john henry associates inc.

ANALYSIS OF A & M 24"
PURGE VALVE
BIG ROCK NPS

REPORT NO. JHA-79-138

Prepared For:

ATWOOD & MORRILL CO., INC.
285 Canal Street
Salem, Massachusetts 01970

April 14, 1980

Engineers / Structural Analysis & Applied Mechanics • 126 New Boston Park, Woburn, Massachusetts 01801 • (617) 933-3502

ANALYSIS OF A & M 24"
PURGE VALVE
BIG ROCK NPS

REPORT NO. JHA-79-138

PREPARED FOR:

ATWOOD & MORRILL CO., INC.
285 CANAL STREET
SALEM, MASSACHUSETTS 01970

APRIL 14, 1980

TABLE OF CONTENTS

1.0 INTRODUCTION

John Henry Associates was requested by Atwood & Morrill to analyse the 24" Purge Valve (2841-E) for the purpose of determining the upper bound of disc impact velocity. Since the valve has to open at least once to break the containment vacuum, plastic deformation in any of the parts should be held to acceptable levels.

2.0 SUMMARY AND CONCLUSIONS

The onset of deterioration of the neoprene disc seat was evident in the static load deflection tests at .125" deflection. This deflection was considered to be the basis of the limiting case for successful closure.

The resultant maximum velocity of the disc was 33.2 ft/sec. At this velocity some plastic deformation was shown to exist in the disc at the edge of the center disc post. The plastic strain was .6% strain, confined to the outer (surface) fibers. Strains as high as 30% are considered adequate for a single (faulted) condition.

The disc arm also showed slight yielding of .6% strain during impact. This was not considered sufficient to inhibit the subsequent operation of the valve.

The shaft appears to be adequate for all loadings, during a slam at 33.2 ft/sec. a plastic strain of 45% was experienced due to the torque placed in the end of the shaft by compression of the air in the air cylinder

The limiting item in air cylinder is the lever arm. The air cylinder tube has to be lengthened by <u>3.5</u>" inches to increase the amount of compressed volume. This reduces the air cylinder pressure to less than 304 psi.

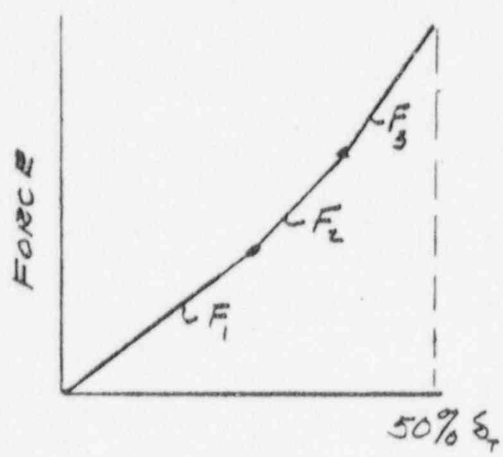
2.1 SEAT FORCES

THE STATIC LOAD-DEFLECTION TEST RESULTED IN ADVERSE TEARING OF THE NEOPRENE SEAT AT 50% OF TOTAL DEFLECTION. IT HELD TOGETHER BUT WAS BADLY CUT BY THE BODY SEAT. THEREFORE THE VULNERABILITY OF THE DISC SEAT WILL BE LIMITED TO 50% DEFLECTION.

FROM FIGURE 1

$\delta_T = .25''$

$50\% \, \delta_T = .125''$



$$F_1 = A\delta = \left\{\frac{15000}{.074-.02}\right\}\delta$$

$$F_1 = (277{,}777)\delta$$

THE EFFECTIVE LENGTH OF THE DISC SEAT MATERIAL TESTED IS THE LENGTH OF THE FIXTURE ($8\frac{7}{8}$") MINUS DOUBLE THE NEOPRENE THICKNESS TO ACCOUNT FOR THE FREE END EDGE EFFECTS.

$$\text{LENGTH } L = 8.875 - .5$$

$$L = 8.375''$$

$$f_1 = \frac{F_1}{L} = (33167)\delta \ (\#/\text{in.}) \quad ; \quad 0 \leq \delta \leq .054$$

$$F_2 = -A + B\delta \quad ; \quad B = \frac{35000}{.097 - .023} = 472973$$

$$A = (472973)(.023) = 10878$$

$$F_2 = -10878 + (472973)\delta$$

$$f_2 = \frac{F_2}{L} = -1299 + (56474)\delta \quad ; \ .054 \leq \delta \leq .097$$

$$F_3 = -A + B\delta \quad ; \quad B = \frac{55000}{.110 - .053} = 797101.4$$

$$A = (79101.4)(.053) = 42246.4$$

$$F_3 = -42246.4 + (797101.4)\delta$$

$$\underline{f_3 = \frac{F_3}{L} = -5044 + (95176.3)\delta \quad ; \quad .097 \leq \delta \leq .125}$$

TOTAL ENERGY IN DISC SEAT @ 50% δ_T

$$E_s = \pi D \int_0^{.125} f_m \, d\delta = \pi D \left\{ \int_0^{.054} f_1 \, d\delta + \int_{.054}^{.097} f_2 \, d\delta + \int_{.097}^{.125} f_3 \, d\delta \right\}$$

$$\int_0^{.054} f_1 \, d\delta = (33167) \int_0^{.054} \delta \, d\delta = (33167) \frac{(.054)^2}{2} = \underline{48.36}$$

$$\int_{.054}^{.097} f_2 \, d\delta = (-1299)(.097 - .054) + (56474) \frac{(.097^2 - .054^2)}{2}$$

$$= \underline{127.5}$$

$$\int_{.097}^{.125} f_3 \, d\delta = (-5044)(.125 - .097) + (95176.3) \frac{(.125^2 - .097^2)}{2}$$

$$= \underline{154.6}$$

$$E_S = \pi D \sum f_m \delta = \pi D \{ 48.36 + 127.5 + 154.6 \}$$

$D = 24.625''$ (AVE. BODY SEAT DIA.)

