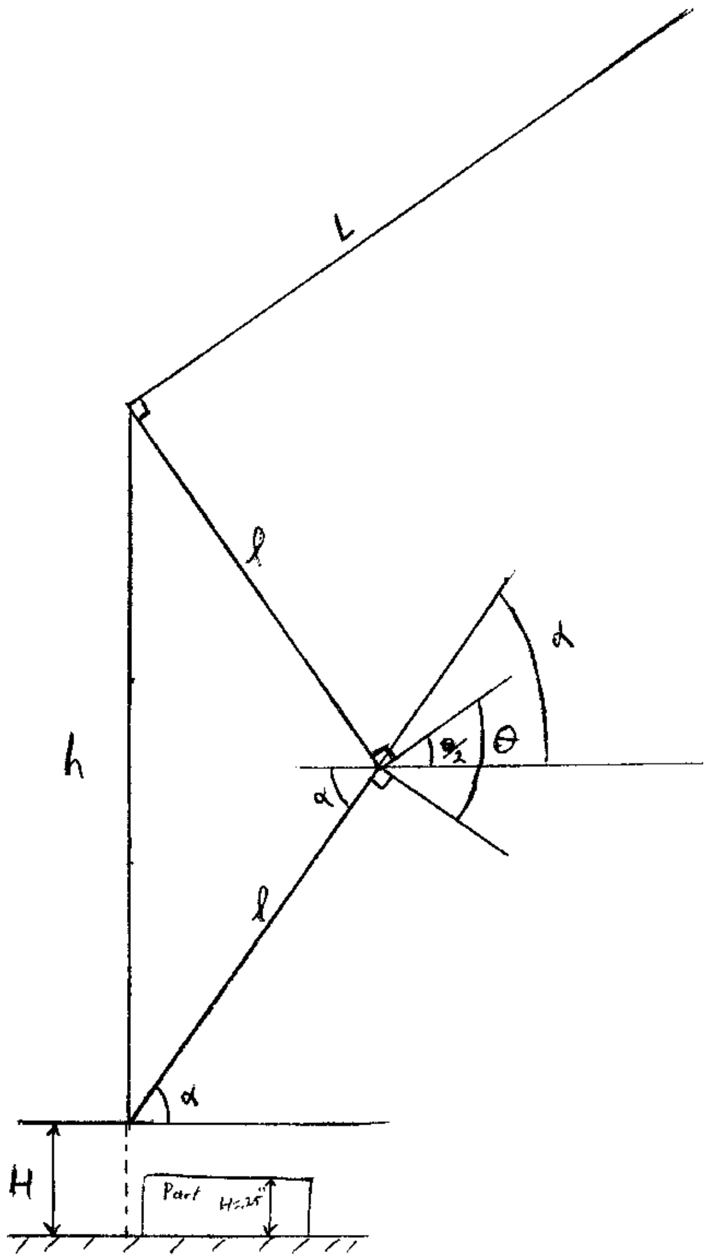
Preferred sizes: Table 4.

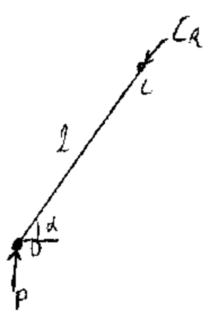
Table A-20

Deterministic ASTM Minimum Tensile and Yield Strengths for Some Hot-Rolled (HR) and Cold-Drawn (CD) Steels [The strengths listed are estimated ASTM minimum values in the size range 18 to 32 mm ($\frac{3}{4}$ to $1\frac{1}{4}$ in). These strengths are suitable for use with the design factor defined in Sec. 1–10, provided the materials conform to ASTM A6 or A568 requirements or are required in the purchase specifications. Remember that a numbering system is not a specification.] *Source:* 1986 SAE Handbook, p. 2.15.

