Consumers Power Company

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November 9, 1976

Director of Nuclear Reactor Regulation
Att: Mr Albert Schwencer, Chief
Operating Reactor Branch No 1
US Nuclear Regulatory Commission
Washington, DC 20555

DOCKET 50-255 - LICENSE DPR-20 -
PALISADES PLANT - MAIN STEAM
ISOLATION VALVES

By letter dated July 20, 1976 we provided additional information regarding the design of the Palisades main steam isolation valves. An attachment to that submittal discussed a formal report being prepared by Atwood & Morrill Co.

Discussion with members of your staff indicated their desire to review this report when it was completed. This report has now been completed and is submitted as an enclosure to this letter.

David A. Bixel

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Assistant Nuclear Licensing Administrator

CC: JGKeppler, USNRC

ATWOOD & MORRILL CO., INC. SALEM, MASS. 01970	Procedure No. 201-13938-00
MANUFACTURING PROCEDURE OR REPORT	Date August 12, 1976

TITLE Evaluation of Palisades Nuclear Plant
Main Steam Isolation Valve

For Consumers Power Company
Order No. 72575

Approvals | Revisions

Original			(1)	(2)	(3)	(4)
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REVISION SHEET		
Rev. No.	Date	Revision
1	Oct. 4, 1976	Added the evaluation portion of the Disc Arm, Disc Post, and Shaft.

TABLE OF CONTENTS

APPENDIX

1.0 INTRODUCTION

At the request of Consumers Power Company, Atwood and Morrill Company (A&M) has performed a structural analysis and a strain energy analysis to determine the adequacy of a main strain isolation valve for the Palisades Nuclear Plant. The analysis was performed to qualify the disc of these swing-disc type valves for impact resulting from a pipe rupture. The analysis for this faulted condition consists of two parts, (1) a fluid dynamic analysis, done by MPR Associates, Inc. [5]* from which an impact velocity of the slamming disc is determined and (2) a structural evaluation which involves determining the maximum equivalent strain and compares it to established criteria [1]. Meeting these criteria, which have been accepted by the U.S. Nuclear Regulatory Commission, ensures that fracture will not occur.

The valves analyzed herein are already installed and operating, but their discs are to be modified so that they will be similar to discs used in MSIV's at the Farley Nuclear Station. These discs were qualified by Teledyne Materials Research (TMR) in a previous report [2]. This disc is a flat disc, as the Palisades discs are now, but it is constructed of 304 stainless steel rather than carbon steel. Also, the rim is thicker, there is a radius on the back side which avoids high strain concentration at this point, and the hole at the center has been eliminated by redesigning the arm which is now much stronger. This new arm allows deformation of the disc but resists centrifugal and inertial loads during valve closure.

TMR analyzed the Farley disc with the aid of PISCES [3], an elastic-plastic finite difference computer code capable of solving impact problems. Equivalent strains were determined with this code. TMR also reported a method [4] to determine the equivalent strains in other flat discs from the results of the Farley analysis. This report, entitled "Further Interpretation of Farley Isolation Valve Closure Analysis", gives a graphical procedure for finding equivalent strains by comparing the total kinetic energy of the Farley disc to that of the disc under investigation. Hence, a new computer analysis of a similar valve may be avoided. The redesigned Palisades valves are similar but somewhat smaller.

2.0 SUMMARY OF RESULTS

1. The design closing transient conditions were calculated by MPR Associates, Inc. That calculation used a computer model of the fluid dynamic and valve mechanics to obtain the transient pressures, flow, accelerations, and velocities resulting from the postulated main steam line break. The predicted closing velocity of the disc centerline at the time of impact of the seat will be taken as 120 feet per second. This value is used in the structural evaluation of the disc and seat upon impact.

* Numbers in brackets correspond to references, Section 6.

2. Structural Analysis gives the following results;

   A. Equivalent translational velocity of the disc

      $V_{EQ} = 145.5$ ft/sec

   B. Kinetic energy at impact

      KE = 179,183 ft-lb

   C. Energy density

      e = 93.0 ft-lb/in^3

   D. Estimated equivalent strain

      1. $\bar{\varepsilon}$ =(Rim) = 16.8%

      2. $\bar{\varepsilon}$ = (Center Section) = 10.2%

For item 2A, the equivalent translation velocity of the disc is higher than the actual centerline velocity because the kinetic energy of the massive arm has been lumped with the disc. This technique is used because it is conservatively assumed that all of the kinetic energy is dissipated by strain energy absorption of the disc alone. This is a reasonable assumption since the arm is so rigid.