$$E_S = (77.36)(330.46)$$

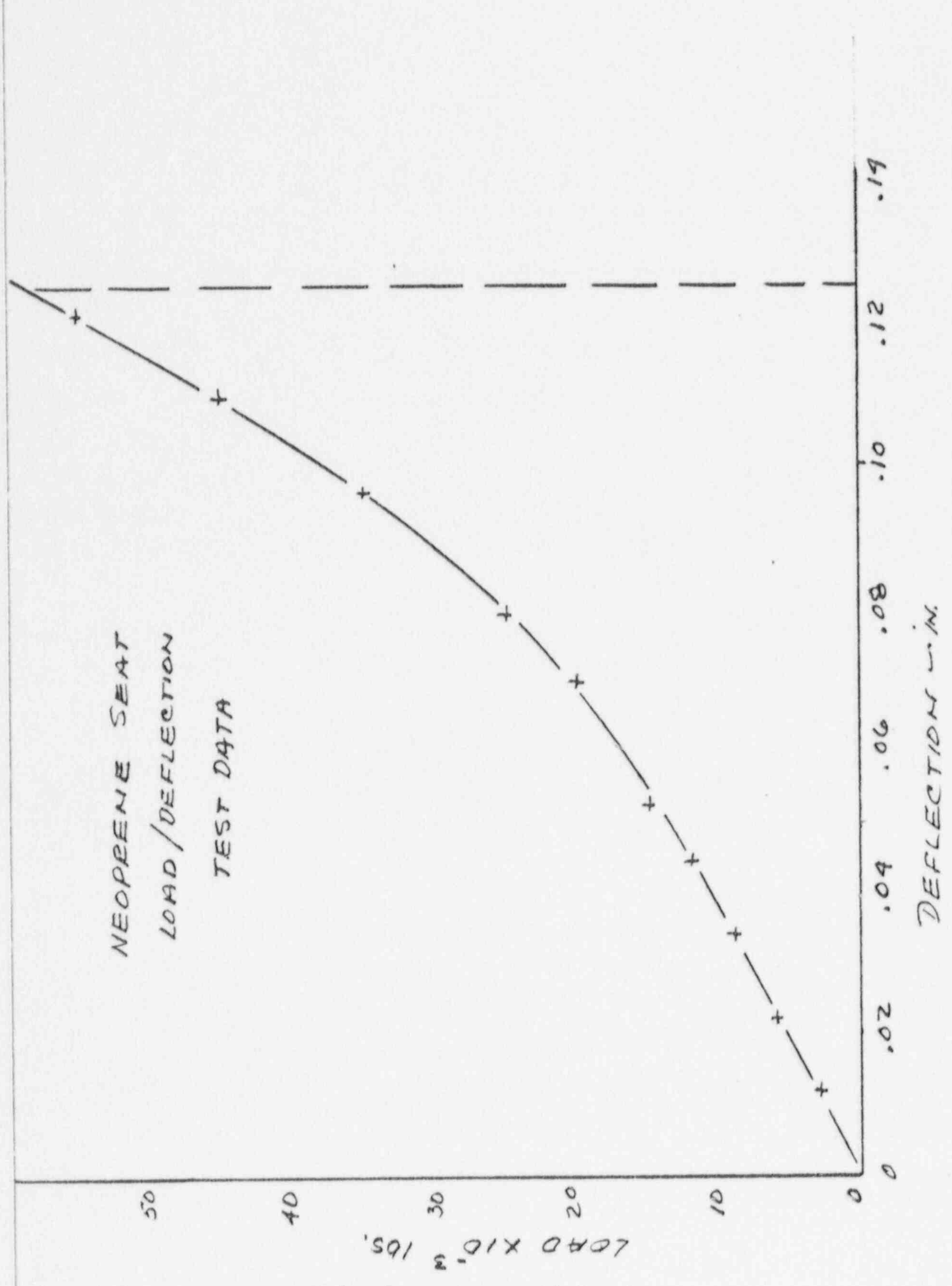
$\underline{E_S = 25564.4}$ IN-#

MAX. FORCE ON DISC ASSEMBLY

$$F_M = \pi D f_3 \quad @ \ \delta = .125$$

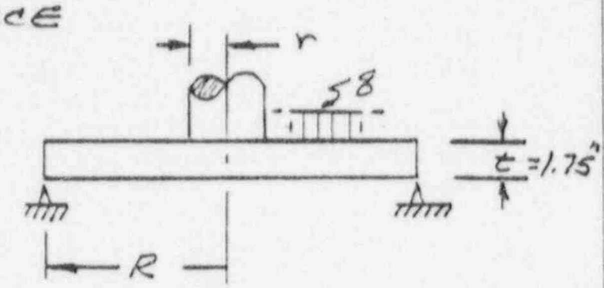
$$F_M = \pi(24.625)\{-5044 + (95176.3)(.125)\}$$

$\underline{F_M = 530162.8^{\#}}$ ($w = 6853$ #/IN. CIR.)


NEOPRENE SEAT
LOAD/DEFLECTION
TEST DATA
DEFLECTION ~ IN.
0
.02
.04
.06
.08
.10
.12
.14
LOAD X10^-3 lbs.
0
10
20
30
40
50

3.2 DISC

ALTHOUGH THE SEAT FORCE IS DERIVED FROM THE INERTIA OF THE DISC AT IMPACT, WE ARE ASSUMING A EQUIVALENT UNIFORM PRESSURE ACTING OVER THE DISC AS THE INERTIA FORCE IN EQUILIBRIUM WITH THE SEAT FORCE.



$W = 530162.8^{\#}$

$q = \dfrac{W}{\pi(R^2 - r^2)} = 1130 \ \#/in^2$

STRESS $\sigma_{MAX.} = k \dfrac{q R^2}{t^2}$ (REF. TIMOSHENKO PLTS & SHELLS)

$k = 1.86$, $R = 12.3125"$

$\underline{\sigma = 104{,}038 \text{ psi}}$

DEFLECTION $w = k_1 \dfrac{q R^4}{E t^3}$; $k_1 = .64$; $E = 29.25 \times 10^6$ psi

$\underline{w = .106"}$

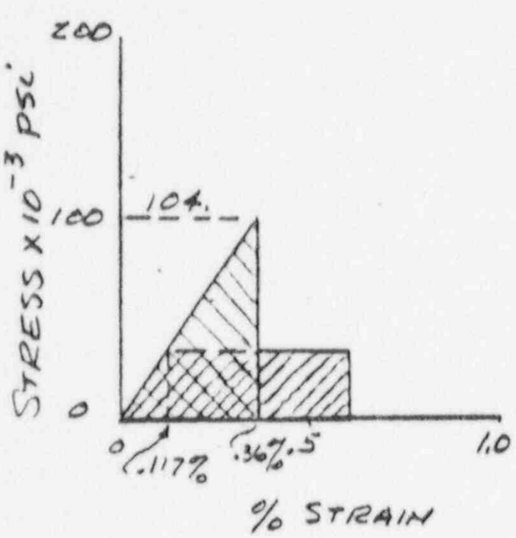
DISC MAT'L IS A-515 GR 70 (.35% C)

@ TEMP = 250°F

σ_y = 34150 psi

E = 29.25×10^6 psi

SINCE SIGNIFICANT YIELDING TAKES PLACE, THE AMOUNT OF PLASTIC STRAIN WILL BE ESTIMATED ASSUMING (CONSERVATIVELY) THE MATERIAL ELASTIC - PERFECTLY PLASTIC.



AREA ($\sigma\varepsilon$) OF 

$A_\Delta = \frac{1}{2}(.239)(69888) = 8352$

EQUIVALANT AREA UNDER

σ_y = 34150 psi.

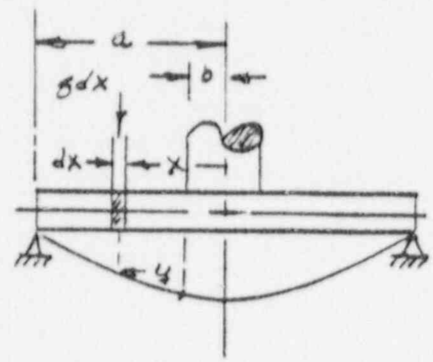
$A_\Delta = (34150)\varepsilon_e = 8352$

$\varepsilon_e = .245\%$

$\varepsilon_r = .36 + .245 = .605\%$

THE DISC ENERGY IS CALCULATED FROM THE WORK DONE BY q (THE ASSUMED INERTIA FORCE) ACTING THROUGH THE DISC DISPLACEMENT.

$$E_D = \int_b^a q y \cdot 2\pi x \, dx$$



$q = 1367.8$ #/in²

$y = y_0 \cos \frac{\pi}{2} \cdot \frac{x}{a}$ (ASSUMED DEFL.)

$a = 12.3125''$

$b = 1.5''$

WHEN $x = b$, $y = .106''$

$.106 = y_0 \cos(10.965°)$

$y_0 = .108''$

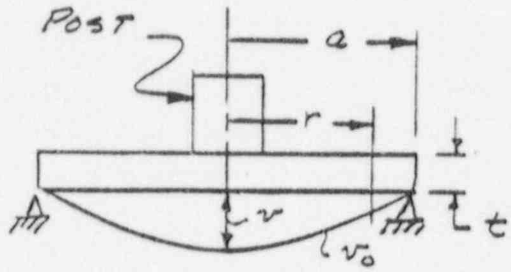
$$E_D = (.108) \cdot 2\pi q \int_b^a \cos \frac{\pi x}{2a} \cdot x \cdot dx$$

$$E_D = (766.6) \left[\frac{4a^2}{\pi^2} \cos \frac{\pi}{2a} x + \frac{2a}{\pi} x \sin \frac{\pi}{2a} x \right]_b^a$$

$$E_D = (766.6) \left[0 - \frac{4a^2}{\pi^2} \cos\left(\frac{\pi}{2}\frac{b}{a}\right) + \frac{2a^2}{\pi} - \frac{2a}{\pi} b \sin\left(\frac{\pi}{2}\frac{b}{a}\right) \right]$$

$E_v = 26033$ in-#

DECELERATION OF DISC.



FREQUENCY OF THE DISC.

