Title
82"" HYDROGEN BUBBLE CHAMBER BELLOW

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LAWRENCE RADIATION LABORATORY - UNIVERSITY OF CALIFORNIA ENGINEERING NOTE		CODE HD0200	SERIAL M4009X	PAGE 1 OF 47
AUTHOR R.A. BYRNS	DEPARTMENT M.E.	LOCATION B	DATE JAN 24, 1968	

PROGRAM—PROJECT—JOB
82" HYDROGEN BUBBLE CHAMBER
BELLOWS

TITLE
MISC. NOTES

INDEX

THESE NOTES ARE A MISCELLANEOUS COLLECTION OF RUFF, CRUDE, UNEDITED, UNCORRECTED, UNSUMMARIZED NOTES COLLECTED OVER A HECTIC BELLOWS DEVELOPMENT PERIOD OF 2 YEARS. RB.

LAWRENCE RADIATION LABORATORY - UNIVERSITY OF CALIFORNIA

ENGINEERING NOTE

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AUTHOR	DEPARTMENT	LOCATION	DATE
R. Byrns	M.E.	Berk	April 11, 1966

From EN 4312-03 M4. p.5.

2 dimensional analysis – 1st approximation – {ignore hoop stress

IF: $b = 1.500''$ $h = .035''$ $2\delta = 0.300''$ $\delta = 0.150''$

$p = 180\,psi$

$$S = \frac{Pb}{h} + \frac{Eh\delta}{2\pi b^2}$$

$$S = \frac{180(1.5)}{3.5\times10^{-2}} + \frac{(30\times10^{6})(3.5\times10^{-2})(1.5\times10^{-1})}{2\pi(1.5)^2} = 7720 + \frac{157\times10^{3}}{14.1}$$

$$S = 7{,}720 \pm 11{,}100 = \boxed{\boxed{18{,}800\,psi}\ \text{2 DIM}} \quad (\text{or } 3380)$$

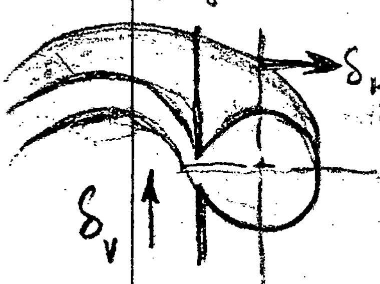
IF $h = .050''$

$$S = \frac{35}{50}(7720) + \frac{50}{35}(11{,}100) = 5{,}400 \pm 15{,}850 = \boxed{21{,}250\,psi\ \text{2 DIM}}$$

(10,450)

$h = .025$ – $S = 2700 \pm 31{,}700 = \boxed{34400\,psi}$

page 6. error due 2 dim. analysis –



$$\delta_H = \frac{\delta_V}{2\pi} \qquad \delta_V = 0.150'' \qquad \delta_H = \frac{\delta_V}{2\pi} = .024''$$

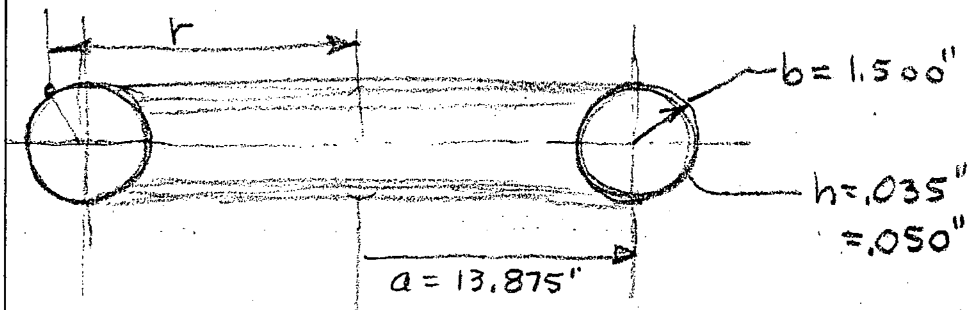
Hoop strain, $e = \frac{\Delta L}{L} = \frac{\Delta C}{C} = \frac{\Delta R}{R}$

(for circular section)
not flat sides

$$S_h = Ee = (30\times10^{6})\left(\frac{.024}{15}\right) = (30\cdot10^{6})(1.6\times10^{-3})$$

$S_{Hoop} = 48{,}000$ psi FOR $\delta_V = 0.150''$

THIS CONDITION APPLIES ONLY TO THE CURVED ENDS!
WHERE THEY HAVE RADIAL RESTRAINT DUE HOOP STRESS,
IN FLAT SECTIONS – NO HOOP STRESS.
(M. MALKUS SED ON PHONE $S \simeq 54{,}000$ psi)



$E = 30 \times 10^6$
$\nu = 0.26$
$\delta_v = 0.150''$
$p = 180$ psi

from p. 8 – 3 dimensional analysis (meridional stresses are highest)

Deflection stress, 5.0×10^{-2} – – – 50,000 ······ 116 ······ 4.87

max meridional (bending) 3.5×10^{-2}

$$\sigma_{\phi B} = 0.1374\, E \delta_v (1 - \nu^2)^{-2/3} \left(\frac{h}{a^2 b^2}\right)^{1/3} = \left[\frac{35{,}000}{432 \times 10^6}\right]^{1/3} = \left[\frac{81}{10^6}\right]^{1/3} = \frac{4.32}{10^2}$$

$(1-\nu^2)$: .0675, .932; a^2b^2: (192.)(2.25) = 432

$= 0.137 (30 \times 10^6)(.150) = (4.12 \times 10^6)(.150) = (.617 \times 10^6)$

$$\frac{1}{(932 \times 10^{-3})^{2/3}} = \frac{1}{(869 \times 10^{-3})^{1/3}} = \frac{1}{9.54 \times 10^{-1}} = 1.050$$

$$\sigma_{\phi B} = [.617 \times 10^6](1.050)(4.32 \times 10^{-2}) = 2.80 \times 10^4 = \boxed{28{,}000 \text{ psi}}\ (.035)$$

$\boxed{31{,}600 \text{ psi}}$ (.050)

Max. Hoop stress.

$$\sigma_{\theta D} = .0988\, E \delta_v (1 - \nu^2)^{-1/6} \left(\frac{h}{a^2 b^2}\right)^{1/3}$$

$$= (.444 \times 10^6) \frac{1}{0.988} (4.32 \times 10^{-2}) = 1.94 \times 10^4 = \boxed{19{,}400 \text{ psi}}\ (.035)$$

$\boxed{21{,}800 \text{ psi}}$ (.050)

max. merid. stress (B) occurs @

$$\sin\theta = 1.225 \left(\frac{ah}{b^2}\right)^{1/3} \left[12(1 - \nu^2)\right]^{-1/6}$$

$$\frac{1}{12^{1/6}(0.988)} = \frac{1}{1.495} = .668$$

$12^{1/6} = 1.513$

$$\left\{\frac{(13.88)(3.5 \times 10^{-2})}{2.25}\right\}^{1/3} = (216 \times 10^{-3})^{1/3}$$

306×10^{-3}

0.600 for .035"

0.676 for .050

$\sin\theta = 1.225\,(.668)(.600) = .491 \qquad \theta = 29.4^\circ \; (.035'')$

$(.676) = .553 \qquad \theta = 33.6^\circ \; (.050'')$

Pressure Stresses

Hoop.

$$\sigma_{\theta D} = \frac{pb}{2h} = \frac{180\,(1.5)}{7\times10^{-2}} = \boxed{\begin{array}{c} 3850 \text{ psi} \\ 2700 \text{ psi} \end{array}} \begin{array}{c}(.035)\\(.050)\end{array}$$

Meridian

$$r = a - b\sin\theta = 13{,}880 - 1.5\,(.491) = 13{,}144$$

$$\qquad\qquad (.553) = .829 \;\; 13{,}050$$

736

$$\sigma_{\phi D} = \frac{pb}{h}\left(\frac{r+a}{2r}\right) = 7710\left[\frac{27{,}024}{26{,}288}\right] = \boxed{\begin{array}{c} 7940 \\ 5580 \end{array}} \begin{array}{c}.035\\.050\end{array}$$

$$5400\left(\frac{26{,}930}{26{,}100}\right)$$

3 DIM

Max. meridian stress (total)	.035"	28,000 + 7940 =	35,940 psi
$\Delta\sigma \simeq \frac{1}{30} \simeq 4\%$	.050"	31,600 + 5580 =	37,180 psi
Max Hoop. stress (total)	.035"	19400 + 3850 =	23,250 psi
$\Delta\sigma \simeq \frac{1}{25} \simeq 4\%$	.050"	21,800 + 2700 =	24,500 psi

above stresses at ambient temp. during test

where $\delta_v = \pm 0.150 = \sum\delta = 0.300''$

$p = 180$ psi

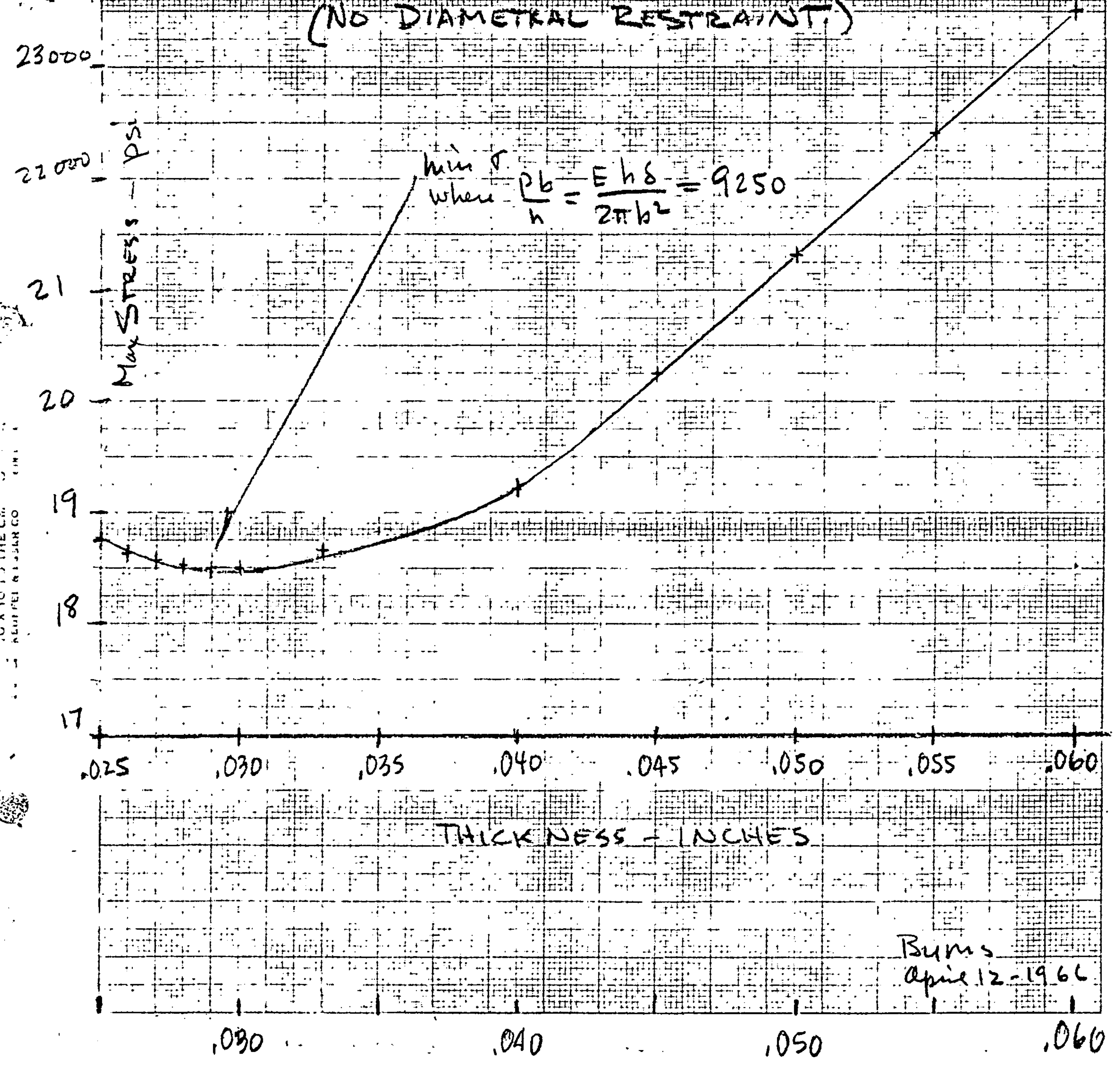
2 DIMEN. STRESS ANALYSIS

From EN 9312-03 M4 page 5

$$S = \frac{pb}{h} + \frac{E h \delta}{2\pi b^2}$$

$b = 1.5$ inches
$p = 180$ psi
$\delta = 0.150$ inches

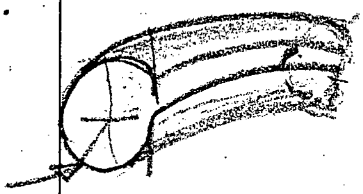
2 DIM. ANALYSIS MOST NEARLY CORRECT FOR RACE TRACK SHAPE (NO DIAMETRAL RESTRAINT)



Byrns
April 12-1966

AUTHOR Byrns | DEPARTMENT Mechanical | LOCATION Berkeley | DATE Apr. 13, 1966

ATTEMPTED Calc. of Bellows Spring Constant and Comparison of .035" VS. .050"

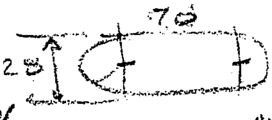


From EN4312-03 M34. J. Myall page 31 — 2 dim.

$$\delta_v = \frac{\pi F b^3}{E I} \qquad \left\{ \text{when } I = \frac{w h^3}{12} \right\} ?$$

b — w = periphery

$$\frac{F}{\delta_v} = \frac{EI}{\pi b^3} = \frac{E w h^3}{12\pi b^3} = \frac{E h^3 w}{12 \pi b^3} \left(\frac{\text{lbs}}{\text{in}^2}\frac{\text{in}^3}{\text{in}^2}\right) = \text{lbs/in}$$



$= 2(70) + \pi 28$
$= 140 + 88$
$w \equiv 228$ inch.

$$\frac{F}{\delta_v} = \frac{(30\times10^6)(3.5\times10^{-2})^3(288)}{12\pi\underset{3.37}{(1.5)^3}} = \frac{(30\times10^6)(288)(42.9\times10^{-6})}{127}$$

$\boxed{2\ \text{DIM.}}$

$$\frac{F}{\delta_v} = (68.1\times10^6)(42.9\times10^{-6}) = \cancel{2920\ \frac{\text{lbs}}{\text{in}}} \text{ for } .035''$$

$$= \quad " \quad (125\times10^{-6}) = \cancel{8510\ \text{lbs/in}} \text{ for } .050''$$

NG

From Argonne Notes. - R.A. Clark & page 8, Myall.
3 dim. with hoop stress.