1 UNS No.	2 SAE and/or AISI No.	3 Process-ing	4 Tensile Strength, MPa (kpsi)	5 Yield Strength, MPa (kpsi)	6 Elongation in 2 in, %	7 Reduction in Area, %	8 Brinell Hardness
G10060	1006	HR	300 (43)	170 (24)	30	55	86
		CD	330 (48)	280 (41)	20	45	95
G10100	1010	HR	320 (47)	180 (26)	28	50	95
		CD	370 (53)	300 (44)	20	40	105
G10150	1015	HR	340 (50)	190 (27.5)	28	50	101
		CD	390 (56)	320 (47)	18	40	111
G10180	1018	HR	400 (58)	220 (32)	25	50	116
		CD	440 (64)	370 (54)	15	40	126
G10200	1020	HR	380 (55)	210 (30)	25	50	111
		CD	470 (68)	390 (57)	15	40	131
G10300	1030	HR	470 (68)	260 (37.5)	20	42	137
		CD	520 (76)	440 (64)	12	35	149
G10350	1035	HR	500 (72)	270 (39.5)	18	40	143
		CD	550 (80)	460 (67)	12	35	163
G10400	1040	HR	520 (76)	290 (42)	18	40	149
		CD	590 (85)	490 (71)	12	35	170

Yield Strength: Table 5.

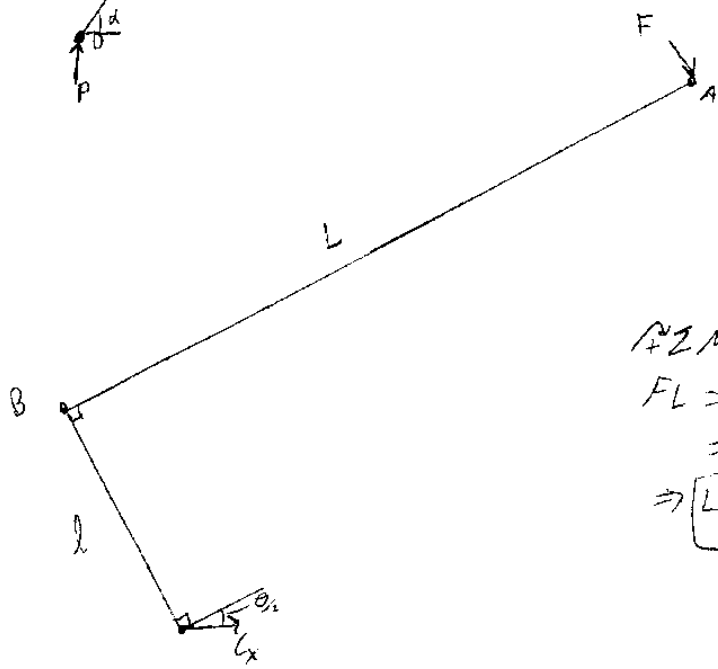

L
l
α
h
θ/2
θ
α
l
α
H
Part
H=.25"



$C_y = C_R \sin\alpha = P$

$C_x = C_A \cos\alpha$

$C_A = P/\sin\alpha$



$\circlearrowleft + \sum M_B = 0 = FL - l C_x \cos\theta/2$

$FL = l C_x \cos\theta/2$

$= l(C_A \cos\alpha)\cos\theta/2$

$\Rightarrow \boxed{L = \dfrac{l C_A \cos\alpha \cos\theta/2}{F}}$

$\alpha = 180^\circ - (\theta/2 + 90^\circ) = 90^\circ - \theta/2$

Law of Cosines:

$h^2 = l^2 + l^2 - 2 l l \cos(2\alpha) = 2l^2(1 - \cos(180^\circ - \theta))$

$h = l\sqrt{2(1-\cos(180^\circ-\theta))}$

$H = 0$ @ $\theta = 10^\circ$

$H = l\sqrt{2(1-\cos(180^\circ-10^\circ))} - l\sqrt{2(1-\cos(180^\circ-\theta))}$

$H/l - \sqrt{2(1-\cos 170^\circ)} = \sqrt{2(1-\cos(180^\circ-\theta))}$

$\left(H/l - \sqrt{2(1-\cos 170^\circ)}\right)^2 = 2(1-\cos(180^\circ-\theta))$

$1/2\left(H/l - \sqrt{2(1-\cos 170^\circ)}\right)^2 = 1 - \cos(180^\circ-\theta)$

$\cos(180^\circ-\theta) = 1 - 1/2\left(H/l - \sqrt{2(1-\cos 170^\circ)}\right)^2$

$180^\circ - \theta = \cos^{-1}\left[1 - 1/2\left(\frac{H}{l} - \sqrt{2(1-\cos 170^\circ)}\right)^2\right]$

$$\boxed{\theta = 180^\circ - \cos^{-1}\left[1 - 1/2\left(H/l - \sqrt{2(1-\cos 170^\circ)}\right)^2\right]}$$