$v = v_0 \cos p\tau$

$\dot{v} = -v_0\, p \sin p\tau$

$(\dot{v})_{MAX} = v_0 p$

$v_0 = F(r) = a_1\left(1 - \frac{r^2}{a^2}\right)$ (ASSUMED SHAPE)

POTENTIAL ENERGY FOR SYMMETRICAL DEFLECTION IS,

$$V = \pi D \int_0^a \left\{\left(\frac{d^2v}{dr^2} + \frac{1}{r}\frac{dv}{dr}\right)^2 - 2(1-\nu)\frac{d^2v}{dr^2}\cdot\frac{1}{r}\frac{dv}{dr}\right\} r\,dr$$

$$V = \pi D \int_0^a \left\{\left(\frac{d^2v_0}{dr^2} + \frac{1}{r}\frac{dv_0}{dr}\right)^2 - 2(1-\nu)\frac{d^2v_0}{dr^2}\cdot\frac{1}{r}\frac{dv_0}{dr}\right\} r\,dr$$

$$\frac{dv_0}{dr} = -\frac{2a_1 r}{a^2}$$

$$\frac{d^2v_0}{dr^2} = -\frac{2a_1}{a^2}$$

$$V = \pi D \int_0^a \left\{ \left(-\frac{2a_1}{a^2} + \frac{1}{r}\cdot\left[-\frac{2a_1 r}{a^2}\right]\right)^2 - 2(1-\nu)\left[-\frac{2a_1}{a^2}\right]\cdot\frac{1}{r}\left[-\frac{2a_1 r}{a^2}\right]\right\} \cos^2 pt\, r\,dr$$

$$V = \pi D \cos^2 pt \int_0^a \left\{ \left(-\frac{4a_1}{a^2}\right)^2 - 2(1-\nu)\left[4\frac{a_1^2}{a^4}\right]\right\} r\,dr$$

$$V = \pi D \cos^2 pt \int_0^a \left\{ 16\frac{a_1^2}{a^4} - 2(1-\nu)\, 4\frac{a_1^2}{a^4}\right\} r\,dr$$

$$V = 8\pi D \frac{a_1^2}{a^4} \cos^2 pt \int_0^a \left\{ 2r - (1-\nu) r\right\} dr$$

$$V = 8\pi D \frac{a_1^2}{a^4}(1+\nu)\frac{a^2}{2}\cos^2 pt$$

$$V_{MAX} = 4\pi D \frac{a_1^2}{a^2}(1+\nu)$$

KINETIC ENERGY

$$KE = \frac{\pi \gamma t}{g}\int_0^a \dot{v}^2 r\,dr + \frac{1}{2} m_p \dot{v}^2$$

m_p = MASS OF CENTER POST ON DISC.

$$\dot{v} = \frac{dv}{dt} = -v_0\, p \sin pt$$

$$(\dot{v})_{MAX} = -v_0\, p$$

$$KE = \frac{\pi \gamma t}{g} p^2 \int_0^a v_0^2 r\,dr + \frac{1}{2} m_p v_0^2 p^2$$

$$KE = \frac{\pi \gamma t p^2 a_1^2}{g} \int_0^a \left(1 - \frac{r^2}{a^2}\right)^2 r\,dr + \frac{1}{2} m_p a_1^2 p^2$$

$$\int_0^a \left\{1 - \frac{2r^2}{a^2} + \frac{r^4}{a^4}\right\} r\,dr = \int_0^a \left\{r - \frac{2r^3}{a^2} + \frac{r^5}{a^4}\right\} dr$$

$$\int \Delta = \left[\frac{r^2}{2} - \frac{2r^4}{4a^2} + \frac{r^6}{6a^4}\right]_0^a$$

$$\int \Delta = \frac{a^2}{2} - \frac{a^2}{2} + \frac{a^2}{6} = \frac{a^2}{6}$$

$$\therefore\ KE = \frac{\pi \gamma t p^2 a_1^2 a^2}{6g} + \frac{1}{2} m_p a_1^2 p^2$$

$$\frac{\partial}{\partial a_1}\left\{4\pi D \frac{a_1^2}{a^2}(1+\nu) - \frac{\pi \gamma t p^2 a_1^2 a^2}{6g} - \frac{1}{2} m_p a_1^2 p^2\right\} = 0$$

$$\frac{8\pi D}{a^2}(1+\nu) - \frac{\pi \gamma t a^2 p^2}{3g} - m_p p^2 = 0$$

$$p^2\left\{\frac{\pi \gamma t a^2}{3g} + m_p\right\} = \frac{8\pi D(1+\nu)}{a^2}$$

$$p = \sqrt{\frac{8\pi D(1+\nu)/a^2}{\frac{\pi \gamma t a^2}{3g} + m_p}}$$

$$p = \sqrt{\frac{24\pi(1+\nu) g D}{\pi \gamma t a^4 + 3 g m_p a^2}}$$

$$p = \sqrt{\frac{24(1+\nu) g D}{\gamma t a^4 + \frac{3 W_p a^2}{\pi}}}$$

$$p = \frac{5.5857}{a^2}\sqrt{\frac{D g}{\gamma t + \frac{3 W_p}{A}}}$$

$A = \pi a^2$

$A = 476.258 \text{ in}^2$

$D = \frac{E t^3}{12(1-\nu^2)} = 1.4355 \times 10^7$

$g = 386 \text{ in}/s^2$

$t = 1.75''$

$W_p = \frac{\pi}{4}(3)^2(7.78125)(.283)$

$W_p = 15.566\#$ (WEIGHT OF CENTER POST)

$a = 12.3125''$

$\gamma t = (.283)(1.75) = .49525 \#/\text{in}^2$ (WEIGHT PER UNIT AREA)

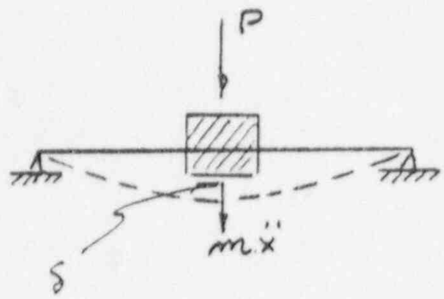
$\underline{p = 3560.76}$ RAD/SEC

$f = 566.7$ CPS.

SINGLE DEGREE OF FREEDOM ANALOGY

DISC SPRING CONSTANT

$\delta = k_1 \dfrac{Pa^2}{Et^3}$ (REF: TIMO. PLTS & SHELLS)



$a = 12.3125''$

$k_1 = .64$

$E = 29.25 \times 10^6$ psi

$t = 1.75''$

$$K_D = \frac{P}{\delta} = \frac{Et^3}{a^2 k_1}$$

$$K_D = 1.6157 \times 10^6 \text{ \#/in.}$$

AND $p^2 = \dfrac{K_D}{m}$

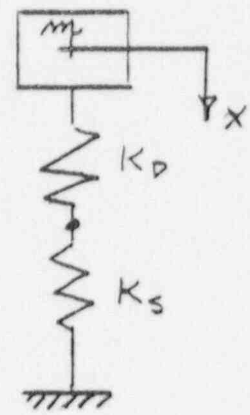
$$m = \frac{K_D}{p^2} = .12743 \text{ \# s}^2\text{/in}$$

SEAT SPRING CONSTANT

$$K_S = \frac{530,160}{.125} = 4.2413 \times 10^6 \text{ \#/in.}$$

MODEL OF DISC AND SEAT

EQUIVALENT K_e



$$\frac{1}{K_e} = \frac{1}{K_D} + \frac{1}{K_S}$$

$$K_e = \frac{K_D \cdot K_S}{K_D + K_S}$$

$$K_e = 1.17 \times 10^6 \text{ \#/in.}$$

SINGLE DEGREE OF FREEDOM OF DISC IS

$$p = \sqrt{\frac{K_D}{m}} = 3560.76 \text{ RAD/S}$$

$$m = \frac{K_D}{p^2} = \frac{1.6157 \times 10^6}{(3560.76)^2}$$

$$m = .12743 \text{ \# S}^2\text{/in}$$

$$\therefore \; m\ddot{x} + K_e x = 0$$

WHERE $\omega_e = \sqrt{\frac{K_e}{m}}$

ASSUME $X = A\cos W_e t + B\sin W_e t$

at $t=0$, $\dot{x} = V_0$

SO. $x = \frac{V_0}{W_e}\sin W_e t$

AND $\ddot{x} = -V_0 W_e \sin W_e t$

$\underline{(\ddot{x})_{MAX} = V_0 \omega_e}$ DECELERATION OF DISC

$$\omega_e = \sqrt{\frac{K_e}{m}} = \sqrt{\frac{1.17\times10^6}{.08155}}$$

$W_e = 3030.1$ RAD/SEC.