$\nu = .26$
$\nu^2 = .068$

$$\delta_v = 10.9\underset{\substack{.932^{1/2}\\.966}}{(1-\nu^2)^{1/2}}\frac{Fb}{\pi E h^2} = 10.5\frac{Fb}{\pi E h^2}$$

$$\frac{F}{\delta_v} = \frac{\pi E h^2}{10.5\ b} = \frac{\pi(30\times10^6)}{15.8}\left[3.5\times10^{-2}\right]^2 = (5.96\times10^6)(12.25\times10^{-4}) \quad \boxed{3\ \text{DIM}}$$

$$\frac{F}{\delta_v} = 7300\ \frac{\text{lbs}}{\text{in}} \text{ for } .035''$$

$$\rightarrow = 14900\ \frac{\text{lbs}}{\text{in}} \text{ for } .050''$$

OK, ✓ 4-21-1967 RB

Compare with measured value

Benson of Process Eng'g Co. said on Feb 2, 1966 they calculated $\frac{F}{\delta_v} = 1100\ \frac{\text{lbs}}{\text{in}}$ with equiv diam = 58.6"
$h = .050''$ $b = 1.5''$

N.

For the 25" bellows - being a circle (use 3 DIM.) $b = 1''$, $h = .019$

$$\frac{F}{\delta_v} = \frac{\pi E h^2}{10.5\, b} = \frac{\pi (30\times10^6)(1.9\times10^{-2})^2}{10.5\,(1)} = 8.98\times10^6\,(3.6\times10^{-4})$$

3DIM

$$\boxed{\frac{F}{\delta_v} = 3250 \text{ lbs/in}}$$

Measured stiff on 25" B.C.

~~Watt says about 100psi on oil moves chamber ~ 0.100"~~

~~2000psi in oil at $P_{chamber} \cong 90$ psi~~ $A = \frac{\pi}{4} 27^2 = 571 \text{ in}^2$

~~" " " = 51400 lbs force.~~ NG ?

~~∴ 100psi in " =~~ $\boxed{2570 \text{ lbs}/0.10 \text{ in}} \approx 25{,}700$ lbs/in

From Test log on 25" bellows via Eckman.

$$\boxed{\frac{F}{\delta_v} = 3200 \text{ lbs/in}}$$

2 Dim calc.

$$\frac{F}{\delta_v} = \frac{E w h^3}{12 \pi b^3} = \frac{(30\times10^6)(80)(1.9\times10^{-2})^3}{12\,\pi\,1} = 63.6\times10^6\,(6.86\times10^{-6})$$

$$= \cancel{437 \text{ lbs/in}}$$ No Good.

3" DIA CHAMBER BELLOWS

DESIGN BELLOWS NIPPLES:

① So that they can be seated into a structure stiff enuf to take race track side loads @ 800psi forming pressure. (with ribs) if hydroformed, and easily removed.

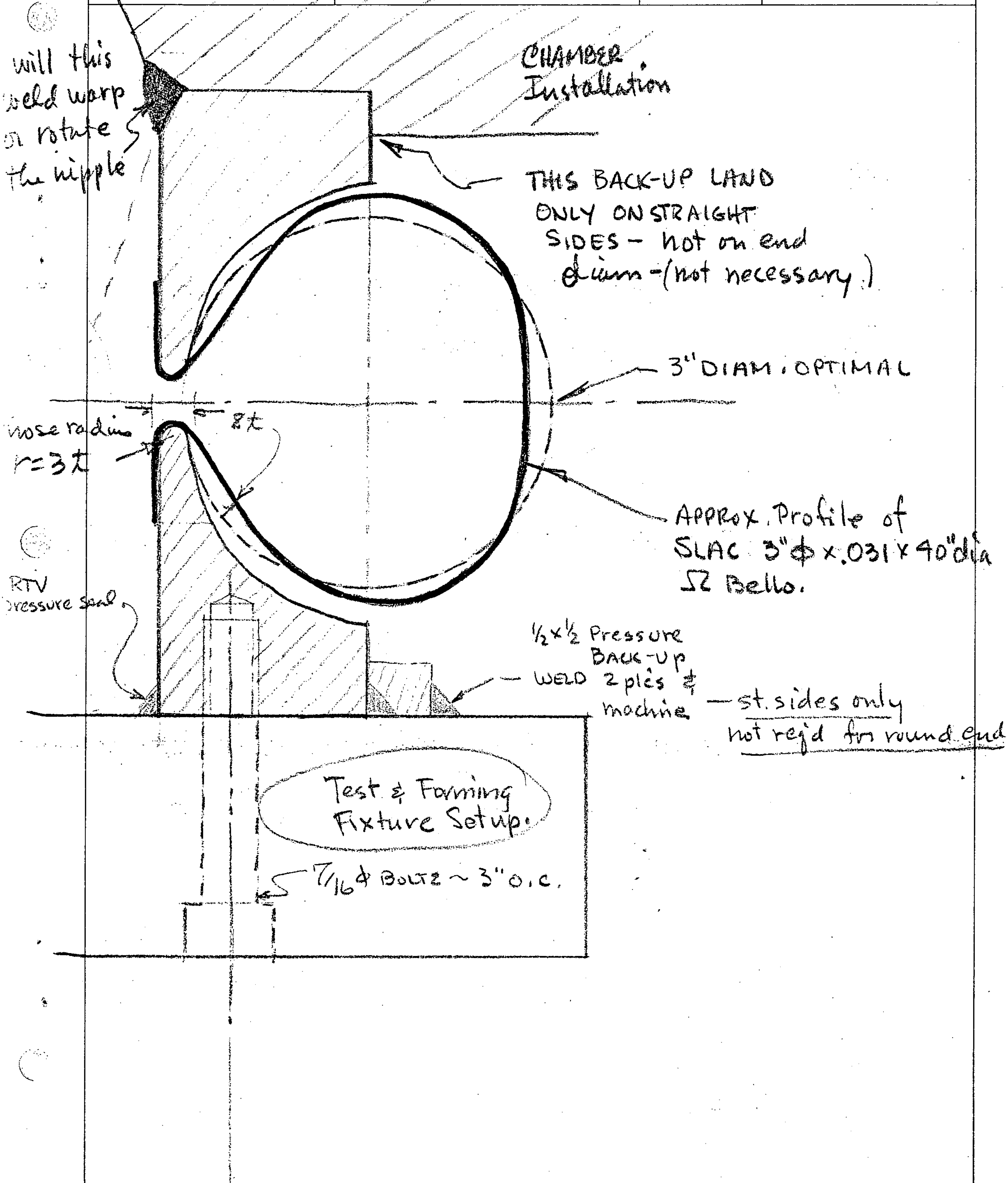
② So that bellows can be installed in test fixture easily and resist pressure load @ 150psi and be removed easily.

③ So that bellows assy with nipples can be installed in chamber castings easily & removed easily.

STRESS ANALYSIS REQ'D.

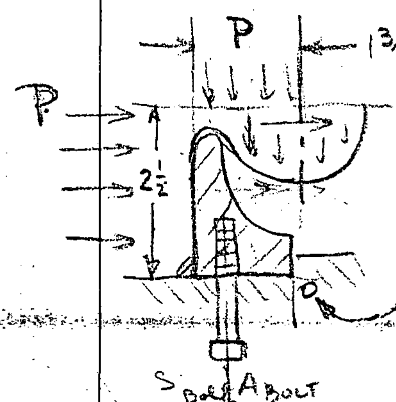
① Stress analysis of all 3 cases above plus stress for forming lips on nipples at pressure ~ 800psi

Check:
1. Stress in nipples, straight section & curve section
2. Vacuum load stability.
3. Turnover moments in nipples
4. Hoop stress in round ends
5. Cantilever beam bending in 1/4" DIA. nose radius at 800psi.
6. Does welding bellow sub assy into chamber casting warp the nipples & bellow?


will this weld warp or rotate the nipple
CHAMBER Installation
THIS BACK-UP LAND ONLY ON STRAIGHT SIDES - not on end diam - (not necessary.)
3" DIAM. OPTIMAL
nose radius r=3t
8t
APPROX. Profile of SLAC 3"Φ x .031 x 40" dia Ω Bello.
RTV pressure seal
1/2 x 1/2 Pressure Back-up WELD 2 plcs & machine
— st. sides only not req'd for round ends
Test & Forming Fixture Setup.
7/16 Φ BOLTS ~ 3" O.C.

AUTHOR R. A. Byrns | DEPARTMENT Mechanical | LOCATION Berkeley | DATE Jan. 25, 1966

TURNOVER MOMENT IN NIPPLE



$\Sigma M_o = 0$ (per 1" lineal.) (+)

$$-S_B A_B \frac{(1.13)}{3} + P(2.5)(1.25) - P(1.75)(.87) = 0$$

3.12   1.52

BOLTS 3" O.C.

$$-S_B A_B \left[\frac{1.13}{3}\right] + P(1.6) = 0$$

$$(800)(1.6) = 1280 = S_B (9.3 \times 10^{-2}) \frac{1.13}{\cancel{3}}$$

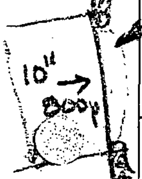
p.909 "Marks" — Area 7/16 Bolt Bottom of thread = .093 in^2

$$1280 = S_B[3.5 \times 10^{-2}]$$

$$S_B = \frac{1280}{3.5 \times 10^{-2}} = \underline{36600 \text{ psi}}$$ (P=800 psi)

not so bad for Allen heads

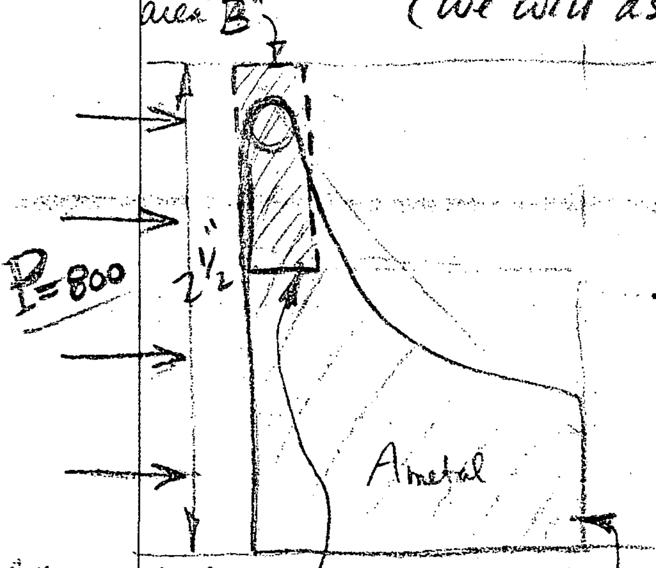
THIS ANALYSIS N.G. when:
$7.5P = S_B A_B$ — BOLT AREA
might need WELD Area

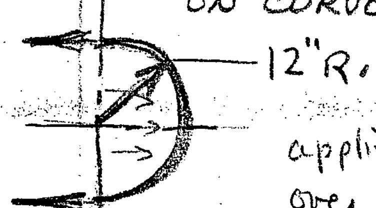


HOOP STRESS IN ROUND ENDS (P=800PSI)

(We will assume no back-up at round ends)

IS NIPPLE SELF SUPPORTING ON CURVED ENDS?





applied pressure = 800 psi over vertical area $2\frac{1}{2}$"

FORCE Balance

$$\Sigma F_{\parallel} = 0$$

$$PA_p = 2 S A_m$$

$$800[2.5](\cancel{12}^{6}) = \cancel{2} S_m A_m.$$

$$(4800)(2.5) = S_m A_m = (1.75)(1.5) S_m = 2.63 S_m$$

$$\underline{S_m = 4600 \text{ psi}}$$

not so bad for Stainless steel

"B" area check.

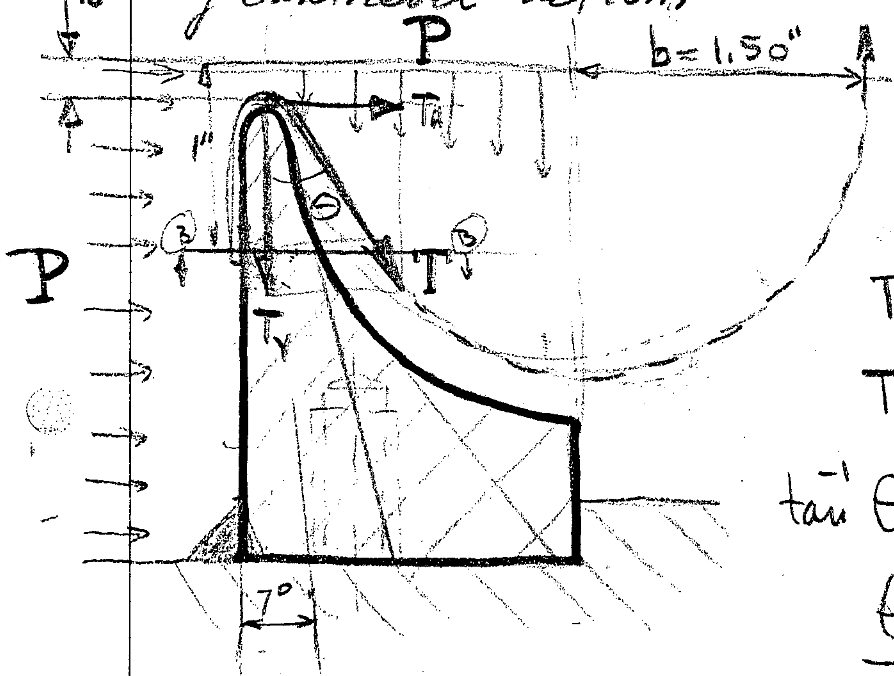
$PA = 2 S_m A_m$
$800(1.00)(12) = 2 S_m(\frac{1}{4} \times 1)$
$S_m = 19{,}200$ psi

∴ Round end is self-supporting — no back-up req'd.

NIPPLE IN STRAIGHT SIDE SECTION REQUIRES BACK-UP. REASON IS OBVIOUS!

CANTILEVER BEAM BENDING ON 1/4 DIA. NOSE RADIUS

Bellows load is taken by hoop stress on round ends, However in straight sides, load must be taken by Cantilever action.



We will consider a unit length of straight Section nipple (1")

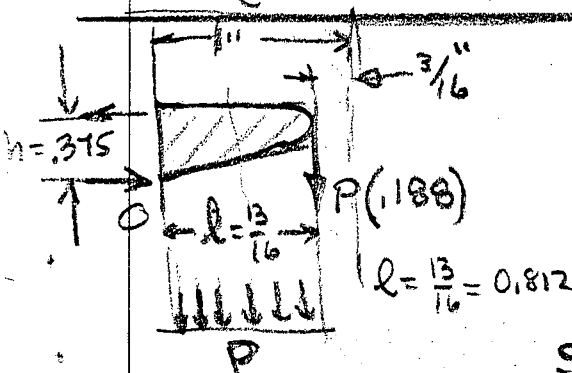
$$T_V = Pb = P(1.5)$$

$$T_H = Ph = P(.188)$$

$$\tan^{-1}\theta = \frac{0.188}{1.500} = .125$$

$$\theta = 7.12^\circ$$

Look @ stress in section BB - 1" below ℄. horiz.