To perform a structural evaluation, the energy of the Farley system must be compared to Palisades. The Farley energy is

$$KE = 225{,}700 \text{ ft-lb}$$

The discs must absorb the kinetic energy by converting it to strain energy. Since they have slightly different volumes, the kinetic energies will be normalized by the disc volume to give the energy density. The energy densities are

1. For Farley

$$e = 98.82 \text{ ft-lb/in}^3$$

2. For Palisades

$$e = 93.0 \text{ ft-lb/in}^3$$

The energy density of Palisades is 94% of the energy density of Farley. Therefore, an evaluation of the Palisades can be performed by the methods of Reference [4] The equivalent strains are

1. For the rim region

$$\bar{\varepsilon} = 16.8\% < 30\%$$

2. $\bar{\varepsilon} = 10.2\% < 18\%$

The allowable equivalent strains for the rim and the center section are 30% and 18%, respectively. Therefore, the disc will retain its integrity for faulted condition.
The Disc Arm, Disc Post, and Shaft integrity is shown in the Appendix to ensure proper Disc seating.

3.0 FLUID DYNAMIC ANALYSIS

The function of the main steam isolation valve in the event of a postulated steam line break is to close promptly and prevent the blowdown of its associated steam generator. At the time of this event the valve would undergo a servere transient due to impact of the valve disc onto the valve seat.

MPR Associates, Inc. determined the design transient conditions in ref [5] The report presents the analysis to determine the angular velocity attained by the disc at impact as a result of the postulated accident.

The method of analysis used a control volume approach to solve the mass, energy, and momentum equations throughout the appropriate region of the main steam piping. The valve internals are modeled to allow a good representation of the torque on the disc using the solution from the mass, energy, and momentum conservation equations. Effects on the geometry as a result of disc position changes are considered in the analysis.

The valve will be shown in the structural evaluation to withstand a predicted disc centerline velocity at the time of impact on the seat of 120 feet per second.

4.0 STRUCTURAL EVALUATION

Structural analysis and evaluation for the Palisades MSIV discs are shown in the Appendix and summarized here.

For the faulted condition, discs in MSIV's close at extremely high velocities and kinetic energy. Large plastic deformation will occur; hence, stress comparisons do not yield meaningful results. On the other hand, strain can be evaluated to determine if fracture will occur anywhere in the disc.

A relationship between the stress-strain diagram and the initial kinetic energy enables the analyst to determine the maximum value of strain. Simply stated, the total kinetic energy at the instant of impact equals the total strain energy of the disc when it comes to a rest. And since strain energy is the area under the stress-strain diagram, the strain is found by monitoring the kinetic and strain energies. The stress and strain due to the applied differential pressure across the disc are negligible.

The PISCES program was used to obtain accumulated strain components and equivalent strain in the Farley disc. The final equivalent strain was then used to determine if fracture would occur by applying the accepted criteria[1]. It was found that the Farley disc was well within the limits of the criteria.

Palisades have valve internals which are similar to the Farley MSIV. A comparison of the disc geometric parameters is given in Table 1. Because of the similarities, it is possible to find the total equivalent strain in Palisades from the Farley valve by comparing the kinetic energies. This procedure is presented in Reference[4]where it is stated that the total energy in a valve to be evaluated should be less than or equal to the maximum kinetic energy of the Farley valve.

A comparison of the impact parameters is given in Table 2. The disc centerline velocities are the actual velocities at impact as found from the fluid dynamics analysis. The Farley valve closes with a greater impact velocity than the Palisades.

It is assumed that all of the kinetic energy is absorbed by plastic straining of the disc. This assumption was made for Farley as well as the present work. This results in the "equivalent translational translational velocity", viz.,

$$\frac{M_D V_{EQ}^{\;2}}{2} = KE$$

where KE is the maximum kinetic energy of the rotating system as found by the fluid dynamics analysis, M_D is the mass of the disc, and V_{EQ} is the equivalent velocity to be determined. For this analysis the kinetic energy of the Farley valve was computed from the above equation using the equivalent velocity from Reference 2.