CLOSING VELOCITY OF DISC

TOTAL STRAIN ENERGY IN DISC AND DISC SEAT.

$E_T = E_S + E_D = 25564.4 + 26033$

$E_T = 51597.4$ #-in

$E_T = \frac{1}{2} M . V_0^2$

$$V_0 = \sqrt{\frac{2E_T}{M}}$$

$M = M_D + M_P$; $R = 12.3125''$

$t = 1.75''$

M_D = DISC MASS $\rho = .283\ \#/in^3$

M_P = CENTER POST MASS

$$M_D = \frac{\pi R^2}{g} t \rho = .6111\ \#\text{-}s^2/in$$

$$M_P = \frac{\pi d^2}{4g} L \rho \quad ; \quad d = 3.0'',\ L = 7.78125''$$

$M_P = .0403\ \#\text{-}s^2/in$

$\therefore\ M = .6514\ \#\text{-}s^2/in$

AND

$$V_0 = \sqrt{\frac{(2)(51597.4)}{(.6514)}}$$

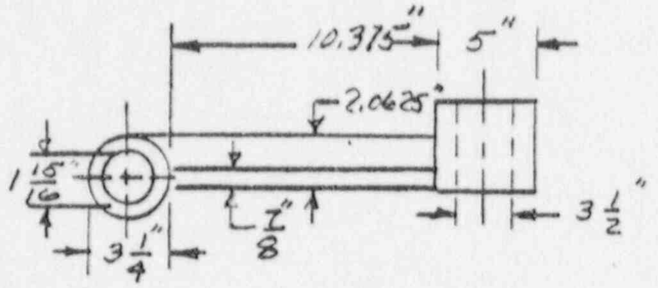
$\underline{V_0 = 398.02\ ''/s\ \text{(OR 33.2 FT/SEC)}}$

DECELERATION OF DISC. (TRANSLATION ACCEL)

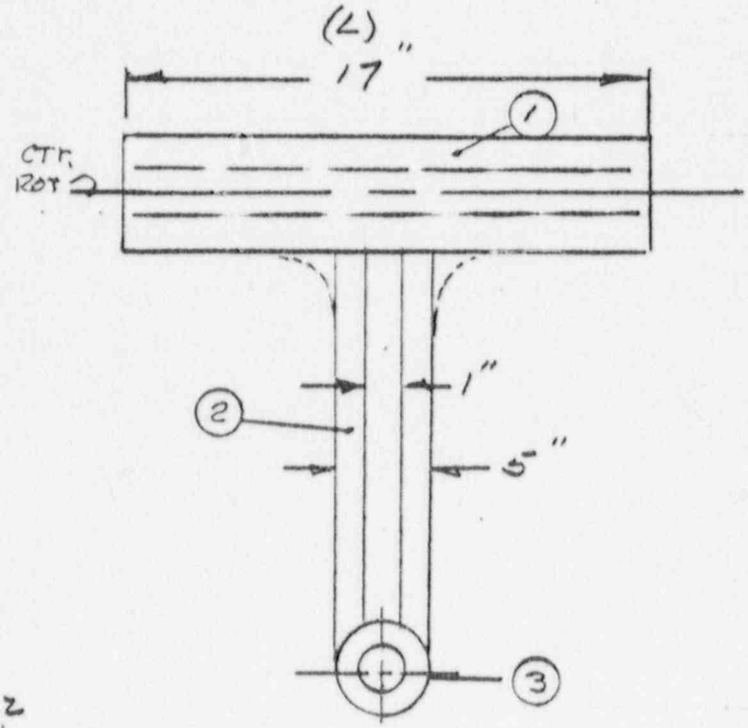
$\underline{(\ddot{X})_{MAX} = V_0 \omega_e = 1,206,041\ in/s^2}$

3.3 ARM EVALUATION (AM 8258-C)

PROPERTIES OF ARM (A-216, WCB)



MASS MOMENT OF INERTIA ABOUT CTR. OF ROTATION



(1)

$I_1 = \frac{1}{2} m (R_2^2 + R_1^2)$

$m_1 = \frac{\pi}{4}(D_o^2 - D_i^2) L \rho$

$m = 25.727\#$

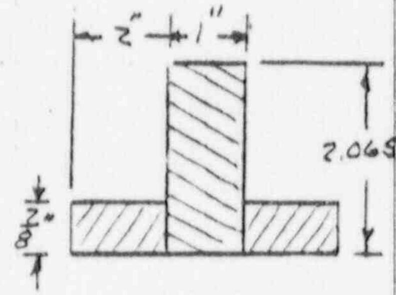
$I_1 = 46.039 \ \# - in^2$

(2) $m_2 = A L \rho$; $L = 10.375"$

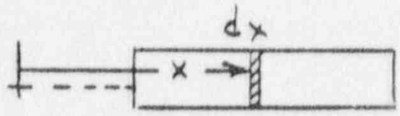
AREA $A = (1)(2.0625) + (2)(2)(.875)$

$A = 5.5625 \ in^2$

$m_2 = 16.332\#$



$$I_2 = \int_{1.625''}^{12''} x^2 dm$$



$$dm = A\rho\, dx$$

$$I_2 = A\rho \int x^2 dx = A\rho \left.\frac{x^3}{3}\right|_{1.625}^{12} = \frac{A\rho}{3}\left\{12^3 - 1.625^3\right\}$$

$$I_2 = 904.48 \text{ \#-in}^2$$

(3) $m_3 = \frac{\pi}{4}(D_o^2 - D_i^2) L \cdot \rho$; $L = 5.0''$

$$m_3 = 14.17 \text{ \#}$$

$$I_3 = m_3 r^2 = (14.17)(14.5)^2$$

$$I_3 = 2979.15$$

$$I_o = \sum_1^3 I_n$$

$$I_o = \underline{3929.67 \text{ \#-in}^2}$$

RADIUS OF GYRATION k,

$$k = \sqrt{\frac{I}{m}} = 8.36''$$

DISC CLOSING VELOCITY $V_0 = 33.2 \text{ FT/S}$

$\omega = \frac{V_0}{r}$; $r = 14.5''$

$\omega = \frac{33.2 \times 12}{14.5} = 27.5 \text{ RAD/SEC.}$

DISC ARM ENERGY AT INSTANT OF IMPACT U

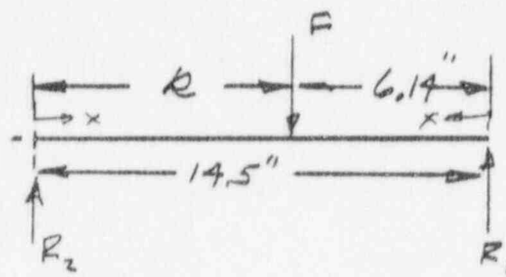
$U = \frac{1}{2} I_0 \omega^2$

$U = \left(\frac{1}{2}\right)\left(\frac{3929.67}{g}\right)(27.5)^2$

$\underline{U = 3843 \text{ in-\#}}$

R = RADIUS OF GYRATION

R = 8.36"



INTERNAL ENERGY

$$E_0 = \frac{1}{2EI}\int_0^{8.36} M_2^2\,dx + \frac{1}{2EI}\int_0^{6.14} M_1^2\,dx$$

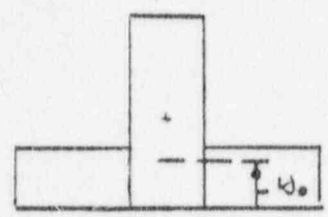
$M_2 = R_2 x$; $M_1 = R_1 x$

$$E_0 = \frac{1}{2EI}\left\{ R_2^2 \frac{x^3}{3}\Big|_0^{8.36} + R_1^2 \frac{x^3}{3}\Big|_0^{6.14} \right\}$$

$$E_0 = \frac{1}{2EI}\left\{ (194.8) R_2^2 + (77.16) R_1^2 \right\} \quad ; E = 29.25\times10^6$$

I = ARM MOM. OF INERTIA

$N.A. = y_0$



$$y_0 = \frac{(1)(2.065)(\frac{2.065}{2}) + (4)(.875)(\frac{.875}{2})}{(2.065) + (4)(.875)}$$

$y_0 = .6583''$

$$I = \frac{(1)(2.065)^3}{(12)} + (2.065)(\frac{2.065}{2} - .6583)^2 + \frac{(4)(.875)^3}{(12)} + (4)(.875)(.6583 - .4375)^2$$