$l = \frac{13}{16} = 0.812$

ΣM_o

$$(0.188)(P)\overset{.812}{l} + \frac{P\overset{.66}{l^2}}{2} = P[.153 + .33] =$$

$$M = 0.483P$$

$$\underset{\text{at B-B}}{S} = \frac{M}{Z} = \frac{0.483P}{\frac{bh^2}{6}} = \frac{\overset{2.90}{6(0.483)P}}{(1)\underset{0.140}{(.375)^2}} = 20P$$

IF P = 800 psi

S_{B-B} = 16,000 psi ← SUPERIMPOSED ON THIS IS COMPRESS STRESS FROM: $T_V = 1.5P$

Bending

Compress. stress from $T_v = 1.5P$

$$S_c = \frac{T_v}{A} = \frac{1.5P}{0.375} = 4P \quad (\text{IF } P=800),\ S_c = 3200 \text{ psi}$$

∴ at sect BB, 1" Below horiz ℄. , when P = 800 psi

$$\Sigma S_{comp} = 16{,}000 + 3200 = 19{,}200 \text{ psi comp.}$$

$$S_{Tens} = 16000 - 3200 = 12{,}800 \text{ psi tens.}$$

By similar maneuvering: use $l = 2"$ $h = 3/4"$ {h is really > 1"}

$$M = 0.188\,Pl + \frac{Pl^2}{2} = 0.376P + 2P = 2.376P$$

$$S_{Bend} = \frac{2.376P\,(6)}{(1)(.750)^2_{.563}} = 25.3P, \quad \{P=800 \quad S \simeq 20{,}000 \text{ psi}$$

$$\Sigma S_{comp} \simeq 21{,}600 \text{ psi}$$

$$S_{comp} = \frac{1.5P}{0.750} \simeq 1600 \text{ psi}$$

$$S_{tens} = 18{,}400 \text{ psi}$$

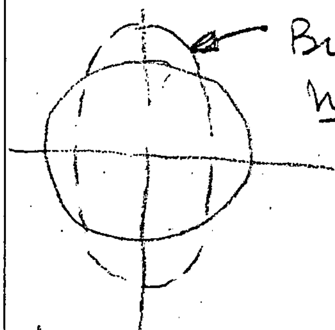
ABOVE NUMBERS FOR STRESS VALUES IN NIPPLE SEEM OK FOR FORMING PRESSURE ≃ 800 psi — FOR OPERATION STRESS VALUES ABOVE would reduce by $\frac{150 \text{ psi}}{800 \text{ psi}}$

{Initial Forming from Sheet Metal TUBE might bend nipple — Let Mfgr. Check This.}

VACUUM STABILITY OF Bellows:

From Myall EN4312-03 M37 page 7.
Timoshenko & Gere "Theory of Elastic Stability" 2nd Ed. 1961 page 289.

Buckling of a circular tube under ext. pressure



Buckles one mode as shown

Non-uniform circ. might affect values.

$$q_{cr} = \frac{E}{4(1-\nu^2)}\left(\frac{h}{b}\right)^3$$

$$\frac{h = .035''}{b = 1.5''} = \frac{3.5\times10^{-2}}{1.5} = \underline{2.34\times10^{-2}} \qquad E = 30\times10^6 \quad \nu = 0.25$$

$$q_{crit} = \frac{30\times10^6}{4(1-.063)}\left[2.34\times10^{-2}\right]^3 = (8)(12.8) = \underline{102\ \text{psia}}$$

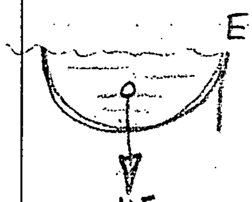
$.94$

3.75

GIVES FACTOR SAFETY ≅ 7 FOR VAC. STABILITY!

One could check if they have evacuated the SLAC 40" DIA × 3" ϕ × .031" successfully —

STRESS DUE WT. OF LIQUID IN BELLOW EN4312-03 M37 page 5.



area of ½ bello is $\frac{\pi}{2}(1.5)^2 = \underline{3.52\ in^2}$

$$\rho_{H_2O} = \frac{62.4}{12^3} = 0.0361\ \#/in^3$$

unit length.

wt. of $H_2O = 3.52(.0361) = \underline{0.128\ \text{lbs.}}$

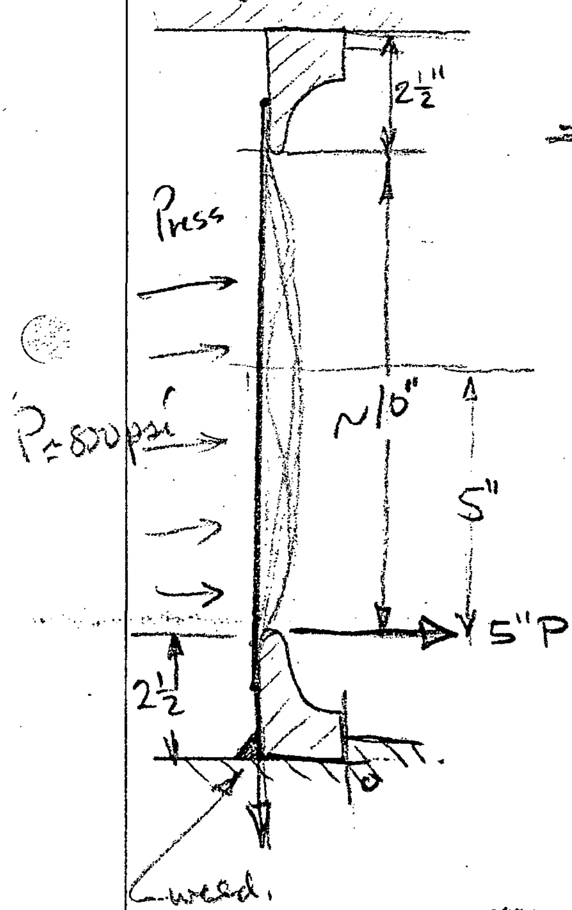
$$M_{oment}\ \text{at}\ E = (1.5'')(.128) = \underline{.192\ \text{lb.in.}}$$

$$\sigma_{max\ \text{at ext. fiber}} = \frac{Mh}{2I} = \frac{6M}{h^2} = \frac{(0.192)(6)}{(3.5\times10^{-2})^2} = .0942\times10^4$$

12.25×10^{-4}

$$= \underline{942\ \text{psi}}$$

All Previous calc's seem to have OK stress values for 800 psi pressure and geometry of bellows as finally formed.
However, it is said Press. started at 800psi and finished about 150psi in model.
For the starting geometry, as shown below, the nozzle lips will probably deform with 800psi.



Calculating for unit length ~ 1" (⊥ to paper)

overturn moment of nipple –

$\Sigma M_0 = 0$

$$(5P)(2\tfrac{1}{2}) = 1.75\, S_w A_{weld}$$

$$7.15\, P = S_w A_w$$

$$5720 = S_w A_w$$

IF THROAT AREA OF WELD = 1/4

$$\boxed{S_w = 23{,}000 \text{ psi}}$$

Cantilever stress from page 4 sec BB

$$M = 5P(1) + \frac{P(1)^2}{2} = P(5.5)$$

$$S_B = \frac{M}{Z} = \frac{5.5(P)(6)}{1\,\underset{.150}{(.380)^2}} = \frac{33.0P}{.150} = 220P$$

$$\boxed{S_B = 176{,}000 \text{ PSI}}$$ ← nozzle lips yield

HOOP STRESS IN ROUND ENDS See p. 3

GROSS HOOP σ

$$PA = 2SA$$

$$800(\underset{15}{7.5})(12) = 2\, S_m \overset{2.6}{A_m}$$ $S_m \cong 15{,}000$ psi

1" NOSE HOOP STRESS

$$\boxed{S_M \simeq 120{,}000 \text{ psi}}$$ ← some nose yield –

Nov. 8 Benson reports - wide flanges on die crumpled some and at 280 psi max internal pressure round ends still flat ⌒ and strait sides start to "peak up" - go triangular → ⌒ approaching rupture strength of material @ corners! ∴ Internal pressure 280 psi is not enuff to push out round ends into circle → Ω and too much for straight sides!

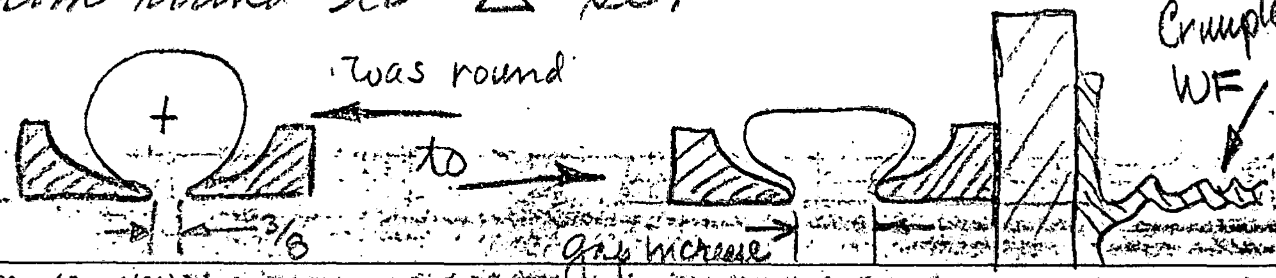
Our (R. Watt & me) conclusions are: Things in nature don't go triangular - like sheet metal under pressure would go round ⊕ like soap bubbles, etc. The calculated stress in the straight sides under internal pressure is! — 3" dia .037" wall $y.s \geq 30{,}000$ psi

$$S = \frac{PD}{2t} \rightarrow \frac{2tS}{D} = P$$

$$P = \frac{2(.037'')(30{,}000)}{3''} = \boxed{740\text{ psi}}$$ max. internal pressure before reaching yield of 30,000 psi

more likely → 800 psi.+ with yield of 40,000 psi

What might have happened is: If the wide flange crumpled it allowed the nipple noses to move apart and so distort the st. side profile from round to △ so! —



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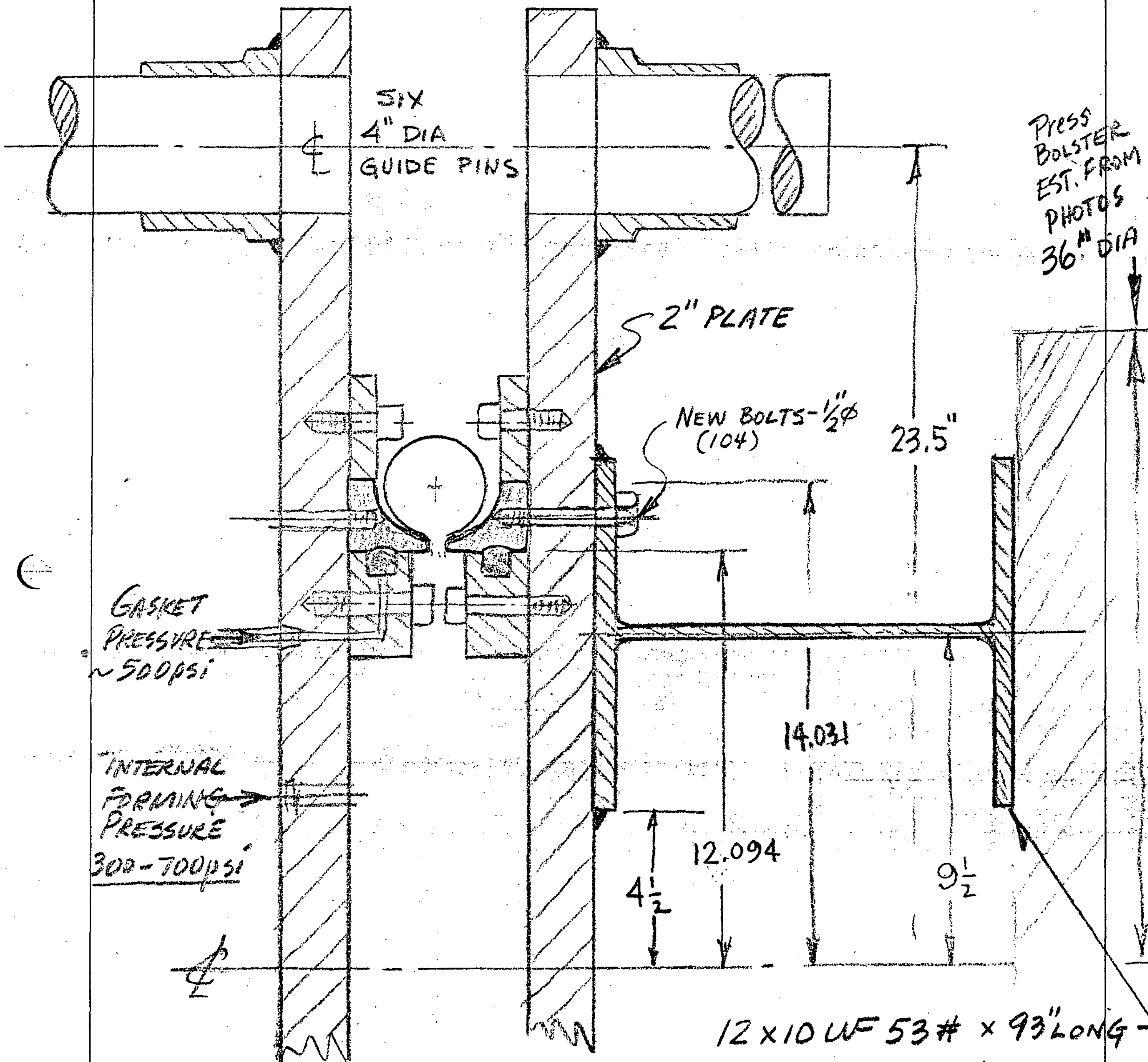
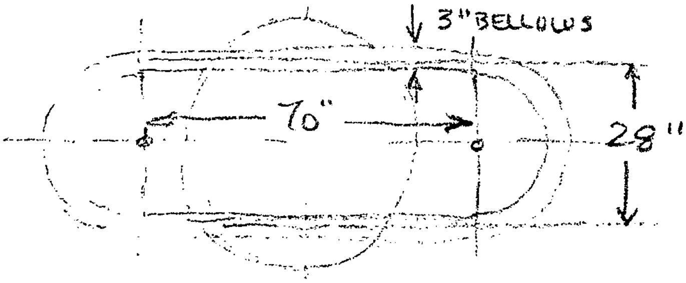


PRESS RAM Area - DIA = 17"
(EST. FROM PHOTOS) $A = \frac{\pi}{4} D^2 = \underline{227\ in^2}$ — 5000psi ram pressure
$\simeq$ 600 TON PRESS