The Palisades valves are slightly smaller than the Farley MSIV. The discs for Palisades have somewhat less volume, about 16%. This means that the strain energy is more concentrated in the Palisades MSIV discs; there is less material to absorb the kinetic energy. Therefore, the kinetic energies were normalized with respect to disc volume for the comparison to give the energy densities.

As seen in Table 2, the energy density of Palisades is 94% of that of Farley. Figure 1 of the report now enables us to find the equivalent strain for Palisades from the Farley results. The equivalent strains, in the rim region and in the center region, are plotted as a function of accumulated strain energy for the Farley valve. Hence, the equivalent strains for other valves are found from the plot by

locating the impact energy on the abscissa and finding the corresponding equivalent strains. Because of the difference in disc volumes, A&M compared energy densities rather than energy. For the rim region, the equivalent strain from figure 1 is

$$\bar{\varepsilon} = 16.8\%$$

For the center section, the equivalent strain is

$$\bar{\varepsilon} = 10.2\%$$

The allowable equivalent strains from the criteria of Reference [1] are 30% and 18% for the rim and center sections, respectively. Therefore, the discs will not fail by fracture because of the circumferential pipe break.

The Palisades disc arm integrity is shown in the Appendix by comparison with Farley [Ref. 6]. It is shown that the disc arm will not distort and prevent proper seating.

The disc and disc arm are designed in the same manner as Farley which allows the disc post to move independent of the disc arm at impact.

The shaft is the same material and size as Farley and is shown to maintain its structural integrity.

5.0 DISCUSSION AND CONCLUSION

By comparing the strain energy densities of the Palisades MSIV's, we have found that the disc will not fracture as a result of the circumferential pipe break. Since the Farley valve qualified for the faulted condition, the Palisades valves would qualify simply because their energy density is lower. The body also will qualify for Palisades by virtue of the fact that the impact energy is less than that of Farley. The maximum effective strain in the Farley valve body is 15%. Because it is accompanied by a compressive hydrostatic stress state, an effective strain of up to 29% is acceptable for the carbon steel body.

CONCLUSION

It has been found that the Palisades MSIV's will retain their structural integrity during and after a circumferential break in the main steam line.

TABLE 1

Comparison of Geometric Properties

	Farley	Palisades
Disc radius, r	13.75 in	12.625
Disc thickness, t	3.846 in	3.846
r/t ratio	3.575 in	3.283
Disc volume, V	2284 in^3	1926
Disc weight, W	646 lb_m	545
System moment of inertia about shaft centerline, I	255,200 $lb\text{-}in^2$	192,600
Pressure area, A	594 in^2	501

TABLE 2

Comparison of Impact Parameters

	Farley	Palisades
Centerline velocity at impact, V_{CL}	139 ft/sec	120.0
Rotational velocity, w	104.3 rdn/sec	92.89
Equivalent translational velocity, V_{EQ}	150 ft/sec	145.51
Kinetic energy, $KE = W\ V_{EQ}{}^2/2g$	225,700 ft-lb	179,183
Energy density, $e = KE/V$	98.82 $ft\text{-}lb/in^3$	93.0

6.0 REFERENCES

1. T. Slot, O. Batum, and R. A. Genier, "Dynamic Analysis of Steam Isolation Valve for Closure Under Faulted Conditions", Proceedings of Third International Conference on Structural Mechanics in Reactor Technology, London, September, 1975.

2. Joseph M. Farley Nuclear Plant Final Safety Analysis Report Appendix 10A Amendment No. 45 dated 2/21/75.

3. PISCES-2DL, Version 3, Manuals A, B and C, Physics International Company, 2700 Merced, San Leandro, Calif. (1974).

4. Teledyne Materials Research Technical Report TR-2196, "Further Interpretation of Farley Isolation Valve Closure Analysis", November 5, 1975.

5. MPR Associates, Inc. Report MPR 500, "Analysis of Disc Impact Velocity for Palisades Main Steam Isolation Valve as a Result of a Main Steam Line Rupture", November 1975.

6. Teledyne Materials Research Technical Report TR-2000, "Disk Arm Analysis of Main Steam Swing Trip Valve, Farley Nuclear Power Station", February 20, 1975.