$\underline{I = 1.4169\ in^4}$

$$E_0 = (2.3496\times10^{-6}) R_2^2 + (9.3087\times10^{-7}) R_1^2$$

$$R_1 = \left(\frac{8.36}{14.5}\right) F = (.577) F$$

$$R_2 = \left(\frac{6.14}{14.5}\right) F = (.423) F$$

$$E_0 = 9.9495\times10^{-7} F^2 + 5.3669\times10^{-7} F$$

$$E_0 = 1.5316\times10^{-6} F^2$$

AND $E_0 = U$

$$(1.5316 \times 10^{-})F^2 = 3843$$

$$\underline{F = 50091^{\#}}$$

$R_1 = 28880^{\#}$

$R_2 = 21211^{\#}$

MAX. STRESS IN ARM

$$\sigma = \frac{Mc}{I} = \frac{(28880)(6.14)(1.407)}{(1.4169)}$$

$\sigma = 176{,}047$ psi ($\sigma_{ALL} = 34150$ psi)

YIELDING TAKES PLACE

APPROXIMATE PLASTIC STRAIN

$$\varepsilon = \frac{\sigma}{E} = \frac{176047}{29.25 \times 10^6} \cdot 10^2 = .60\% \text{ STRAIN}$$

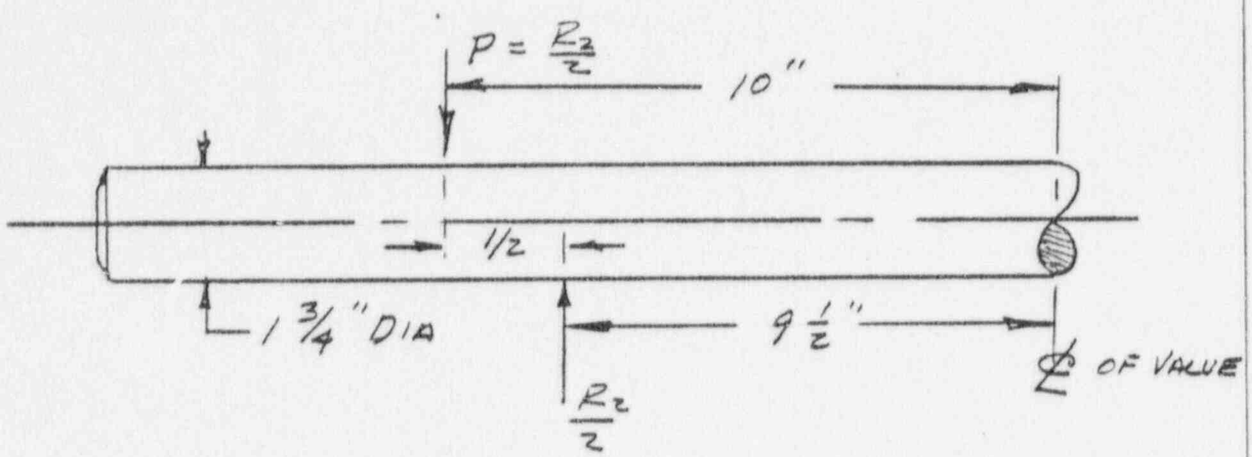
THIS AMOUNT OF PLASTIC STRAIN IS ACCEPTABLE.

3.4 SHAFT EVALUATION (20329-B)

MAT'L ; S.S.-410, $\sigma_y = 80000$ psi

LOAD FROM DISC ARM (SEC. 3.3)

$R_2 = 21211$ #



SHEAR STRESS (TRANSVERS)

$$\sigma_s = \frac{\frac{R_2}{2}}{A} \; ; \; A = 2.405 \text{ in}^2$$

$\sigma_s = 4409$ psi ($\sigma_{ALL_S} = 45600$ psi)

BENDING STRESS

$$\sigma_B = \frac{Mc}{I} \; ; \; I = \frac{\pi d^4}{64} = .46 \text{ in}^4$$

$$c = .875" \; ; \; M = \frac{R_2}{2} \cdot \left(\frac{1}{2}\right) = \frac{R_2}{4}$$

$\sigma_B = 10{,}087$ psi ($\sigma_{ALL} = 80000$ psi)

TORSION OF THE SHAFT DUE TO AIR CYLINDER PRESSURE

$M_o = 53467$ in-#

$\tau = \dfrac{M_o r}{I_p}$; $r = .875''$; $I_p = \dfrac{\pi d^4}{32} = .920$

$\tau = 51000$ psi

$\tau_{ALL} = 45600$ psi

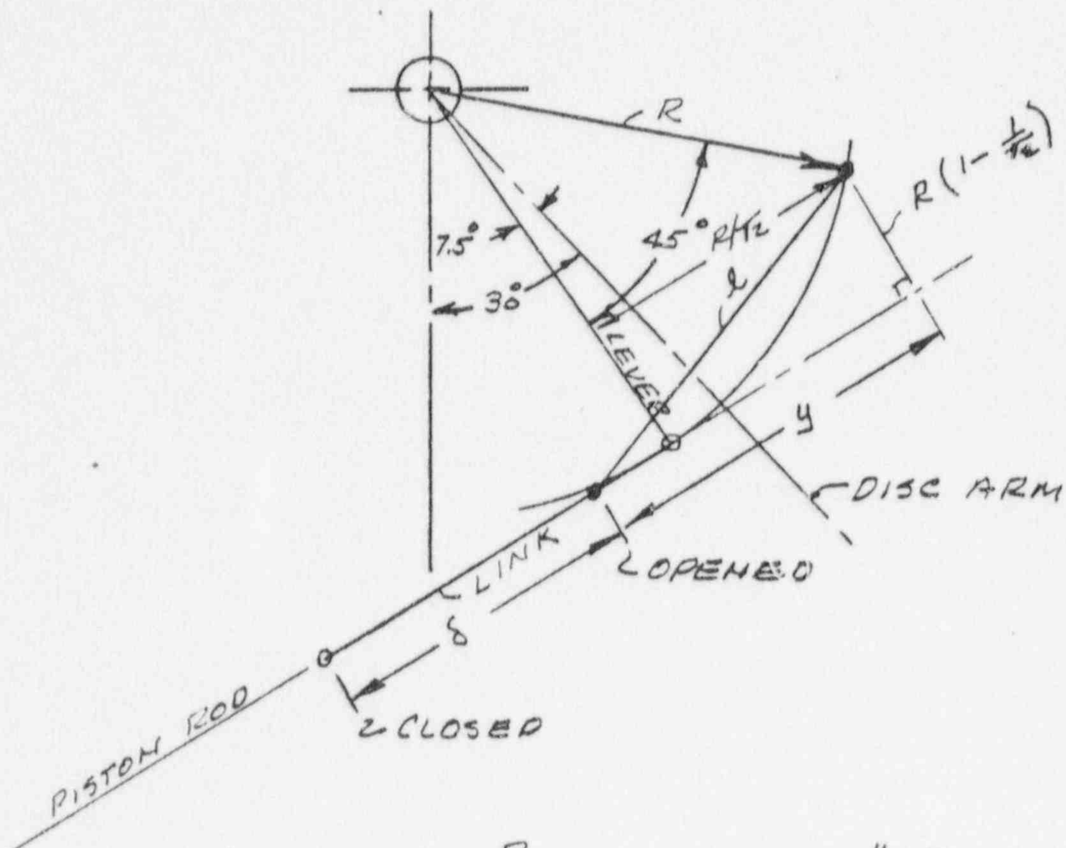
SLIGHT YIELDING TAKE PLACE

$\varepsilon = \dfrac{\tau}{G} = \dfrac{51000}{11.25 \times 10^6} = .45\%$

3.5 AIR CYLINDER (6" DIA.)

KINETICS OF AIR CYLINDER.