AUTHOR R Byms | DEPARTMENT ME | LOCATION B | DATE 11-8-66

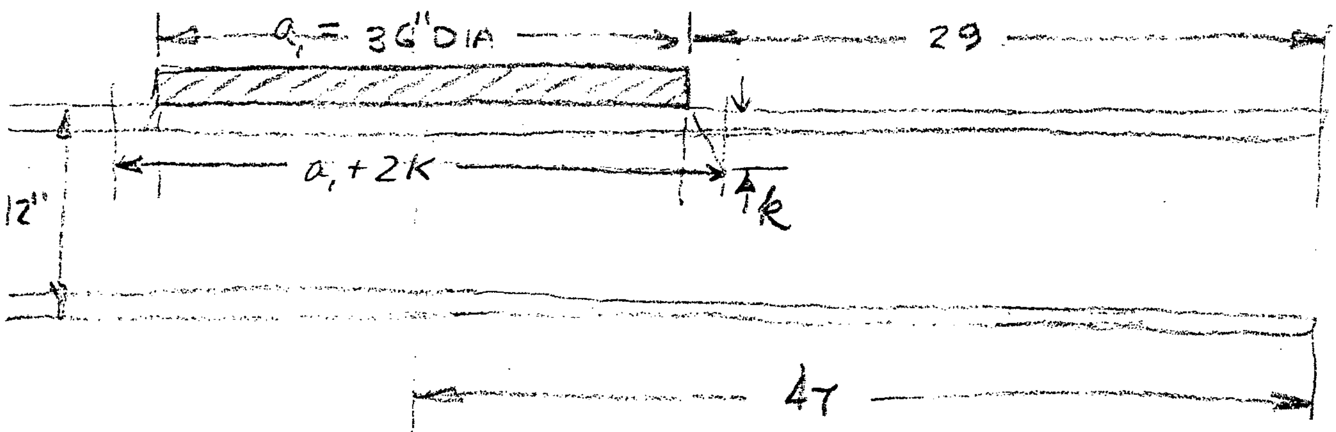
FORCE ANALYSIS OF LOADS ON PRESS & DIE



$$AREA = 70(28) + \frac{\pi}{4} 28^2$$
$$A = 1960 + 615$$
$$\underline{A = 2575 \text{ in}^2}$$

FORCE = PA = 2575 P {where P is FORMING PRESSURE.}

PRESS BOLSTER CONTACT ON 12×10 WF is ≃ 36" DIA.
(EST. FROM PHOTOS.)



internal pressure at die failure was 280 psi (Benson)
total press load was 280(2575) = 720,000 LBS. (360 TON

FROM AISC Handbook (1947) p.172
CRIPPLING VALUE OF BEAM WEBS.
CONCENTRATED LOAD SHOULD NOT EXCEED 24 KIPS/IN.²
MAX. INTERIOR LOAD. $= 24t(a_1 + 2K)$.

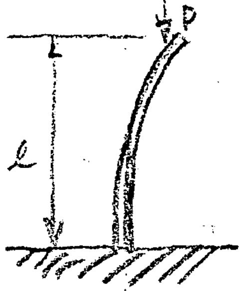
t = web = ~~3/8"~~ .345
a_1 = 36"
k = 1 3/16"

max INT. LOAD $= 24(\cancel{3/8})^{.345}(36 + 2\tfrac{3}{8}) = 9(38.375) =$ ~~345~~ 318 KIPS.

FOR 2 - 12×10 WF max. load is 636 KIPS.
TOTAL LOAD WAS 720 KIPS.
∴ WE EXPECT THE WEBS TO BUCKLE!

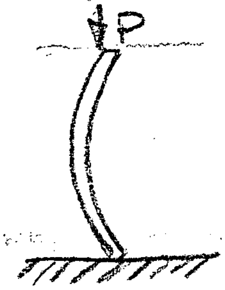
Addenda Notes re: column stability

Column Euler loads - from Roark p.340 Case I & II



$$P_{cr} = \frac{\pi^2 EI}{4l^2}$$

one end free
other end fixed



$$P_{cr} = \frac{\pi^2 EI}{l^2}$$

Both ends hinged

$$I = \frac{bh^3}{12} = \frac{(38.4)(3.75)^3(10^{-1})^3}{12} = \underline{0.168 \text{ in}^4}$$

52.6

{ error
h = web is 0.345"
instead of .375" }

$E = 29 \times 10^6$ psi

$$l = l - 2k \text{ (for 12WF)} = 12 - 2\tfrac{3}{8} = \underline{9.625''}$$

$$A = bh = (38.4)\frac{3}{8} = \underline{\cancel{14.4} \text{ in}^2} \quad 13.3$$

$$k = \sqrt{\frac{I}{A}} = \sqrt{\frac{16.8 \times 10^{-2}}{14.4}} = (1.16 \times 10^{-2})^{1/2} = 0.108$$

Case I

$$P_{crit} = \frac{\pi^2 (29 \times 10^6)(0.168)}{4\,(9.625)^2} = \frac{(29)(1.655)10^6}{371} = 1.295 \times 10^5$$

$$\underline{{}_{I}P_{crit} = 129.5 \text{ KIPS}}$$

$$\underline{S_{crit} = \frac{P}{A} = 9000 \text{ psi}}$$

Case II

$$P_{crit} = 4\left(P_{crit_I}\right) \quad - \quad \underline{{}_{II}P_{crit} = 530 \text{ KIPS}}$$

$$\underline{S_{crit} = 36{,}000 \text{ psi}}$$

$$\frac{l}{k} = \frac{9.625}{0.108} = \underline{89.2.}$$

(Note this stress much lower than Beam web crippling value on p.10 main members)

allowable stress (AISC . p.209) $l/k = 90$, $\boxed{S_a = 13{,}100 \text{ psi}}$

~~stress was 360 KIPS~~ $\frac{}{14.4 \; 13.3}$ ~~$\boxed{25{,}000 \text{ psi}}$~~ 27,200 psi

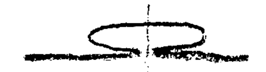
PER AISC. HANDBOOK. p.172

"When the above values are exceeded the webs of the beams should be reinforced, or bearing length increased. Lack of proper support for the top flanges of beams at the reaction points so decreases the crippling strength of the webs as to render such practice inadmissable"

Thus; as predicted by AISC. - The webs collapsed. @ 280 psi

The now obvious and "quick & dirty" answer is to weld stiffeners into the WF webs.

But lets look a little farther, and take the engr approach - plugging some numbers in.

The 1st question is: What is the necessary internal forming pressure to push the round ends - (now flat ) into a 3" circle.

On p.8 we have shown max. pressure to yield strength is 750-800 psi internal. In Jan. Benson said they took the model to 800 psi to start a bow and finished at 150 psi (in .037" ss.)

On Argonne report "ANL-BBC-73 Lyle GENENS 10-29-65" of a visit to Badger to watch the SLAC bellows being formed sez: The major portion of the forming had taken place at a pressure below 200 psi with needle fluctuating between 275 and 400 psi at the time the press was stopped. (5/8" from the stop.)
The press was then closed to the fixed stop and the gage fluctuated between 400 and 575 psi

SLAC Bellows is a nominal 40" dia. x 3" ϕ omega bellows and thickness is .031"

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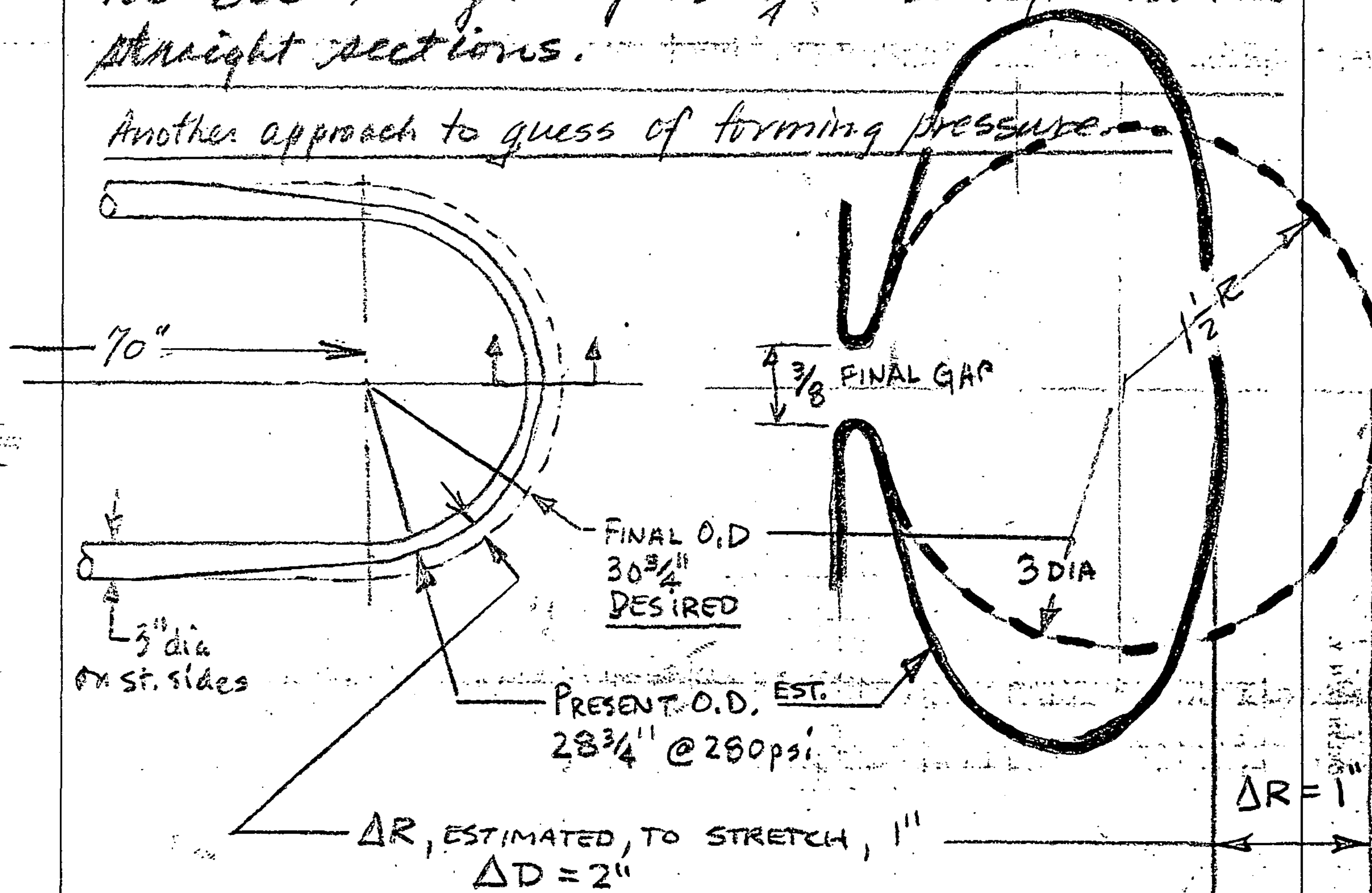
Our bellows is .037" and these numbers would indicate that (.037" > .031") forming pressure in round ends with 2 dimensional stretch might be in XS of 500 psi – possibly even 700-800 to get up to yield strength in the straight sections.

Another approach to guess of forming pressure.



STRETCHING occurs in plastic region
∴ $S = E\epsilon$ not proportional or valid, Poisson ratio not constant.

1st approx. Hoop stress. $S = \frac{PD}{2t}$

$$P = \frac{2ts}{D} = \frac{2(.037")s}{29} = 2.55 \times 10^{-3} s$$

RL-3220-a (Rev. 4/64)

$P = 2.55 \times 10^{-3} S$

REQ'D ELONGATION OF OUTER FIBER IS –

$\epsilon = \frac{\Delta R}{R} = \frac{1}{15} \simeq 6.7\%$ (THIS IS not very much elongation to make!)

To get this elong. – est. stress → $10^5 - 1.2 \times 10^5$ psi

IF $S = 10^5$ $P = 255$ psi

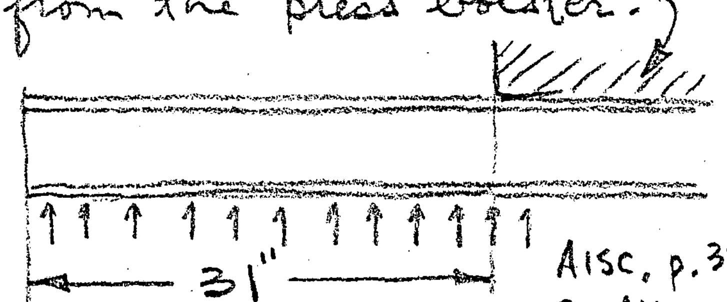
IF $S = 1.2 \times 10^5$ $P = 300$ psi

Benson sed he had 280 psi and ends are still flat. ? (∴ This theory not so good)

Maybe the 12×10WF in the end area bent, (CANTILEVER) let the 3/8" gap open up and flattened the sheet metal profile?

Refer to die plan view – next page.

The 12×10WF is cantelevered out to each end, from the press bolster.



LOAD Area is 14" wide

Pressure was 280 psi

$w = 280(14) = 3920$ lbs/in.

AISC, p. 372 #case 19
Cantilever – uniform load.