To simplify use a square bar for link CD:

$A = t^2$ $I_{in\ plane} = \frac{t^4}{12} = I_{out\ plane}$ $E = 30E6$ psi 1040 CD $S_y = 71E3$ psi

1020 CD $S_y = 57E3$ psi

1040 CD: in plane $(l/k)_1 = \sqrt{\frac{2\pi^2(1)(30E6)}{71E3}} = 91.3264$

out of plane $(l/k)_1 = \sqrt{\frac{2\pi^2(1.2)(30E6)}{71E3}} = 100.0431$ } material

Choose Euler: $P_{cr} = C_R \cdot n = \frac{C\pi^2 EI}{l^2} \Rightarrow I = \frac{C_R \cdot n\, l^2}{C\pi^2 E} = \frac{t^4}{12}$

$t = \left(\frac{12 C_R \cdot n \cdot l^2}{C\pi^2 E}\right)^{1/4}$ $t_{in\ plane} = .1758''$

$t_{out\ of\ plane} = .1680''$

geometry: $\frac{l}{k} = \frac{l}{\sqrt{I/A}} = \frac{l}{\sqrt{\frac{t^4/12}{t^2}}} = \frac{l}{\sqrt{t^2/12}} = \frac{l\sqrt{12}}{t} = 59.11 < 91.3264$ use Johnson

Johnson:

$\frac{P_{cr}}{A} = S_y - \frac{1}{CE}\left(\frac{S_y l}{2\pi k}\right)^2 = \frac{C_R \cdot n}{t^2} = S_y - \frac{1}{CE}\left(\frac{S_y l}{2\pi t/\sqrt{12}}\right)^2$

$\left(\frac{C_R \cdot n}{t^2} = S_y - \frac{1}{CE}\frac{S_y^2 l^2 12}{4\pi^2 t^2}\right)t^2$ $C_R n = S_y t^2 - \frac{S_y^2 l^2 12}{CE 4\pi^2}$

$S_y t^2 = C_R \cdot n + \frac{S_y^2 l^2 12}{CE 4\pi^2}$ $t = \left[\frac{C_R \cdot n}{S_y} + \frac{S_y l^2 12}{CE 4\pi^2}\right]^{1/2}$

$\Rightarrow$ $t_{in\ plane} = .2082''$

$t_{out\ of\ plane} = .2056''$ $\frac{l}{k} = \frac{l\sqrt{12}}{t} = 50.57$

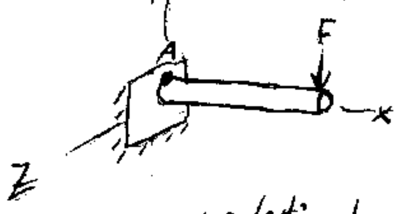
Standard size: .25''

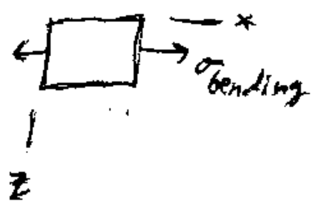
factor of safety: $\frac{P_{cr}}{P} \Rightarrow P_{cr} = S_y A - \frac{A}{CE}\left(\frac{S_y l}{2\pi k}\right)^2 = 3977.8$ lbs.

$P = C_R = 523.8$ $\frac{3977.8 \text{ lbs}}{523.8 \text{ lbs}} = 7.594 = n$

Bending in AB:

Treat lever AB as a cylindrical beam, cantilevered at B





$$\sigma_x = \frac{Mc}{I} = \frac{(FL)\,r}{\frac{\pi (2r)^4}{64}} = \frac{4FL}{\pi r^3}$$