TRAVEL OF PISTON ROD



$$\delta + y = l + \frac{R}{\sqrt{2}} \quad ; \quad l = 8.25'', R = 8.0625''$$

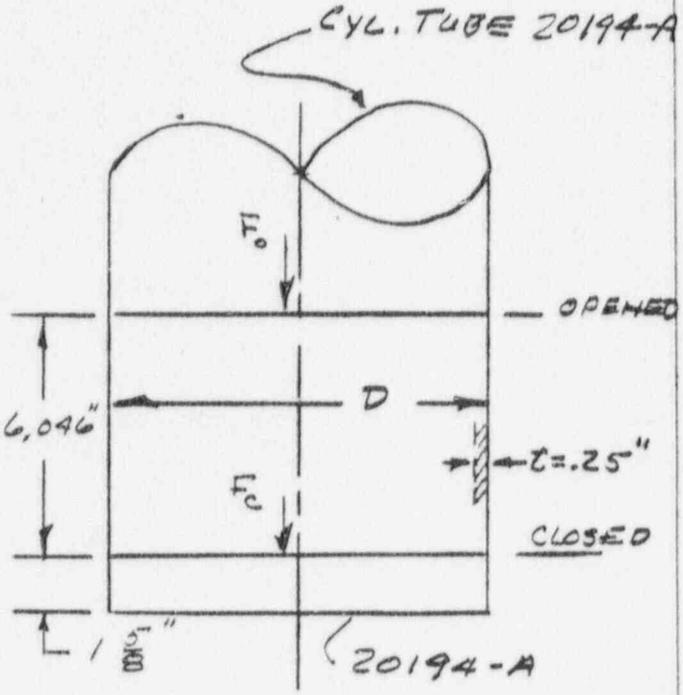
$$\delta = l + \frac{R}{\sqrt{2}} - \sqrt{l^2 - R^2\left(1-\frac{1}{\sqrt{2}}\right)^2}$$

$$\delta = 13.951 - 7.9048$$

$$\underline{\delta = 6.046''}$$

ANALYSIS OF AIR CYLINDER JUST PRIOR TO DISC SLAMMING.

WE ASSUME THAT THE COMPRESSED VOLUME ($1\frac{5}{8}$" GIVEN BY AEM) IS SEALED OFF, AND THAT THE AIR DOES NOT HAVE TIME TO PRESSURIZE THE 3/4" COPPER TUBING. THIS IS CONSERVATIVE WITH REGARD TO THE CYLINDER TUBE. DEPENDING ON THE LGTH. OF 3/4" TUBING ATTACHED TO THE LOWER PLATE THE PRESSURE SHOULD BE LESS THAN 50% OF THE CALCULATED PRESSURE IN THE CYLINDER TUBE.



VOLUME: $V_0 = A \cdot (6.046 + 1.625)$

$V_0 = A \cdot (7.671)$

$V_f = A\,(1.625)$

$\dfrac{V_0}{V_f} = 4.7206$

$P_f = P_0 \left(\dfrac{V_0}{V_f}\right)^{1.4}$; $P_0 = 14.7 + 80 = 94.7$ psia

$P_f = (94.7)(4.7206)^{1.4}$

$P_f = 831.7$ psia

$\Delta p = P_f - 14.7$

$\Delta p = 816.96$ psi (COMPRESSION OF CYL.)

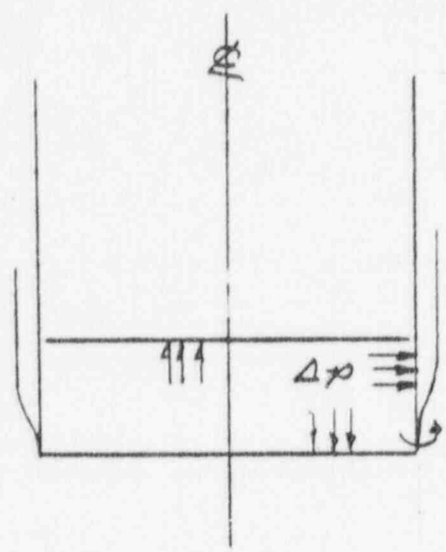
$M_0 = \dfrac{\Delta p}{2\beta^2}$; $\beta^4 = \dfrac{3(1-\nu^2)}{r^2 t^2}$

$r = 3.0''$
$t = .25''$

$M_0 = 185.4$ in-#

$\sigma = \dfrac{6M_0}{t^2} = \underline{17800 \text{ psi}}$

$\sigma_{ALL} = 30,000$ psi



BUCKLING OF PISTON ROD

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

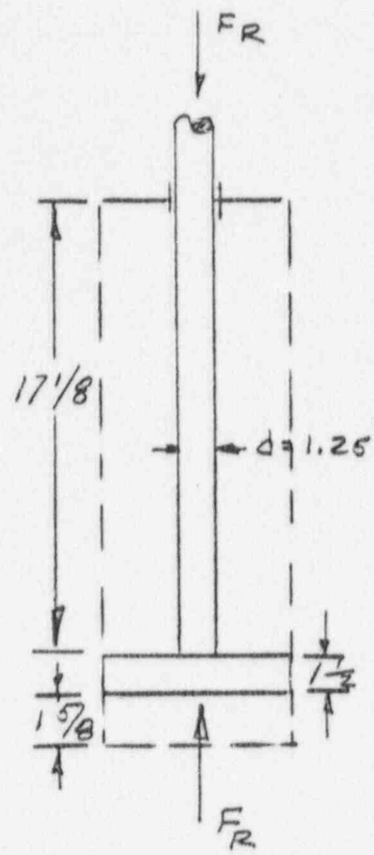
$E = 29 \times 10^6$ psi

$$I = \frac{\pi d^4}{64} = .11984 \text{ in}^4$$

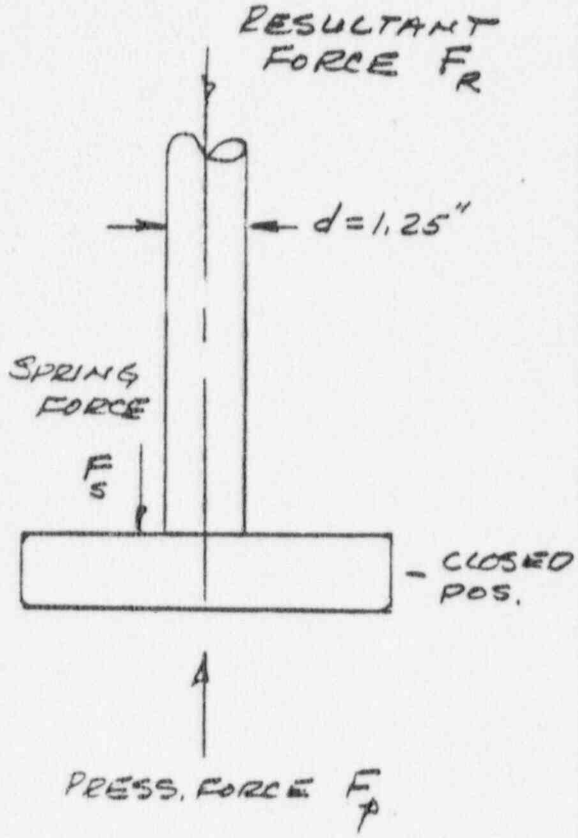
$$P_{CR} = \frac{\pi^2 (29 \times 10^6)(.11984)}{(17.125)^2}$$

$P_{CR} = 116,960$ #

$P_{ACT.} = 23078$ # (SEE pg. 31)



PISTON ROD (14608-B)



SPRING FORCE IN CLOSED POSITION

$F_s = 440^{\#}$

PRESS. FORCE

$F_p = \pi r^2 P_f$; $P_f = 831.7$ psia

$F_p = 23516^{\#}$

so

$F_R = F_p - F_s$

$F_R = 23078^{\#}$

AXIAL STRESS

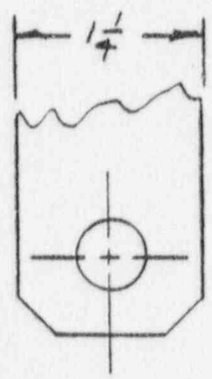
$\sigma = \frac{F_R}{A} = \frac{F_R}{1.23}$