APPENDIX

A. Angular Velocity

B. Equivalent Translation Velocity

C. Kinetic Energy

D. Energy Density

E. Estimated Equivalent Strain

F. Mass Moment of Inertia

G. Disc Arm Evaluation

H. Disc Post & Shaft Evaluation

A. Angular Velocity

The angular velocity ω_p of the Palisades disc is,

$$\omega_p = \frac{V_{\mathfrak{C}}}{r}$$

$$= \frac{120}{1.29} = 92.89 \text{ rdn/sec} \qquad \frac{\text{ft/sec}}{\text{ft}} = \text{rdn/sec}$$

Where: $V_{\mathfrak{C}}'$ = centerline velocity = 120 ft/sec

r = distance from ℄ of shaft to the ℄ of the disc, = 1.29 ft

B. Equivalent Translational Velocity

The equivalent translational velocity must be calculated to determine the kinetic energy and energy density.

Using figure 1 and the energy density ratio determine the strain - percentage to conclude that the Palisades disc will meet the established criteria.

Equating rotational kinetic energy to equivalent translational kinetic energy.

$$\frac{I\ \omega^2}{2g} = \frac{M\ V_{eq}^2}{2g}$$

Where: I = System moment of inertia about shaft centerline = 192600 lb-in²

ω = angular velocity = 92.89 rdn/sec

M = disc Weight = 545 lbm

Solving for the equivalent translational velocity (V_{eq})

$$V_{eq} = \sqrt{I\ \frac{\omega^2}{M}}$$

$$V_{eq} = \sqrt{\frac{192600 \times (92.89)^2}{545} \times \frac{1}{144}} \qquad \sqrt{\frac{\cancel{lb\text{-}in^2}\ rdn^2/sec^2}{\cancel{lbm}} \frac{ft^2}{\cancel{in^2}}}$$

$$\boxed{V_{eq} = 145.51\ ft/sec}$$

C. Kinetic Energy

$$KE = W V_{EQ}^2 / 2g$$

Where W = weight of the disc

$$= (545)(145.51)^2 / (2)(32.2) \quad 16 \left(ft^2/sec^2\right) / \; ft/sec^2$$

$$\boxed{KE = 179\,183} \; ft\text{-}16$$

D. Energy Density

$$e = KE/V$$

Where e = energy density
V = Volume of Disc

$$e = 179183/1926 \quad ft\text{-}16/in^3$$

$$\boxed{e = 93.0} \; ft\text{-}16/in^3$$

when Volume of Disc:

$$V = \frac{1}{4} \pi D^2 \times t$$

Where D = Diameter
t = thickness

$$V = \frac{1}{4} \pi (25.25)^2 \times (3.846)$$

$$\boxed{V = 1926 \; in^3}$$

E. The Estimated Equivalent Strain

The energy density must first be compared to the energy density of the Farley value.