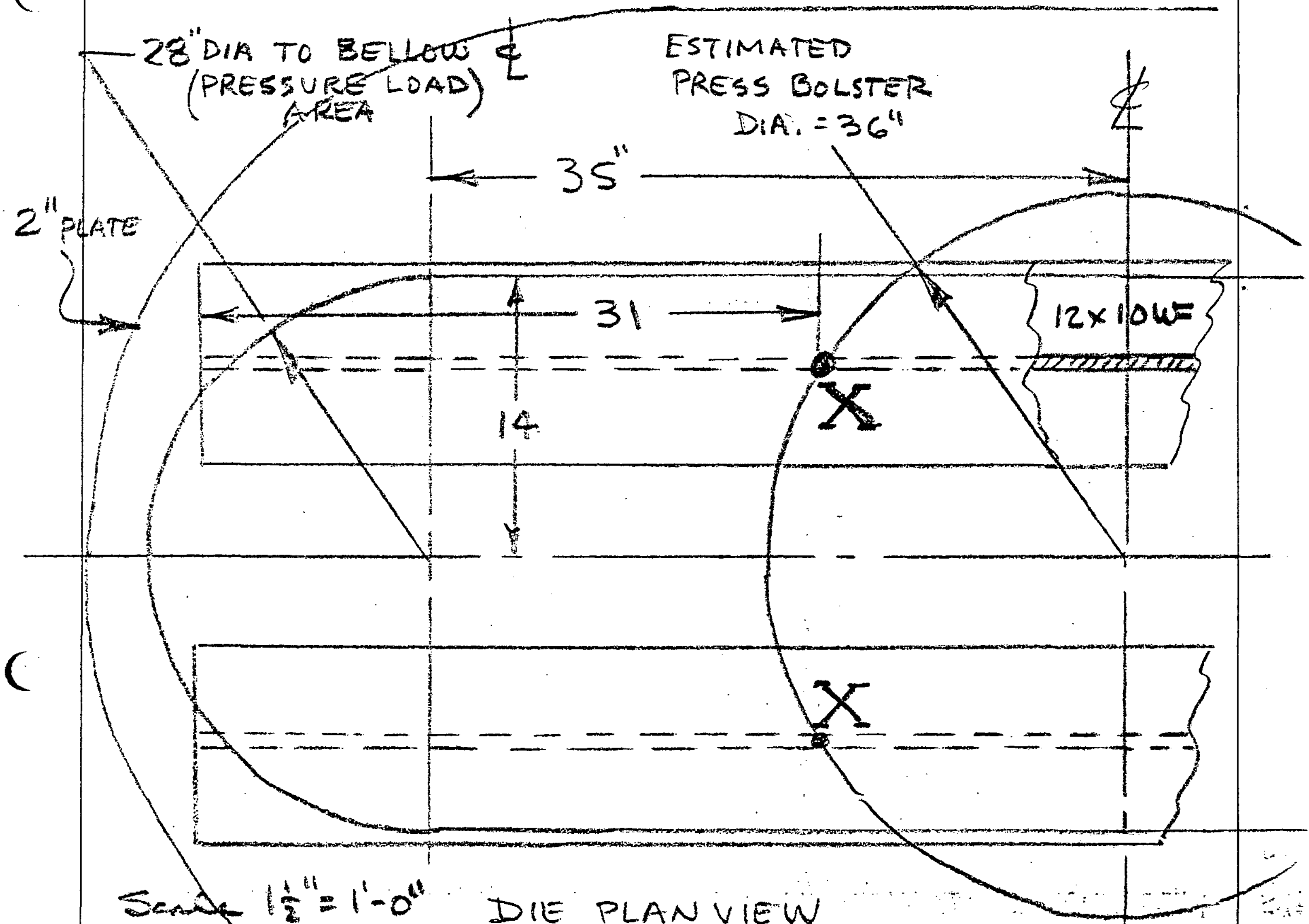
$M_{max} = \frac{w\ell^2}{2}$
fixed end

$\Delta_{max} = \frac{w\ell^4}{8EI}$

For 12×10WF×53#

$I = 426.2$ in.^{4}

$Z = 70.7$



Scale $1\frac{1}{2}" = 1'-0"$ DIE PLAN VIEW

$$s = \frac{M}{Z} = \frac{wl^2}{2Z} = \frac{(3920)(31)(31)}{2(70.7)} = \underline{26,600\text{ psi.}}$$

(seems OK.)

(The 2" plate behind the WF lowers stress greatly.)
But this throws high stress (compressive) into web of WF at press bolster - contributing to buckling.)
We had 27,200 psi direct compression on the web, now <u>superimposed</u> is 26,600 psi compression due bending at 90°, biaxial compression at pt. X.

(Can reduce bending stress by 30% with 12×8WF×45# in center

Deflection:

$$\Delta = \frac{wl^4}{8EI} = \frac{3920(961)^2}{8(29\times10^6)426} = \frac{(1.15)(92.5\times10^4)}{29\times10^6}$$

$$\underline{\Delta = 3.67\times10^{-2} = .037''}$$ (not so much)

Pressure	Stress	Deflection
280 psi	26,600	.037"
300 psi	28,500	.040"
400	38,000	

too close to yield – Do not exceed 300psi

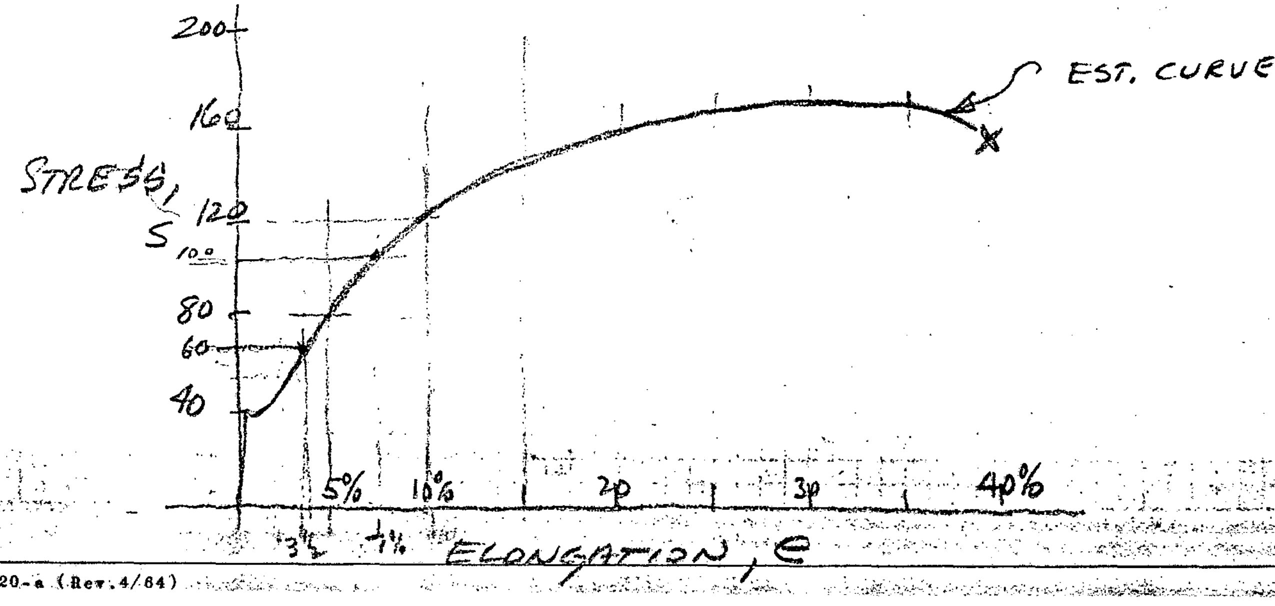
I still don't know what the necessary forming pressure is. But will think about a strain energy balance.

NOV 14.

(FIRST) APPROXIMATION OF FORMING PRESSURE REQ'D FROM STRAIN ENERY BALANCE

MAKE A GUESS OF STRESS-STRAIN CURVE FOR S.S. #316 – FULL ANNEALED.

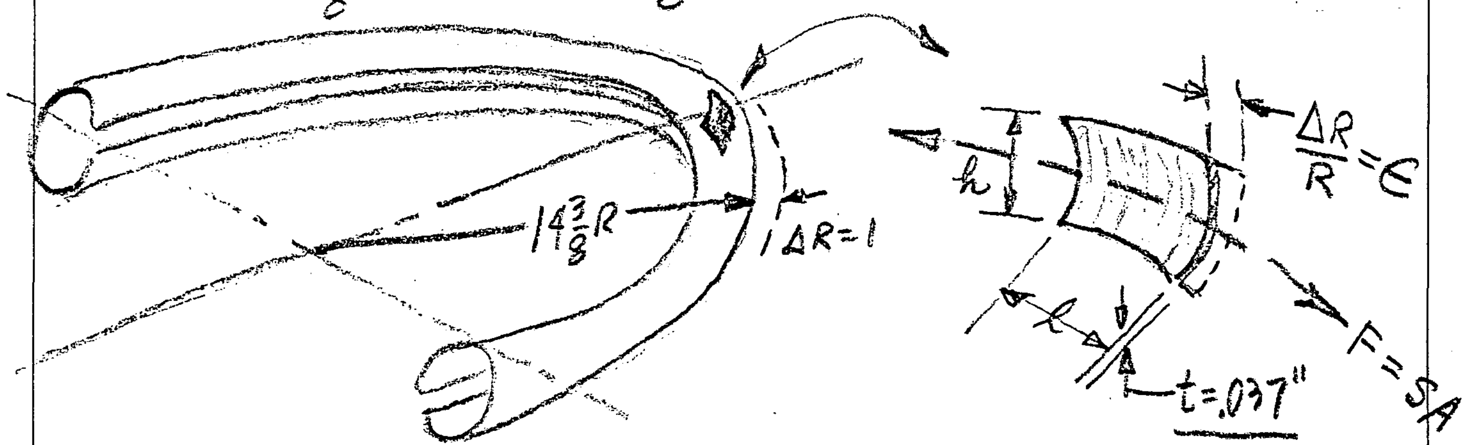
(#304 can be work-hardened up to an ultimate ≃ 200K.)



REFER TO SKETCH ON PAGE 13.

THE ROUND ENDS OF RACETRACK BELLOWS ARE FLAT, BY 1". THE PRIMARY RESTRAINT TO ALLOWING THEM TO GO ROUND IS THE HOOP OR FILAMENT STRESS ON the $14\frac{3}{8}$ RADIUS.

THE RADIUS (PROP. TO CIRCUM.) HAS TO STRETCH FROM $14\frac{3}{8}R$ TO $15\frac{3}{8}R$ OR $\Delta R = 1"$



TAKE A little element out of neutral axis on HORIZ. ℄ AS SHOWN.

THIS element must stretch or elongate $\frac{\Delta R}{R} = \frac{1}{14.38} = 7\%$

The work to stretch it is equal to FORCE x dist integrated over the stress-strain curve.

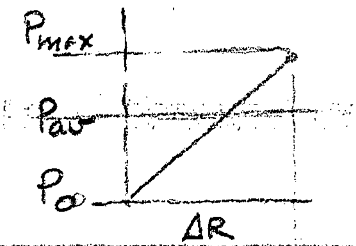
$$W_i = \int F d = \int SA\epsilon = \int S t h e$$

$$W_i = th\left[50{,}000(.035) + 80{,}000(.035)\right] = 1750 + 2800$$

$W_i = 4550\ th.$ → REQ'D INTERNAL WORK IN METAL (in-lbs)

The req'd external work is the force × distance
This is = Pressure · Area (ΔR)
let length of element = $l = 1''$ (then total $\epsilon = 7\% = .070''$)

$W_{external} = \int P\,h\,l\,\Delta R$ $\quad l = 1''$, $\Delta R = 1''$



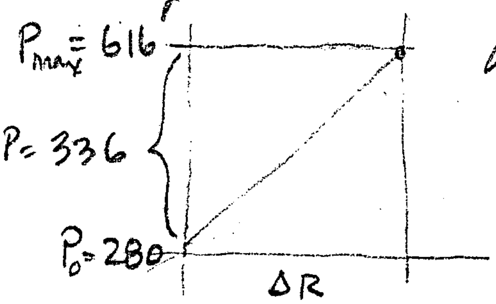
$W_E = \frac{1}{2} P h$

Equate for energy balance $W_E = W_i$

$W_i = 4550\,th = W_E = \frac{1}{2} P h$

$P = (2)\,4550\,(.037) = \boxed{336\,psi}$

THUS: 336 psi is minimum forming pressure.
depending on how well we have guessed the stress-strain curve.
We know that at 280 psi (per Benson) round ends are still flat, 1" per measurement.
and if we redraw the P-ΔR curve thus:-



we get $P_{max} = 280 + 336 = \boxed{616\,psi}$

Finishing pressure.
which more nearly approaches the SLAC BELLOWS values.
as noted on page 12.

THUS: Final forming pressures are above ~350 psi and may approach 600 psi (?)

This is still below the yield stress value in the st. section if remains perfectly round (3" dia.) – ($p = 740$ psi.) see page 8.

The die has reached maximum at 280 psi the 12×10WF beams buckle in the web.

Recommend: weld stiffeners into 12×10WF webs at press bolster reaction points add 12×8 WF×50# down the center to take cantilever loads & STIFFEN WEB.

Stress all die for 600 psi internal pressure.

Bellows exp. joint area is 2575 in.^{2}

at press = 600 psi, total load is $\frac{2575(\cancel{600}^{3})}{\cancel{2000}} =$

770 TONS. (I dunno if the press will do this? I think it's a 600 ton.)

(IT might be necessary to take the SIX 4" dia. leader pins and put threads & nuts on them to take the force.)

ADDENDA: WALL THINNING IN END ZONE

POISSON RATIO $= \nu = \frac{dt/t_0}{\cancel{dl}/\ell_0} = 0.25$ $\quad dt = t_0(0.25)(.07) = .037(.0175) \simeq .00065$

less than .001" – negligable

AUTHOR SHUCK YEE | DEPARTMENT MECH. ENGINEERING | LOCATION 90 | DATE 2-8-67

PROGRAM — PROJECT — JOB
82" BUBBLE CHAMBER

TITLE
EXPANSION BELLOW: REF. EQUATIONS, NAA-SR-4527

$$S_{mp} = \frac{Ph}{N_p t} = \frac{150 \times 1.5}{1 \times t} = \frac{225}{t}$$

$$S_{bd} = \frac{.412 E_c t \Delta \bar{R}}{h^2 N_c C_d R}$$

$$= \frac{.412 \times 28 \times 10^6 \times t \Delta \times 13.875}{1.5^2 \times 1.4 \times 12.125}$$

$$= 4.19 \times 10^6 \times t \Delta$$

$$S_d = 1.5 S_{mp} + .5 S_{bd}$$

$$K_a = \frac{.431 \times \bar{R} E_h t^3 N_p}{N_c h^3 C_f}$$

$$= \frac{.431 \times 13.875 \times 28 \times 10^6 t^3 \times 1}{1 \times 1.5^3 \times .41} = 1.212 \times 10^8 t^3$$

$P = 150$ PSI
$h = 1.5''$
$N_p = 1$
$\bar{R} = 13.875''$
$N_c = 1$
$C_d = 1.4$
$R = 12.125$
$C_f = .41$
$E_h = 28 \times 10^6$
$t = .049''$

t	.049	.049	.049	.100	.100	.100
Δ	.100	.150	.200	.100	.150	.200
S_{mp}	4600	4600	4600	2250	2250	2250
S_{bd}	20,500	30,700	41,000	21,700	32,550	43400
S_d	17,150	22,250	27,400	14,225	20,650	25,075
K_a	14300#/IN	14,300#/IN	14,300#/IN	49,600	49,600	49,600

AUTHOR	DEPARTMENT	LOCATION	DATE
SHUCK YEE	ME. ENGR.	90	2-8-67

$S_{mp} = \frac{225}{.031} = 7260 \text{ PSI}$

$S_{bd} = \frac{.412 \times 28 \times 10^6 \times .031 \Delta \times 13.875}{1.5^2 \times 1 \times 12.125}$

$= 1.82 \times 10^5 \Delta$

$S_d = 1.5 S_{mp} + .5 S_{bd}$

$t = .031$

$X = .778\sqrt{13.875 \times .031}$

$= .51$

$\frac{a}{X} = \frac{1.5}{.51} = 2.94$

$C_d = 1$

$C_f = .29$

$K_a = \frac{.431 \times 13.875 \times 28 \times 10^6 \times \overline{.031}^3 \times 1}{1 \times \overline{1.5}^3 \times .29}$

$= 5100 \text{ \#/IN.}$

t	.031	.031	031
Δ	.100	.150	.200
S_{mp}	7260	7260	7260
S_{bd}	18200	27,300	36400
S_d	19,990	24,540	29,090
K_a	5100 #/IN	5100 #/IN	5100 #/IN

$S_{mp} = \frac{225}{.037} = 6080$ PSI

$S_{bd} = \frac{.412 \times 28 \times 10^6 \times .037 \Delta \times 13.875}{1.5^2 \times 1.2 \times 12.125}$

$= 1.81 \times 10^5 \Delta$

$S_d = 1.5 S_{mp} + .5 S_{bd}$

$K_a = \frac{.431 \times 13.875 \times 28 \times 10^6 \times .037^3 \times 1}{1 \times 1.5^3 \times .37}$

$= 6780$ #/IN.