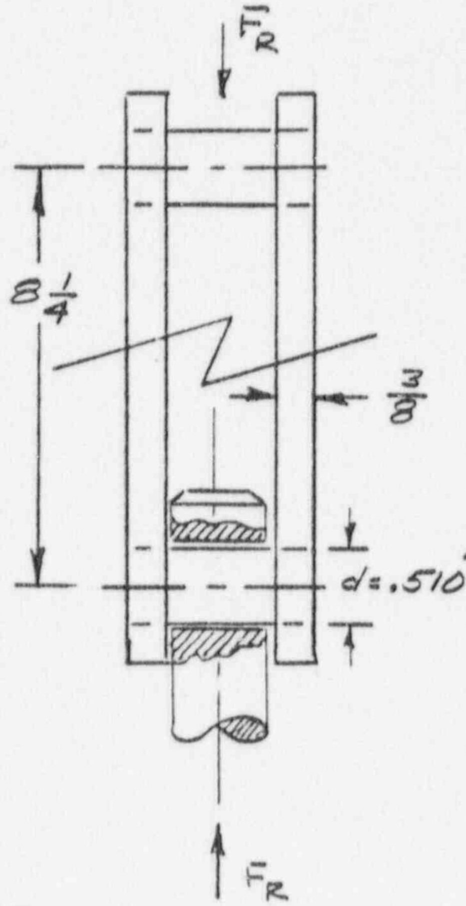
$\underline{\sigma = 18763 \text{ psi}}$

$\sigma_{ALL} = 30,000$ psi

LINK (20328-B)

MAT'L; STL, COM'L





$F_R = 23078^{\#}$

STRESS IN SINGLE MEMBER

$$\sigma = \frac{F_R}{2 \cdot A} = \frac{(23078)}{(2)(1.25)(.375)}$$

$\underline{\sigma = 24616 \text{ psi}}$ ($\sigma_{ALL} = 30{,}000$ psi)

BUCKLING OF MEMBER

$$P_{CR} = \pi^2 \frac{EI}{l^2} \quad ; \quad E = 30 \times 10^6 \text{ psi}$$

$$I = \frac{(1.25)(.375)^3}{(12)} = .005493 \text{ in}^4$$

$l = 8.25"$

$P_{CR} = 23{,}896^{\#}$

$P = 11{,}539^{\#}$

john henry associates inc.

BEARING LOAD ALLOWABLE OF LINK AT PIN

$P_{bry} = 1.85\, \sigma_y \cdot A_{br}$; $\sigma_y = 30,000$ psi (COMPR.)

$A_{br} = D \cdot t = (.53125)(.375) = .1992\ in^2$

So $P_{bry} = 11056^{\#}$

ACTUAL LOAD IS $\frac{F_R}{2} = 11539^{\#}$

$\therefore$ SLIGHT CRUSHING BUT NO CATASTROPHIC FAILURE IS ANTICIPATED SINCE THIS LOAD IS NOT IN THE DIRECTION TO "PULL OUT" THE LINKAGE LUG.

PIN-SHEAR

$\tau_u = (.57)\sigma_u = 31350$ psi ($\sigma_u = 55,000$ psi)

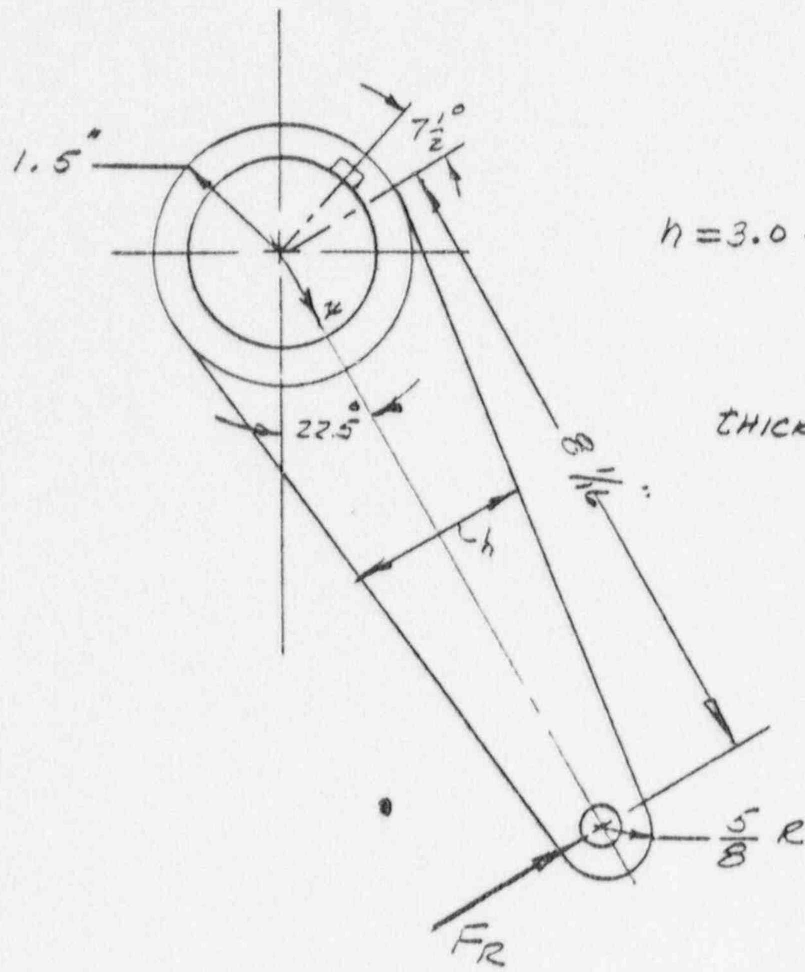
$P_s = \tau_u \cdot 2A$; $A = \frac{\pi d^2}{4} = .196\ in^2$ (PIN)

$\underline{P_s = 12289^{\#}}$

$F_R = 11539^{\#}$

LEVER (20332-B)

MAT'L: STL, COML.



$h = 3.0 - \frac{(1.75)}{(8.0625)} x$

THICKNESS $t = 1.0"$

AT $x = 1.5"$

$h = 2.67"$

$I = \frac{t h^3}{12} = 1.59 \text{ in}^4$

MOMENT ARM $r = 6.5625"$

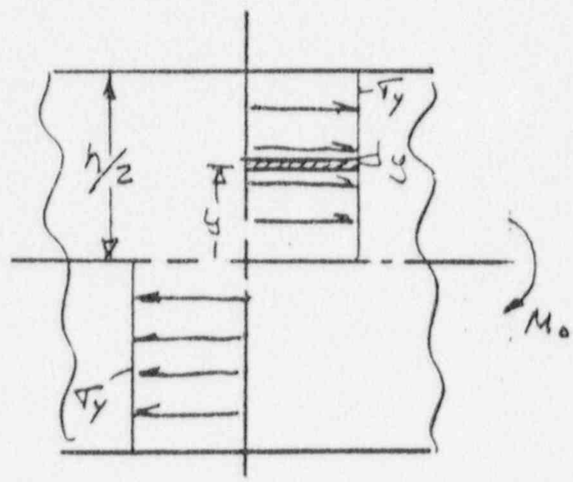
LIMIT MOMENT IN LEVER

$\sigma_y = 30,000$ psi

$$M_o = 2\int_0^{h/2} y\,\sigma_y\,dy$$

$$M_o = 2\,\sigma_y \frac{y^2}{2}\Big|_0^{h/2}$$

$$M_o = \sigma_y \frac{h^2}{4}$$



$M_o = 53467$ in-#

$F_{12} = 8147.3$ #

SINCE THE LEVER ARM IS THE LIMITING MEMBER IN THE LINKAGE THE AIR CYLINDER HAS TO BE LENGTHENED TO REDUCE THE FORCES ACTING ON THE LEVER.

$F_R = F_p - F_s$ (CLOSED POSITION)

$8147.3 = F_p - 440$

$F_p = 8587.3$ #

$F_p = \pi r^2 p_f \quad ; r = 3.0''$

$p_f = 303.7 \text{ psi}$

$p_f = p_o (R)^{1.4} \quad ; p_o = 94.7 \text{ psia} \quad ; R = \frac{V_o}{V_f}$

$R^{1.4} = \frac{p_f}{p_o} = 3.207$

$R = (3.207)^{\frac{1}{1.4}}$

$R = 2.299$

$R = \frac{6.046 + \ell}{\ell} = 2.299$

$6.046 + \ell = R\ell$

$\ell(R-1) = 6.046$

$\ell(1.299) = 6.046$

$\ell \cong 4.65''$

EXTENDED LENGTH OF AIR CYL. TUBE IS

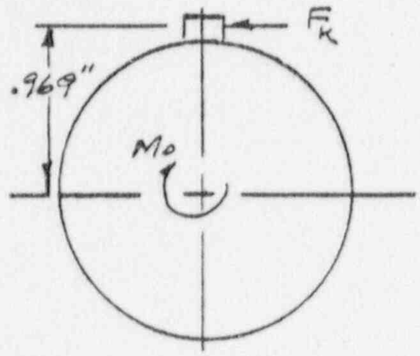
$\Delta L = 4.65 - 1.625$

$\underline{\Delta L \cong 3.0''}$

THIS WILL REDUCE THE LOADS ON ALL OTHER MEMBERS OF THE AIR CYLINDER SYSTEM.