$$\frac{100}{e_f} = \frac{x}{e_p}$$

Where e_f = energy density of the farley valve

e_p = energy density of the pallisades,

X = Percent of maximum effective strain

$$\frac{100}{98.2} = \frac{x}{93.0}$$

$$\boxed{x = 94.11\%}$$

Going to figure 1. we find,

$\bar{\epsilon}$ = 16.8 % for the rim

$\bar{\epsilon}$ = 10.2 % for the center section

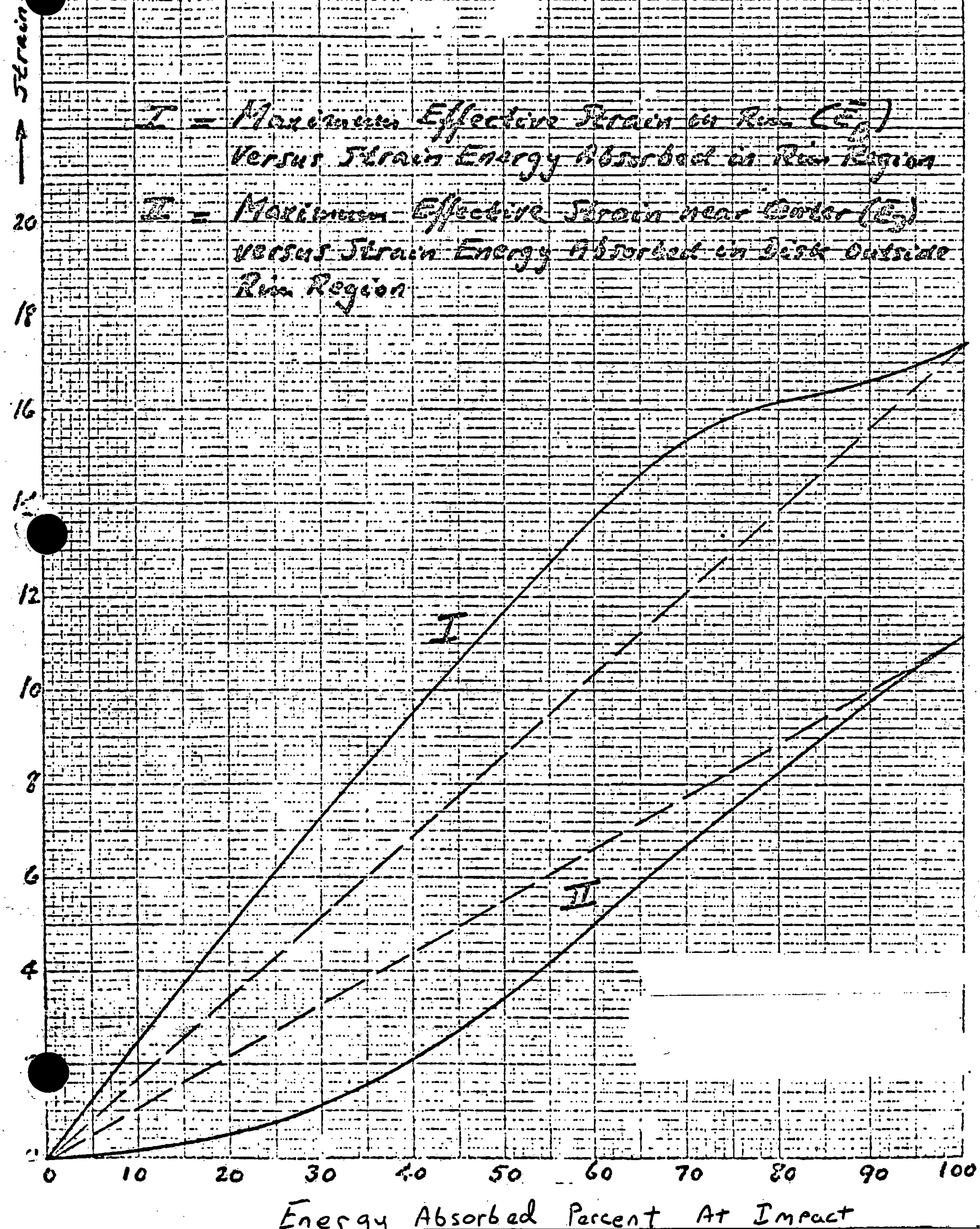
The allowable equivalent strains from the criteria given are 30% and 18% for the rim and center sections, respectively. Therefore the disc will not fail by fracture because of the circumferential pipe Break.