$t = .037$

$X = .778\sqrt{13.875 \times .037}$

$= .567$

$\frac{a}{X} = \frac{1.5}{.567} = 2.64$

$C_d = 1.2$

$C_f = .37$

t	.037	.037	.037	.120	.120	.120
Δ	.100	.150	.200	.100	.150	.200
S_{mp}	6080	6080	6080	1875	1875	1875
S_{bd}	18100	27,150	36200	25,100	37850	50,200
S_d	18,170	22,695	27,220	15,363	21,738	27,913
K_a	6780 #/IN	6780 #/IN.	6780 #/IN.	71,500	71,500	71,500

$S_{mp} = \frac{225}{.065} = 3460$ P.S.I.

$t = .065$

$X = .778\sqrt{13.875 \times .065}$

$= .74$

$S_{bd} = \frac{.412 \times 28 \times 10^6 \times .065 \Delta \times 13.875}{1.5^2 \times 1.8 \times 12.125}$

$= 2.12 \times 10^5 \Delta$

$\frac{a}{X} = \frac{1.5}{.74} = 2.03$

$C_d = 1.8$

$C_f = .7$

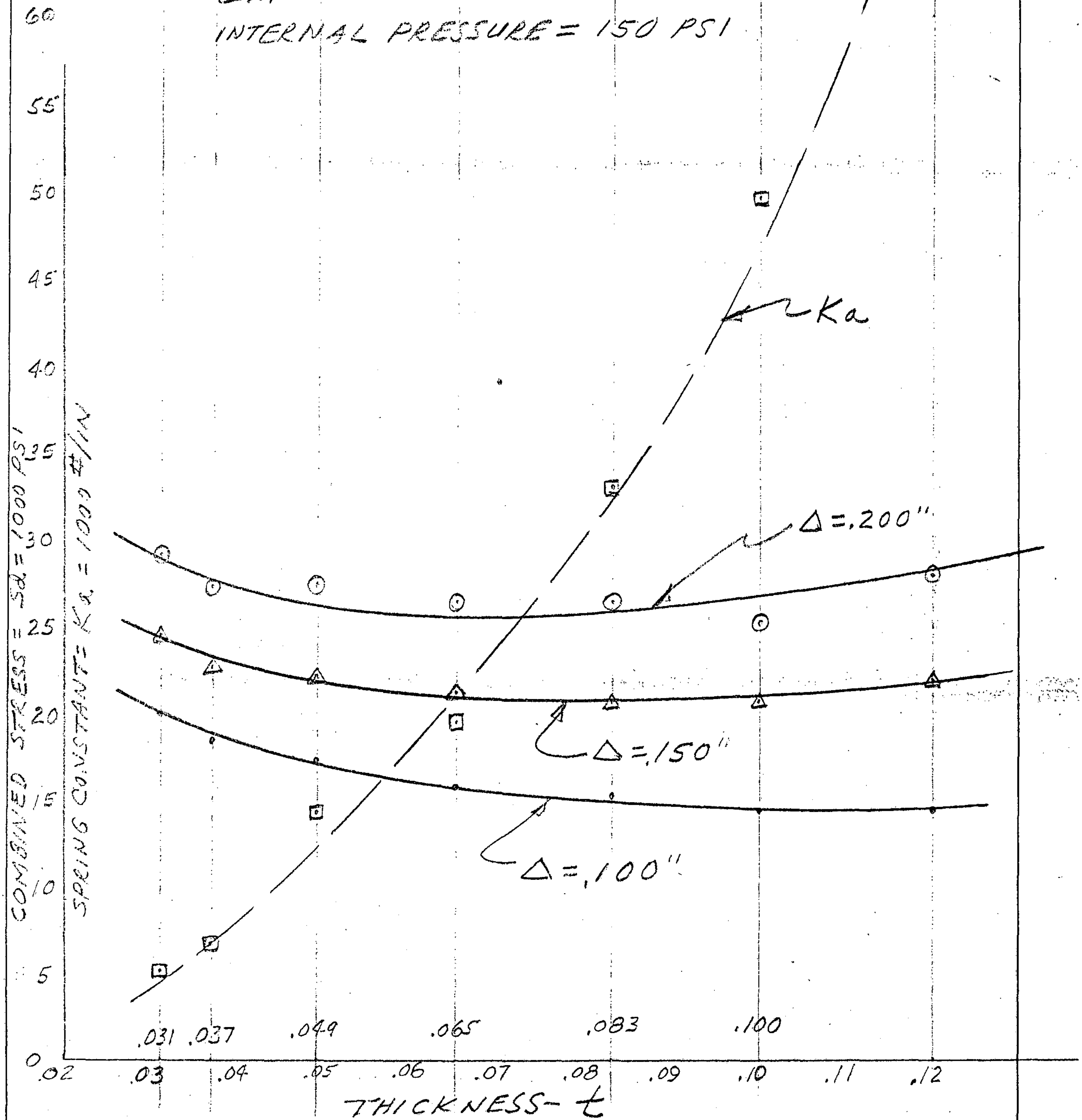
$S_d = 1.5\, S_{mp} + .5\, S_{bd}$.

$K_a = \frac{.431 \times 13.875 \times 28 \times 10^6 \times \overline{.065}^3 \times 1}{1 \times \overline{1.5}^3 \times .7}$

$= 19{,}450$ #/IN.

t	.065	.065	.065	.083	.083	.083
Δ	.100	.150	.200	.100	.150	.200
S_{mp}	3460	3460	3460	2710	2710	2710
S_{bd}	21,200	31,800	42,400	22100	33150	44200
S_d	15,790	21,090	26,390	15115	20640	26,165
K_a	19,450	19,450	19,450	33,000 #/IN	33000	33,000

82" BUBBLE CHAMBER
EXPANSION BELLOW
INTERNAL PRESSURE = 150 PSI



AUTHOR: R Byrne | DEPARTMENT: Mechanical | LOCATION: BErkeley | DATE: 6/ –

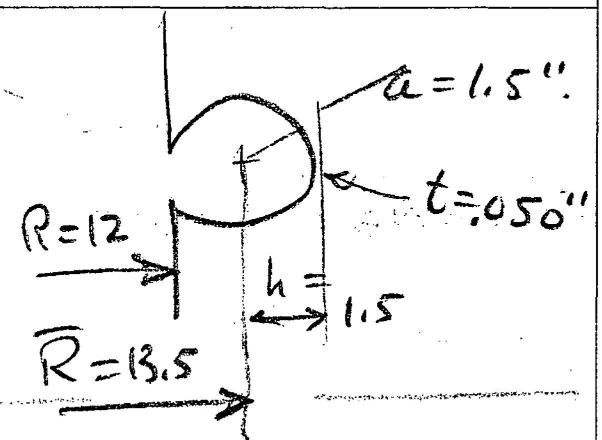
PROGRAM – PROJECT – JOB

NAA - SR - 4527 Vol I p. 28.

TITLE

press. = 150 psi

Δ = 0.180" [± .090"]



~~Allowables~~.

Membrane Stress

$$S_{mp} = \frac{150 (1.5)}{.050} = \underline{4500 psi}$$

t = .050

$$S_{bd} = \frac{0.412 E_c t \Delta\bar{R}}{h^2 N_c C_d R} = \frac{.412 (30\times10^6)(.050")(.090)(13.5)}{1.5^2 \cdot 1 (1.5) 12.}$$

S_{bd} = $\underline{18600 psi}$

Combined

$S_d = 1.5 S_{mp} + 0.5 s_{bd} = (1.5)(4500) + 0.5 (18600)$

6750 + 9325 = $\underline{16,075 psi}$

$\underline{t = .050"}$

t = .030" $1\frac{1}{2}(7500) + 0.5(16,700) = S = \underline{19,600 psi}$

Spring Konst.

t = .050"

$$K_s = \frac{.0431 \bar{R} E t^3 N_e}{N_c h^3 C_f} \#/in = 15,400 \#/in.$$

Bellows Test. (Prelim.)

I. A single convolution bellows as shown in dwg 12C5543A - overall dimensions $30\frac{3}{4}" \times 100\frac{3}{4}" \times 5"$ High. shall be cyclic tested.

Internal pressure shall be 180 psi at full closure.

Stroke shall be $\pm 0.150"$ from equilibrium bellows position, for total of 0.300".

Internal pressure drops on expansion.

Motion shall be parallel within 0.010".

Cyclic test shall extend a minimum of 1,000,000 cycles.

Cycle frequency shall be 1 second or less. (approx. 12 days) $(3600)(24) = 86{,}400/\text{DAY}$

FORCE & MASS ESTIMATES.

PROJECTED Bellows area is: $(27.750)\left[70 + \frac{\pi}{4}(27.750)\right] = 2550 \text{ in}^2$

PRESSURE FORCE . . . . $2550\,(180) = \boxed{460{,}000 \text{ lbs.}}$

BELLOW SPRING CONSTANT - ($t = 0.049"$)

IF BELLOWS WERE CIRCULAR:

FORMULA GIVEN IN NAA-SR-4527 VOL. I page 28

AXIAL SPRING RATE: $$K_s = \frac{0.431\,\bar{R}\,E\,t^3}{h^3\,C_f}.$$

$$K_s = \frac{0.431\,(13.87)(28\times10^6)\,\overset{117.5}{(4.9\times10^{-2})^3}}{\underset{3.37}{(1.50)^3}\,(0.42)}$$

$$K_{s_{CIRC.}} = 14.24\,(8.3)(117.5) = \boxed{13{,}900 \text{ lbs/in.}}$$ FOR CIRC. SECTION

By W. Rechhams

MEASURED (6 Feb 67) K_s FOR 7" st. section was $7.65\ \#/\tfrac{1}{10}$ inch

OR $76.5\ \#$/inch for 7" linear $\sim 11\ \tfrac{\#}{in/in}$

we have $2\times70" = 140$ lin. inch, so:

$$K_{s_{ST.}} = 20\,(76.5\ \#/in) = \boxed{1530 \text{ lbs/in}}$$ FOR ST. SECTION

$$K_{s_{TOTAL}} = K_{ST.} + K_{CIRC.} = 1530 + 13{,}900 = \boxed{15{,}430\ \#/in}$$

REQD FORCE FOR: $\pm 0.150"$

$$F_K = 0.150"\,(15{,}430\ \#/in) = \boxed{\pm 2315\ \#}$$ 1/2 STROKE

or $\boxed{4630\ \#}$ FULL STROKE

(Small compared to pressure loads.) $\simeq (1\%)$

EST MASS. – MOVING PARTS. –

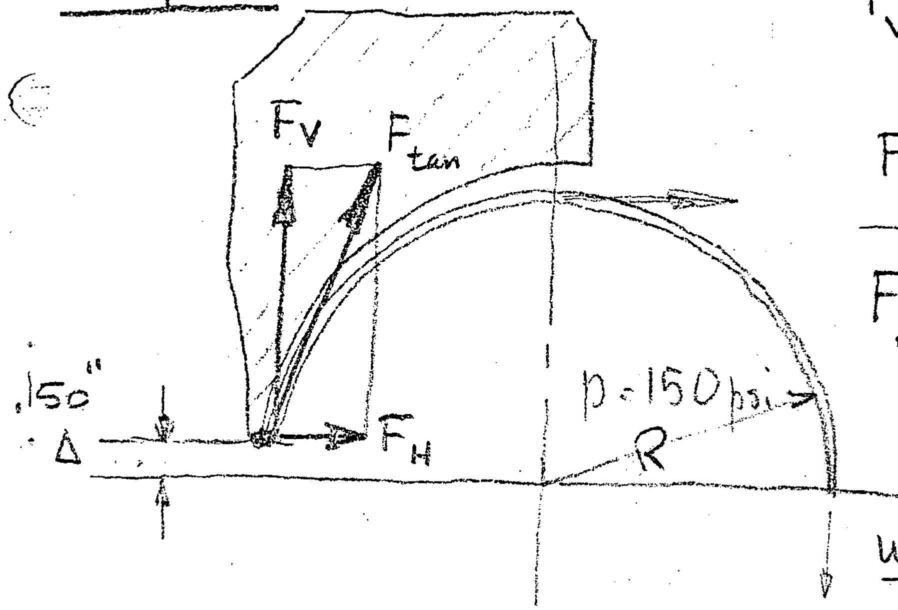
EST. 2" STEEL PLATE OR (3" ALUM. PLATE)

FOR PRESSURE LOADS: @ 200 psi

approx area is $2\tfrac{1}{2}$ FT × 8 FT = 20 ft²

wt. steel (3") is 122 #/ft² · (20 ft) = $\boxed{2440 \text{ lbs.}}$

wt. alum. (3") is 43 #/ft² . . . . . . (860 #)

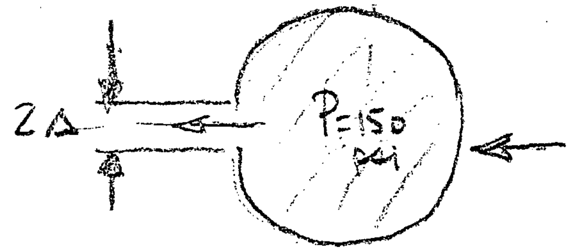


F_V is sum of pressure loads plus deflection loads ($\pm$)

$F_V = F_p \pm F_\Delta$

F_H can be calc. from tangent $\angle$ of bellow or result from unbalanced pressure force

F_H is $= p\Delta$



see pix

$F_p = pR = 150(1.5) = 225$ #/lin in.

$F_\Delta =$ from spring constant & measurements $= 20$ #/lin. in.