KEY-WAY ON LEVER ARM.

$M_o = 53467$ in #



$$F_K = \frac{M_o}{.969} = 55192 \text{ in-\#}$$

SHEAR AREA

$$A = (.375)(2.0) = .75 \text{ in}^2$$

$$\tau_s = \frac{F_K}{A} = 73600 \text{ psi}$$

$$E_s = \frac{E}{2(1+\nu)} = 11.5 \times 10^6 \text{ psi}$$

$$\gamma = \frac{73600}{11.5 \times 10^6} = .64\% \text{ SHEAR STRAIN}$$

SOME YIELDING TAKES PLACE HOWEVER IT IS TOLERABLE.

APPENDIX A

ARNOLD GREENE TESTING LABORATORIES, INC.
EAST NATICK INDUSTRIAL PARK, 6 HURON DRIVE • NATICK, MASS. 01760
AREA CODE 617 • PHONE: 235-7330 • 653-5950

BRANCH LABORATORIES:
SANTURCE, PUERTO RICO TEL. (809) 723-1833
387 PAGE BOULEVARD SPRINGFIELD, MASS. 01104 TEL. (413) 734-6548
98 PARIS STREET EVERETT, MASS. 02149 TEL. (617) 387-3770
2 MILLBURY STREET AUBURN, MASS. 01501 TEL. (617) 832-8000
100 PIONEER AVE WARWICK, RHODE ISLAND 02888 TEL. (401) 467-2444

TEST REPORT

TO: John Henry Associates Inc
126 New Boston Park
Woburn, MA 01801
ATT: John Henry

DATE: 3 Mar 80
JOB NO.: B 26489
LAB. NO.:
ORDER NO.:

MATERIAL:
HEAT NO.:
SPECIFICATIONS:

Compression Tests

Specimen #1		Specimen #1		Specimen #2	
LOAD	DEFLECTION	LOAD	DEFLECTION	LOAD	DEFLECTION
1,320	.014	15,000	.064	3,000	.032
2,000	.017	17,000	.070	6,000	.043
2,500	.019	18,000	.074	9,000	.054
3,000	.021	19,000	.077	12,000	.064
3,500	.022	20,000	.080	15,000	.074
4,000	.024	21,000	.083	20,000	.088
5,000	.027	22,000	.086	25,000	.099
6,000	.030	25,000	.092	35,000	.117
7,000	.033	30,000	.102	45,000	.130
9,000	.042	35,000	.110	55,000	.142
11,000	.049	45,000	.122		
13,000	.058	55,000	.135		

Hardness: SPEC No
1) 85 shore
2) 85 shore
Mat) 60 shore

R L Tessier

MPR ASSOCIATES, INC.

May 2, 1980
P-522

Mr. G. C. Withrow
Consumers Power Company
1945 W. Parnall Road
Jackson, Michigan 49201

Subject: Big Rock Point Nuclear Plant - Closure Forces for Containment Ventilation System Check Valve

References: (1) A&M Report 201-13938-00, "Evaluation of Palisades Nuclear Plant Main Steam Isolation Valve", August 12, 1976

(2) TR-3223-1 Rev 2, "Palisades Nuclear Plant, Stress Calculations for Shaft for 30" MSIV", Teledyne Materials Research, April 13, 1979

(3) JHA-79-131 Rev 2, "Shaft Analysis for Faulted Condition of 30" Main Steam Stop Valve", John Henry Associates, June 6, 1979

Dear Mr. Withrow:

At your request, we have reviewed the stress analysis of the A&M 24" check valve used in the Big Rock Point containment ventilation system described in the John Henry Associates Report No. JHA-79-138 dated April 14, 1980.

Our review of the subject report indicates that the report does not address a number of concerns which should be considered in an evaluation of structural adequacy of the check valve under slam conditions. For your convenience we have listed in Table 1 the parts and loadings which should be evaluated. Those items marked with an asterisk are not covered in the subject report.

It should be noted that A&M reports of analysis of the Palisades Plant main steam isolation valves which are similar to the Big Rock Point check valves cover the areas of concern not covered by the John Henry Associates report (See references 1, 2, and 3 above). We consider that these earlier reports can provide a useful check list to the areas which should be addressed in the analysis of the Big Rock Point containment ventilation check valves.

Another deficiency in the subject report is the lack of acceptance criteria which will assure operability of the valve after the slam occurs. The report appears to be primarily directed toward demonstrating structural integrity of the valve parts under slam conditions. We recommend that A&M be requested to specify the degree of permanent distortion of the various parts of the valve and operator which can be permitted without interfering with the ability to open and close the valve with the operator and for the valve to maintain leak tight conditions.

In addition to the omissions noted in Table 1, and the concerns discussed above, specific comments on the analyses in the subject report are contained in Enclosure 1. We recommend that A&M be requested to revise the purge valve analyses to resolve the comments summarized in Enclosure 1 and the concerns discussed above.

In this regard, the valve transient conditions needed to perform these analyses will be included in the final draft of MPR Report 644, which should be provided to A&M when it is available.

Please do not hesitate to contact us if there are any questions concerning this letter or Enclosure 1.

Sincerely,

RMWeiner

R. M. Weiner

Detailed Comments on John Henry Report JHA-79-138

1. Page 10 - Disc strain due to impact on the seat

   The disc strain calculation apparently assumes the disc is built in at the disc post. Actually, the post is threaded into the disc and seal welded. It appears the "built-in" assumption is not realistic. Furthermore, the strain is calculated in the disc, but no evaluation is made of the weld which is at the point of highest strain. The plastic strain calculation is made by assuming that the elastic-plastic strain energy absorbed at the point of highest strain is the same as the strain energy calculated by the elastic analysis. This assumption is not valid since it does not account for the redistribution of strain energy when a part of the disc is plastic. The result is an unconservatively low strain. The loading model used for calculating disc strain ignores the concentrated load from the disc post and disc arm.

2. Page 11 - Disc energy

   It is not clear that the cosine shape assumed for the disc distortion is valid. It is not consistent with the formula used for disc strain. The assumed deflection and the strain energy formula are based on elastic analyses of the disc. Since the disc goes plastic, the energy calculation is not correct.

3. Page 19 - Disc velocity

   The permitted disc velocity prior to impact is calculated from the energy absorbed by the seat and disc. This calculation takes into consideration the translational kinetic energy of the disc, but does not account for the rotational kinetic energy of the disc or any of the energy of the disc arm.

4. Page 24 - Disc arm strain

   The elastic strain is calculated. Plastic strain and distortion should be calculated. The deceleration of the end of the arm at the disc center could be used instead of assuming a rigid impact.

5. Page 26 - Shaft stresses

The shaft torsional moment is incorrect. The moment used is the moment at the edge of the hub on the lever arm; the moment about the shaft centerline should be used. This same comment applies to the shaft key analysis on page 37.

The shaft stresses from the operator forces should include the transverse shear and bending stresses.

The shaft strain is calculated elastically; the plastic strain should be calculated.

The strain concentration at the shaft keys should be evaluated.

6. Pages 27-37 - Operator stresses

The operator analyses should include an evaluation of shear and tension stresses at all the connections. In addition, the additional compression of the air due to overshoot of the operator parts when the disc slams closed should be included in the analysis.

7. Page 33 - Operator pin shear

The value used for the force on the pin is one-half the correct value.

TABLE 1

Valve Parts and Loading Conditions

Prior to impact:

Parts	Loading
Disc post* and arm*	centrifugal force*, angular acceleration*, pressure on disc*
Operator	acceleration of operator parts*, cylinder pressure, spring force
Shaft*	combined forces from disc arm* and operator

After Impact:

Parts	Loading
Disc	deceleration of disc and arm*, pressure on disc*
Arm	deceleration of arm
Operator	deceleration of operator parts*, cylinder pressure, spring force
Shaft and keys	combined* forces from disc arm and operator*

*Parts and loadings not included in the A&M report.