$$\boxed{30\% > 16.8\%}$$

$$\boxed{18\% > 10.2\%}$$

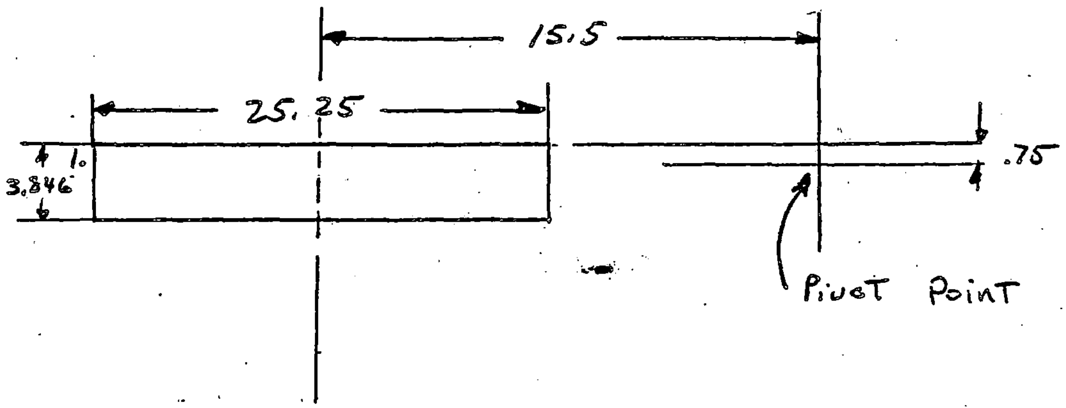
FIGURE 1

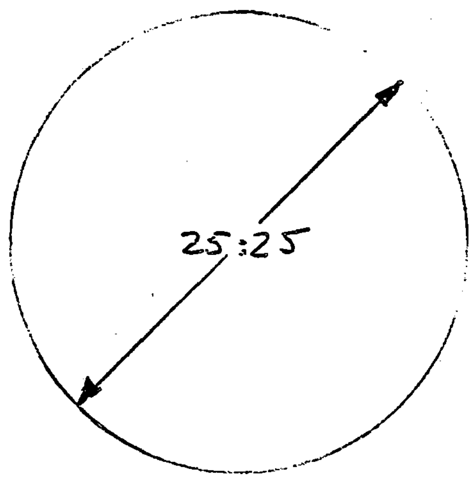
MAXIMUM STRAIN ACCUMULATION AND STRAIN ENERGY ACCUMULATION



F. Moment of Inertia

The moment of inertia of the disc is found here





The mass moment of inertia of the disc about the shaft is given by I

$$I = \bar{\bar{I}} + md^2$$

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1. The additional .096 in above the 3.75 in thickness of the disc is due to added effect of the disc post.

Where

$\bar{I}$ is the mass moment of inertia about an axis parallel to the shaft and passing through the mass center of the disk. This is given by

$$\bar{I} = \frac{m r^2}{4} + \frac{m \ell^2}{12}$$

Where

m = mass of the disc

r = disc radius = 12.625 in

ℓ = disc thickness = 3.846 in

The mass of the disc is given by

$$m = \rho \pi r^2 \ell$$

$$= .283 \frac{lbm}{in^3} \times \pi \times 12.625^2 \, in^2 \times 3.846 \, in$$

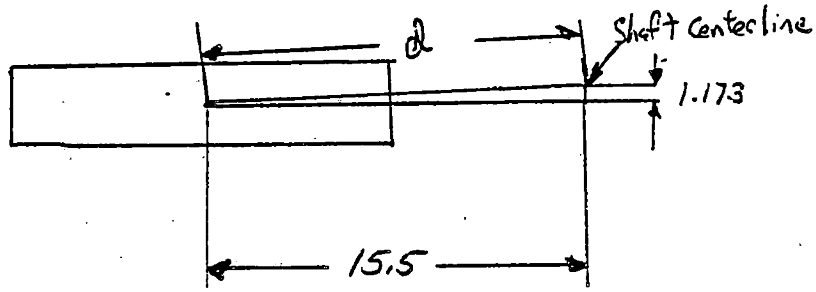
$$\boxed{m = 545 \text{ lbm}}$$

Substituting into $\bar{I}$

$$\bar{I} = \frac{545 \text{ lbm} \times 12.625^2 \, in^2}{4} + \frac{545 \text{ lbm} \times 3.846^2 \, in^2}{12}$$

$$\boxed{\bar{I} = 22389 \text{ lbm} - in^2}$$

Before we solve for I we need d. d is the distance between the shaft axis and the axis parallel to the shaft which passes through the center of mass of the disc. We have the following geometry,