neglecting transverse shear

$$n = \frac{S_y}{\sigma_x} \qquad \frac{5 \cdot FL4}{\pi r^3} = S_y$$

$$r^3 = \frac{20FL}{\pi S_y}$$

$$r = \left(\frac{20FL}{\pi S_y}\right)^{1/3} = .34298''$$

$\Rightarrow d = 2r = .6845''$

Standard size: $d = .75''$

$$n = \frac{S_y}{\sigma_x} = \frac{71E3}{\frac{FL\,(.75/2)}{\frac{\pi (.75)^4}{64}}} = \frac{71E3}{10864} = 6.535$$

```
clc
close all
clear all

P=500;  %press force in lbs
F=15;   %force applied to lever by hand
H=.25;  %hieght for part being stamped
l=3;    %link CD inches design1
l2=2.5  %design2

%% Statics
Theta=(pi-acos(1-.5*(H/l-sqrt(2*(1-cos(170*pi/180))))^2))*180/pi     %degrees
a=(90-Theta/2) %from geometry
Cr=P/sind(a)   %from statics
L=Cr*l*cosd(a)*cosd(Theta/2)/F %lever length, inches
Cy=Cr*sind(a)                 %from statics,(Cy=P)
Cx=F*L/(l*cosd(Theta/2))      %from statics
L2=16  %design2
Theta2=(pi-acos(1-.5*(H/l2-sqrt(2*(1-cos(170*pi/180))))^2))*180/pi %design2
a2=(90-Theta2/2) %Design 2
F2=Cr*l2*cosd(a2)*cosd(Theta2/2)/L2 %design 2

%% Buckling in CD
C=[1,1.2]; %[in plane, out of plane]
E=30e6;%psi
E2=16500e3;%psi design 2
Sy=[71e3,128e3]; %Yield strength in psi [1040CD steel, Ti-6Al-4V(Grade 5), Annealed]
l_k_1=sqrt(2*pi^2.*C.*E./Sy(1)) %1040 design 1
l_k_1_2=sqrt(2*pi^2.*C.*E2./Sy(2)) %Ti design 2
n=5; %design factor
t=(Cr*n/Sy(1)+Sy(1)*l^2*12./(C*E*4*pi^2)).^.5 %[in plane, out of plane] found from Johnson design 1
l_k=l*sqrt(12)./t           %design1
t2=(Cr*n/Sy(2)+Sy(2)*l2^2*12./(C*E2*4*pi^2)).^.5     %design2
l_k2=l2*sqrt(12)./t2        %design2
%% Bending in AB

M=F*L; %Design 1
r=(n*4*M/(pi*Sy(1)))^(1/3)
d=r*2
M2=F2*L2; %Design 2
r2=(n*4*M2/(pi*Sy(2)))^(1/3)
d2=r2*2

%real
t_real=[.24,.2];
r_real=[.4, .3];
n_CD_real_s=(t_real(1).^2-Sy(1)*l^2*12./(C*E*4*pi^2))*Sy(1)/Cr
n_AB_real_s=r_real(1).^3*(pi*Sy(1))/(4*M)

n_CD_real_t=(t_real(2).^2-Sy(2)*l^2*12./(C*E*4*pi^2))*Sy(2)/Cr
n_AB_real_t=r_real(2).^3*(pi*Sy(2))/(4*M2)


%% Results
%Design 1:
D1={'E(psi)',E; 'Sy(psi)',Sy(1); 'l',l;'H',H;'L(in)',round(L); 'F(lbs)', F
%Design 2:
D2={'E(psi)',E2; 'Sy(psi)',Sy(2); 'l',l2;'H',H;'L(in)',round(L2); 'F(lbs)'
```

```
D1 =

    'E(psi)'                    [30000000]
    'Sy(psi)'                   [   71000]
    'l'                         [       3]
    'H'                         [  0.2500]
    'L(in)'                     [      30]
    'F(lbs)'                    [      15]
    '(l/k)1 in plane'           [ 91.3264]
    't (in) in plane ...'       [  0.2082]
    'l/k in plane'              [ 49.9059]
    '(l/k)1 out of plane'       [100.0431]
    't (in) out of pl...'       [  0.2056]
    'l/k out of plane'          [ 50.5387]
    'd (in)'                    [  0.6845]


D2 =

    'E(psi)'                    [16500000]
    'Sy(psi)'                   [  128000]
    'l'                         [  2.5000]
    'H'                         [  0.2500]
    'L(in)'                     [      16]
    'F(lbs)'                    [ 25.0604]
    '(l/k)1 in plane'           [ 50.4431]
    't (in) in plane ...'       [  0.1876]
    'l/k in plane'              [ 46.1598]
    '(l/k)1 out of plane'       [ 55.2577]
    't (in) out of pl...'       [  0.1810]
    'l/k out of plane'          [ 47.8598]
    'd (in)'                    [  0.5424]
```