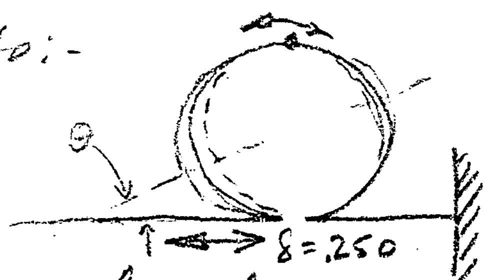
$$\boxed{F_V = F_p \pm F_\Delta = \underline{245 \text{ or } 205 \text{ \#/lin in}}}$$

$$\boxed{F_H = p\Delta = 150(.150) = 22.5 \text{ \#/lin in.}}$$

A section (strait) of toroidal bellow approx. 7" long with various welds was set up in a stroking rig. 3" O.D tube x .049 w. S.S. type 321 Was run at 900 RPM for approx. 1.7×10^6 cycles.

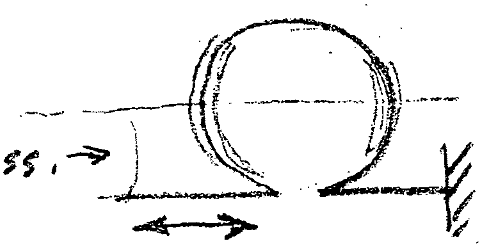
Many things were observed - mostly the type of motion. - stroke was about 0.250" - (oil pumping - too!)

Motion was so:-
at 900 RPM
~ 66 millisec



when drive was slowed up:

θ became less. →



Barrera
Watt
RB

So then we thot about ω_n and stuff like that.
and $\omega_n = \sqrt{k/m}$, $k \propto t^3$ and $m \propto t$

so $\omega_n \propto \sqrt{t^3/t} = t$ (maybe) we want ω_n high (we think.)

The ~~spring constant~~ natural frequency was est. of the above bello by Barrera hitting w/ hammer and Watt looking in Strobe light. Watt est. 10 c.p.s. = ω_n.

All previous stress calc - (Myall, Clark, NAA.) based on slow motion below ω_n - If this is case then deflections & strain should be equally distrib. around the 3" diameter, so.

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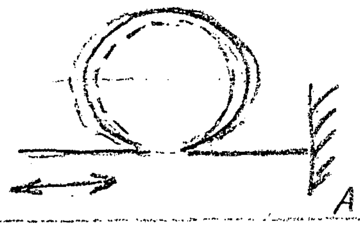
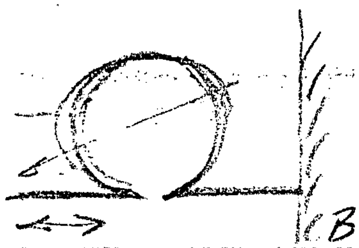
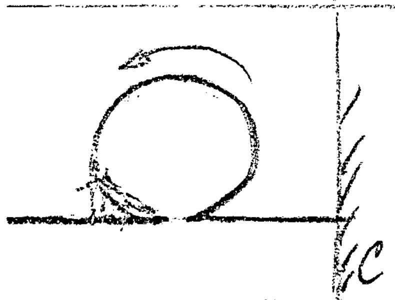
R. Byrns	ME	R	9 Feb 67



strain distrib. at slow motion below W_n.

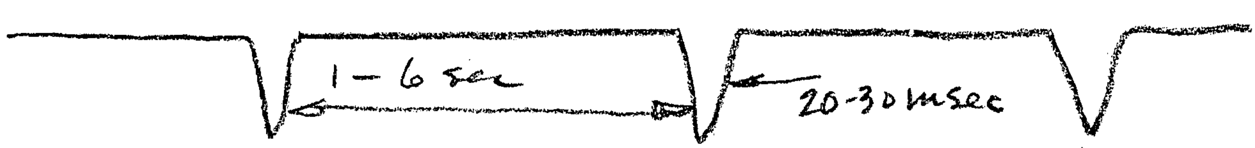


as forcing freq. is incr motion looks like so:



until we get to very high freq. or shock/impulse. We might expect motion to look like so!

Of course for a driven bello (continuous) Fig B above would be true because everything has started to dance in tune.
However bubble chamber operation impulse is not c.w., but pulsed. like so.



This form of drive might do something entirely different to the bellows. motion.

Also 2 other factors enter - one is that the shape is curved instead of straight which induces hoop stresses - (3 dimensional) and pressure - changes mass (effective) and profile and maybe even k.

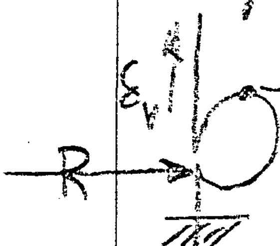
For the 25" bellows: we calc. it based on Mysll's report - (Clark, Timoshenko et al) and then tested it at ambient at about 1 cycle/sec. (probably below ω_n -)

We then installed one in 25" B.C. and have run it at ~ $\boxed{7 \times 10^6 \text{ cycles.}}$ - We dunno the motion, stress, or anything else, except it seems to work OK.

For the 82" Bello we calc. it based on Mysll's 25" B.C. report. and plan some testing.

Also the 82" Bello is racetrack shaped instead of circular - which does funny things to the hoop stresses, which are calculable for a circle.

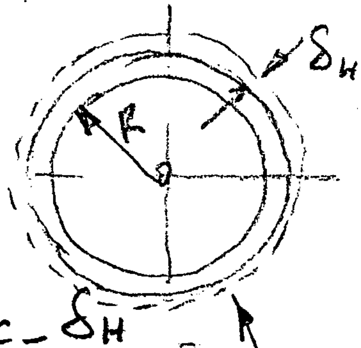
Mysll say:



$\delta_H = \frac{\delta_V}{2\pi}$ where $\frac{\delta_H}{R}$ is hoop strain

and hoop stress is = $\sigma = E\epsilon$

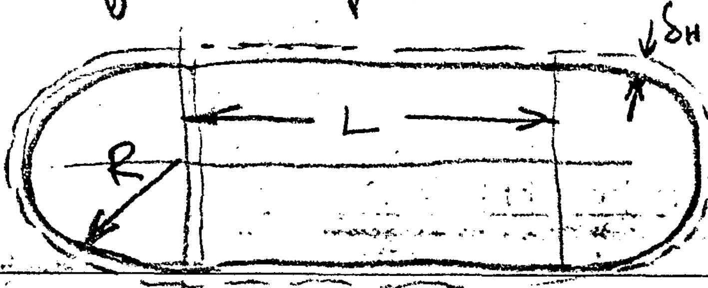
This gives a value for hoop stress and a 3 dimensional analysis.



$\epsilon = \frac{\delta_H}{R}$

FOR CIRCLE

However for a racetrack: Hoop strain is maybe much less a function of $\overline{R}$ and more of L



$\epsilon = \frac{\delta_H}{L} \to 0$

for race track

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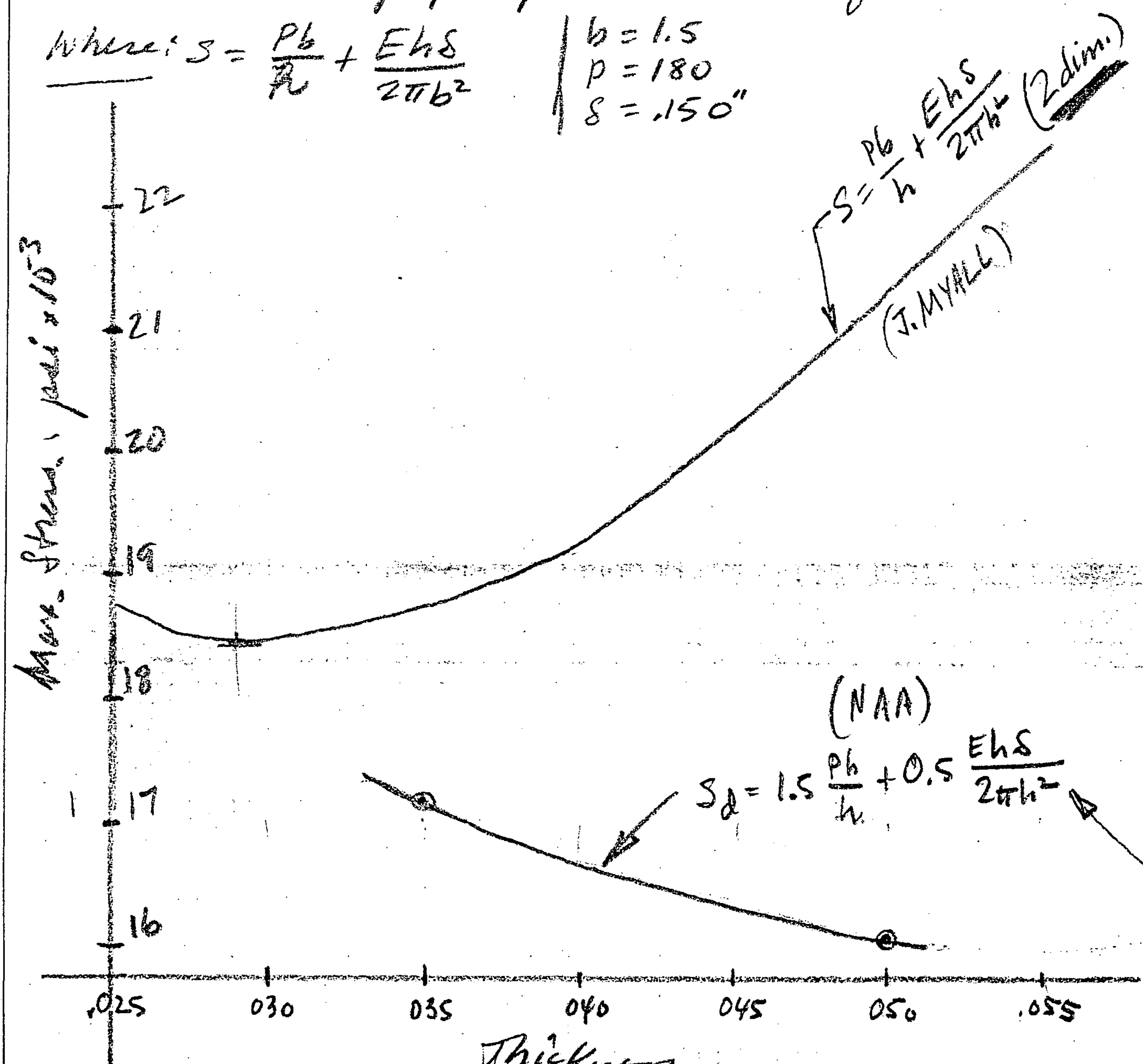
However the radius of curvature in the end zone of racetrack may determine the hoop stresses primarily, altho one could possibly say that the 2 dimensional analysis is more nearly correct than 3 dimensional for racetrack

If we look at graph of t v.s. stress. for 2 dim.

Where: $S = \frac{Pb}{h} + \frac{Eh\delta}{2\pi b^2}$ | $b = 1.5$, $p = 180$, $\delta = .150''$



We could plug in values for NAA-SR-4527 Vol I p 28 where $S_d = 1.5 S_{mp} + 0.5 S_B$ in 2 dim.

This all indicates that maybe procedures are somewhat established and st. foreward? for bellows static design, but dynamic design is different, and while on the 25" B.C. we have used static design as a basis for bellows and dynamic use of this bellows has been OK. ∴ Either our design procedure is OK or we are lucky.

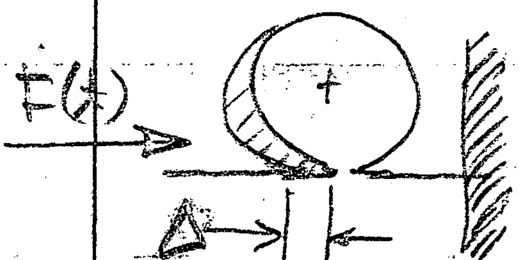
On the 25" Basis, Argonne and SLAC went ahead and designed a bellows. Their design procedure may be OK and they may be either lucky or unlucky.

I now feel that the static design method is not justified for a dynamic bellows, as the imposed strains are completely different from those assumed in stasis.

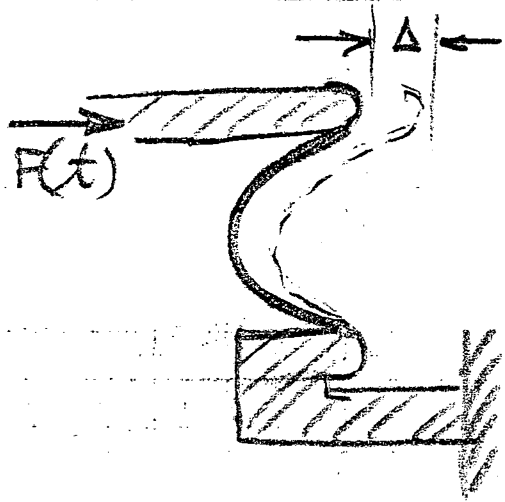
Also UCLRL, Argonne and SLAC have been barking up the wrong tree.

The more we learn the less we know — soon well be experts.

If only one half or less of bello profile moves under impulse - why try to put all XS mat'l in?



This Ω profil
difficult to make. as has been demonstrated.



just build the profil that moves under ~~load~~ impulse.
This profil looks like it it considerably easier to make!

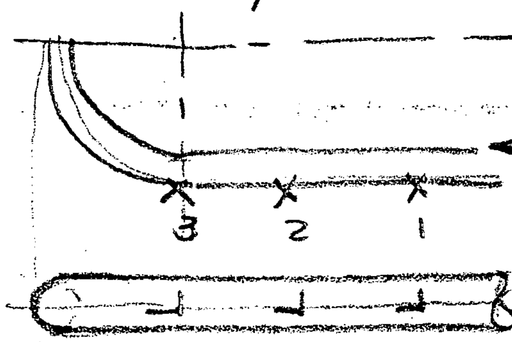
AUTHOR	DEPARTMENT	LOCATION	DATE
R Byms	Mechanical	Berkeley	20 Jun. 67

PROGRAM – PROJECT – JOB

TITLE

(LRL FABR)

Test of II Bellows. 7. Barrera put on strain rosettes – 3 places.



Meridian stress: for $\Delta \cong .150''$ was about 18000 psi

Horiz. stress was 1. $\simeq$ 0 2 $\simeq$ 50 psi 3 $\simeq$ 100 psi

FAR SIDE OF Bellows $\Delta \simeq 0.200''$

pressure $\simeq$ 80 psi.

AFTER ~ 500,000 CYCLES WELD CRACKED, (or metal just under weld) was rewelded and ~ 50,000 more cycles; cracked again ~ 2" away from original

CRACK IN ST. SECTION ABOUT 6" FROM ℄

Crack is on moving side, and side @ $\Delta = .200''$

RB sez spring constant is lower in st. sect than curved end. ∴ FORCE is lower ∴ stress should be lower? and it should crack in round end instead of st. section

Watt counters: sez hoop stresses (filaments) on round end protect weld and don't allow bending at weld area but push bending out into outer part of torus.

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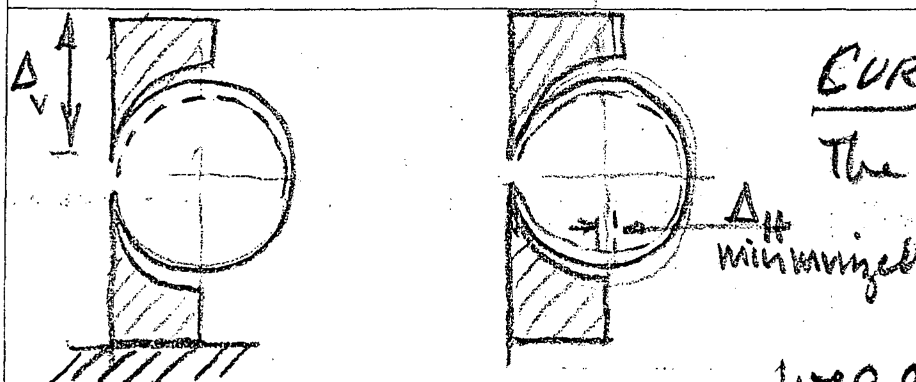
R. Byrns	ME	Berkeley	20 June 67

PROGRAM – PROJECT – JOB

TITLE



STRAIT SECTION.

Low Spring constant
Low Force to move.
Big excursion & I suspect that all. (2x) deflection occurs at moving side – or twice the stress on moving side calculated.?

CURVED END SECTION.

The Hoop stresses tend to keep XS bending away from the weld area, and push the motion out into the torus and distribute ~~the~~ Δ's more uniformly

This all sounds logical – but RB say what about the drill press models? You got 3×10^6 + pulses on them as st. section without failure? – at $\Delta \simeq .250''$ (of course no pressure.)

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ENGINEERING NOTE

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AUTHOR	DEPARTMENT	LOCATION	DATE
R Byrns.	Mechanical	Berkeley	20 June 67

PROGRAM – PROJECT – JOB

TITLE

GEOMETRIC PARAMETER ("sufficiently large & not too large" Dahl)

$$\mu = \frac{b^2\sqrt{12(1-\nu^2)}}{a\,h}$$

$b = 1.500''$
$\nu = 0.3$
$a = 15''$ { FOR curve ends
$a = \infty$ { FOR st. side
$h = .030''$
$.050''$

$$\mu = \frac{(1.5^2)\left[12\,(1-\overset{.91}{.09})\right]^{1/2}}{15\,(.030)} \qquad 10.94$$

$$\mu_{.030} = \frac{1.5\;(3.31)}{10\;(.030)} = \underline{16.6}$$

$$\mu_{.050} = \underline{9.92}$$

FOR $a = \infty$ (ST. SIDES)
$\mu \longrightarrow 0$

∴ It shouldn't break in st. section! *

FOR 25" Bellows.

$$\mu = \frac{(0.9)^2\,(3.31)}{(13.3)(2\times10^{-2})} = \frac{2.68\times10^{2}}{26.6} \simeq \underline{10.}$$

NAA-SR-9848 – results VALID as long as μ is not > 30.

* NAA-SR-9762 – page 6 – "fatigue life appears linear related to $\frac{1}{\mu}$. Fatigue life of high μ ($\mu = 6$) was less than 1/100 low μ ($\mu = 0.25$). Conventional high μ ($\mu \geq 2$) always fails at, or near, the welded junctions of bellows to pipe ends, where stresses are highest. With stresses very uniformly distributed over their profiles, low μ ($\mu \leq 0.25$) bellows appear to be insensitive to stress raising influences. The fatigue test of one toroid shape bellows was excellent."

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R. Byrns.	ME.	B	3 Jan 68

82" Bubble Chamber - Bellows. - "Post Engr Run"

Bellows was orig. designed for stroke ≃ 0.200"
This no. derived from:- 1% expansion-parameter.
Chamber depth is 17.3" - 1% ≃ 0.173"
However, we know that 25" B.C. uses a 0.75-0.80% expansion ratio.
So: (0.8)(.173) = 0.138" is realistic.
also: 82" B.C. is flared at bottom - 20" wide @ top. 24" wide @ bottom - ≃(17% wider) ∴ Bottom area is approx. ≃ 30% greater than top area.
So we can estimate real min. req'd stroke would be ≃ 0.100" — This was demonstrated during Engr Run - Dec. 1967. - Stroke was 0.100"

The present bellows installed in 82" B.C. is LRL fabricated of .049" x 3" O.D. S.S. tube Type 316.
The stroke of 0.100" and operating press ≃ 90 psia @ 30°K showed extend the estimated life out to ∞
Based on the endurance stress curves as shown on following pages; we have safety factors of 10-20 on variable stress, and S.F ≃ 20-40 on mean stress.
These no's based on simple 2 DIMENSIONAL ANALYSIS & North Am. Aviation modified formula. TRUE conditions maybe somewhere in between. 2 DIM. ANALYSIS (not 3 DIM.) is probably most nearly correct FOR RACETRACK BELLOW. All our failures in ambient tests occured in st. section, not ROUND ENDS.
THICKNESS for min. stress is I: .025" and for II: .044"
HEAVIER MAT'L. IS PREFERABLE FOR THIS WELDMENT.

AFTR ENGR RUN ; 82" BUBBLE CHAMBER -

OP'R CONDITIONS DEC. 14, 1967 ~ O.P ~ 70psig - 85psia

FOR H_2 SENSITIVE - V.P - 50psig

$\delta = 0.100"$

BELLOWS STRESS.

2 DIM. ANALYSIS. EN9312-03M4

$$S = \frac{Pb}{h} + \frac{Eh\delta}{2\pi b^2}$$

SUBST. $p = 90$, $\delta = 0.100"$

$h = .035"$

$$S = 3850 \pm 7400 = \boxed{11250\ psi}$$

+13K - 8K

$h = .050"$

$$S = 2700 \pm 10550 = \boxed{13250\ psi}$$

$S_m = \frac{S_{max} + S_{min}}{2}$

$S_m = \frac{5000}{2} = \boxed{2500}$

$S_v = \frac{S_{max} - S_{min}}{2}$

$S_v = \frac{21000}{2} = \boxed{105}$

IF WE APPLY NAA FORMULA :

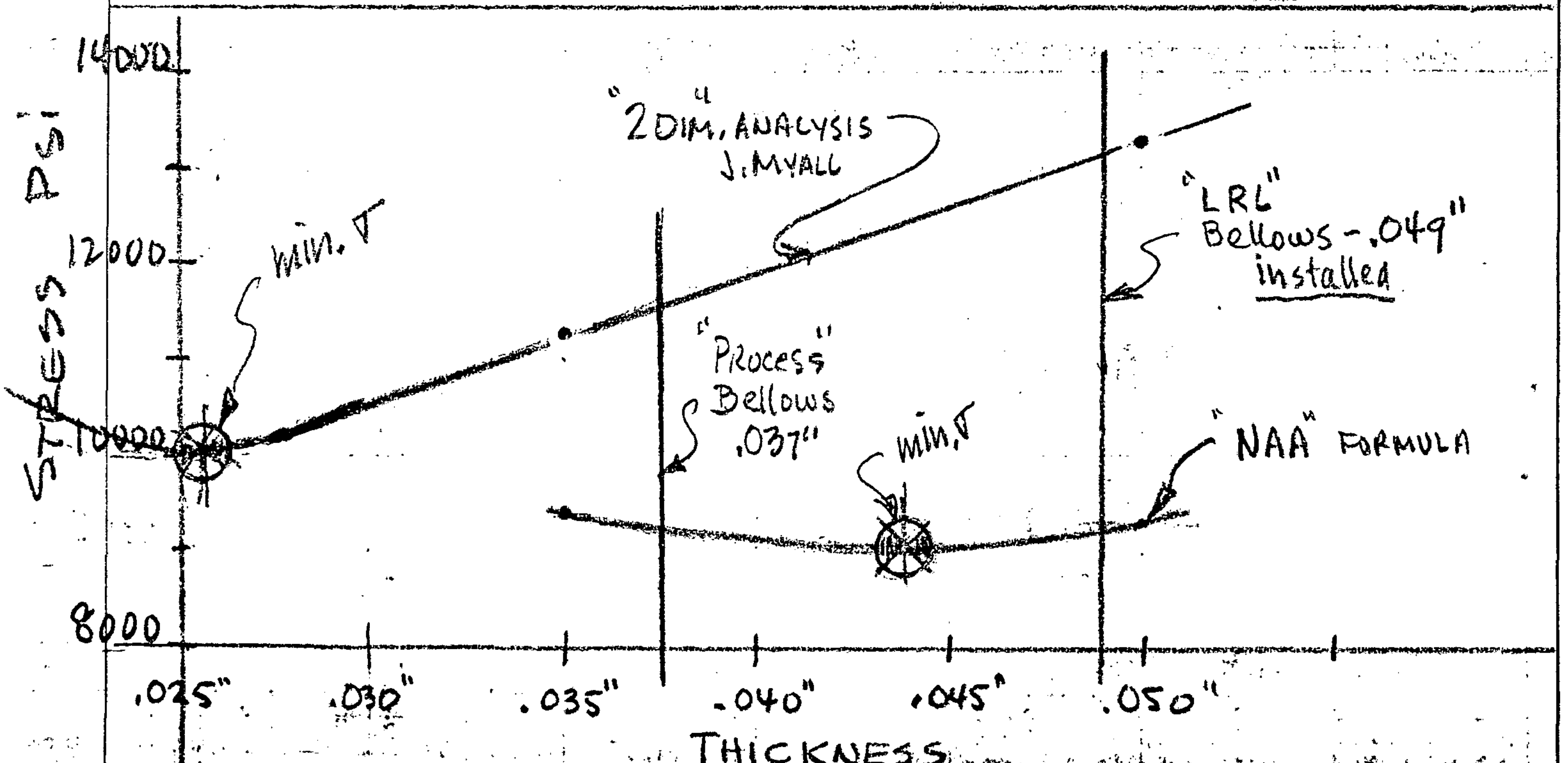
$$S = \underline{1.5}\frac{Pb}{h} + \underline{0.5}\frac{Eh\delta}{2\pi b^2}$$

$h = .035"$: $S = 5780 \pm 3700 = \boxed{9480\ PSI}$

$h = .050"$ $S = 4050 \pm 5280 = \boxed{9330\ PSI}$

$S_m \simeq 4500$

$S_v \simeq 5200$



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min value where; $\frac{Pb}{h} = \frac{Eh\delta}{2\pi b^2}$

$$\boxed{\frac{ds}{dh} = -\frac{Pb}{h^2} + \frac{E\delta}{2\pi b^2} = 0}$$

$$\frac{2\pi P b^3}{E\delta} = h^2 = \frac{2\pi(90)(1.5)^3}{(30\times10^6)\,10^{-1}}$$

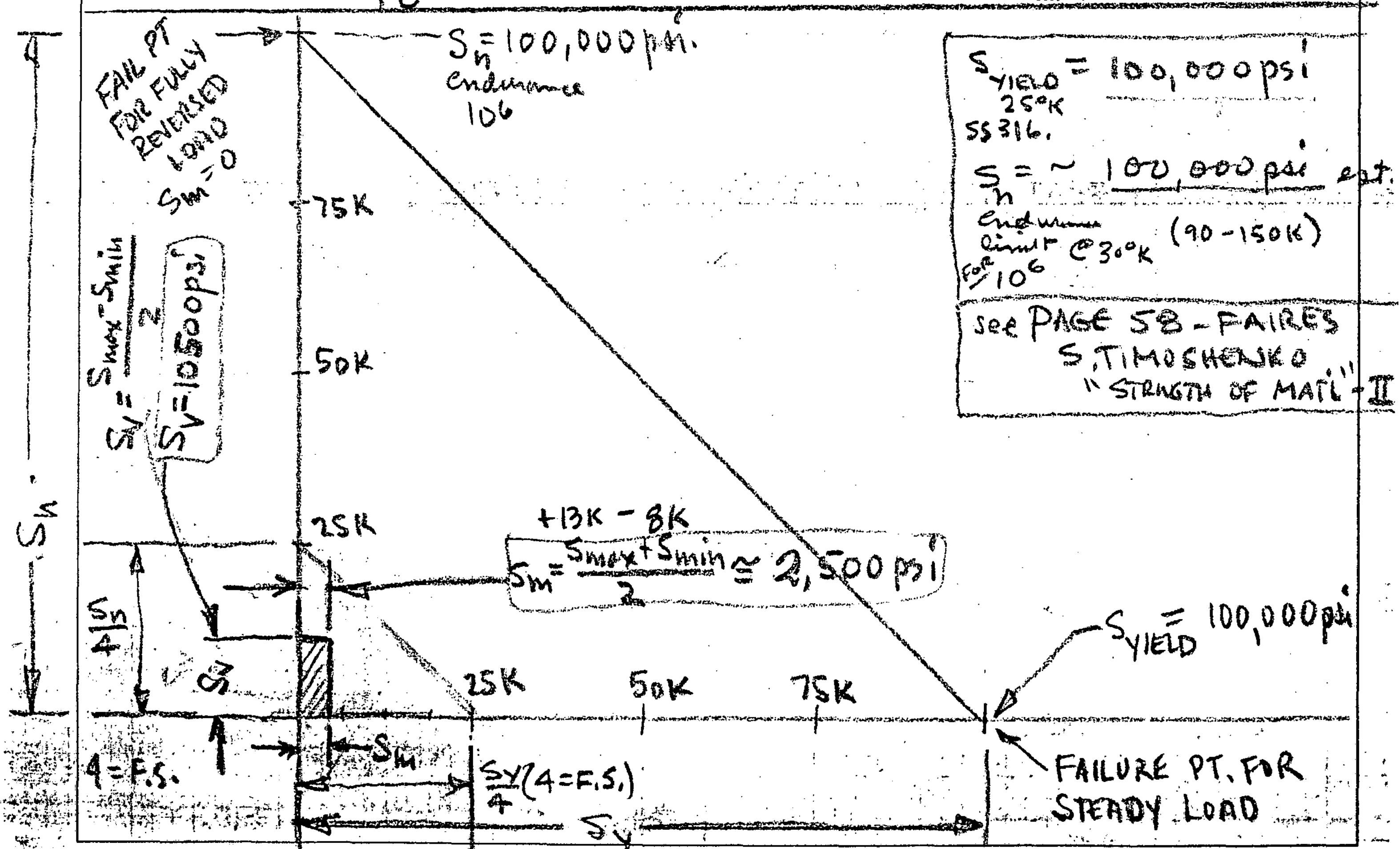
$$h^2 = \frac{636}{10^6} = 25.2^2 \times 10^{-6}$$

$$\boxed{h = .0252'' \quad \text{FOR MIN } S}$$

IF NAA FORM. USED:

$$1.5\,\frac{Pb}{h} = 0.5\,\frac{Eh\delta}{2\pi b^2} \qquad \frac{1.5}{0.5}\,\frac{Pb^2\,2\pi}{E\delta} = h^2$$

$$h^2 = 3 \cdot \frac{636}{10^6} = \qquad \boxed{h = .0436''} \text{ FOR min } S$$



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