$$d = \sqrt{15.5^2 + 1.173^2}$$

$$\boxed{d = 15.54 \text{ in}}$$

We now have all the parameters necessary to solve for I

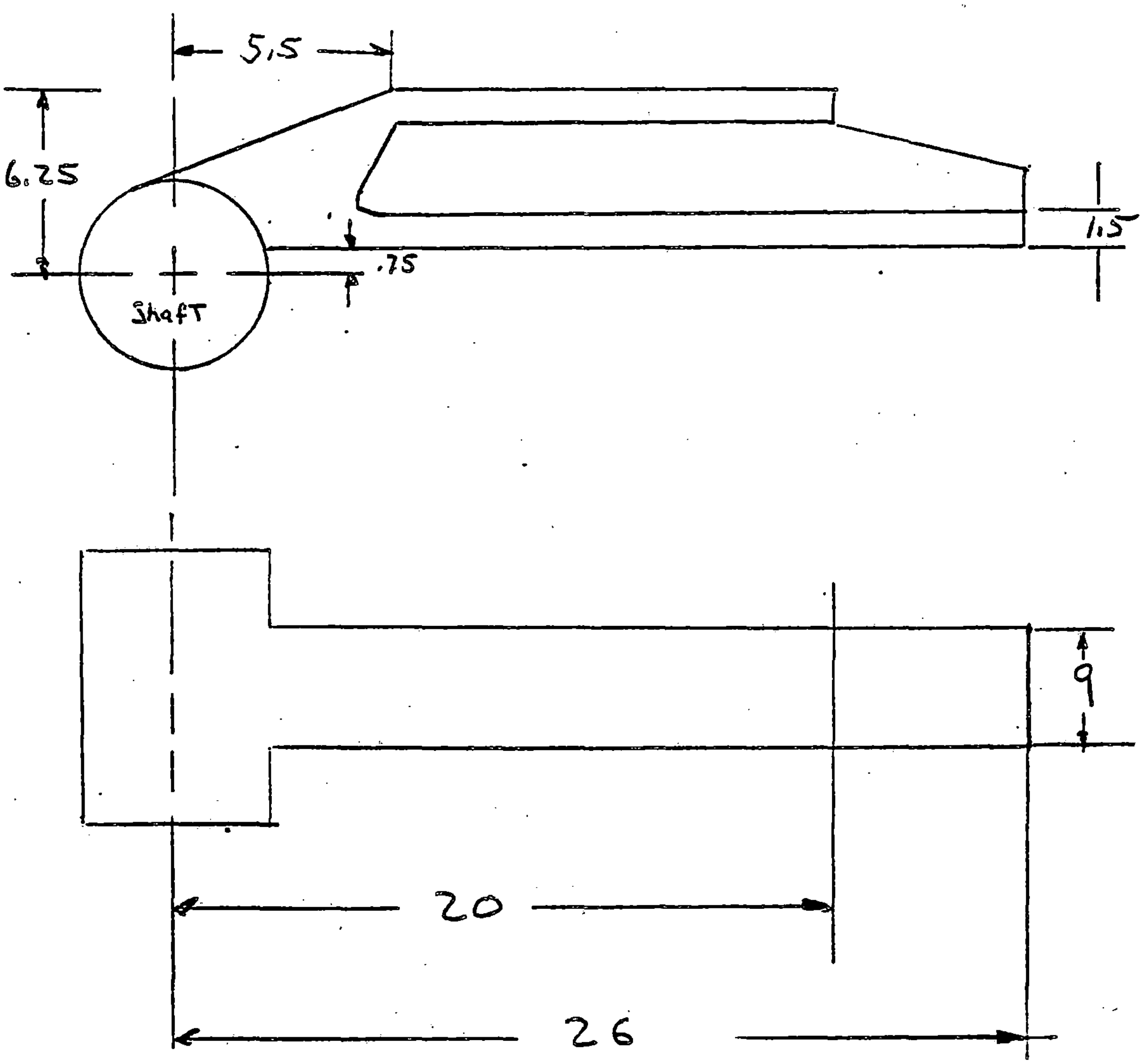
$$I = \bar{I} + md^2$$

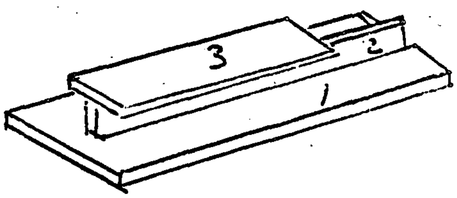
$$= 22389 \text{ lbm-in}^2 + 545 \text{ lbm} \times 15.54^2 \text{ in}^2$$

$$= 22389 \text{ lbm-in}^2 + 131613 \text{ lbm-in}^2$$

$$\boxed{I = 154002 \text{ lbm-in}^2}$$

Mass Moment of Inertia of Disc ARM





Section 1 bottom plate

Section 2 web

Section 3 top plate

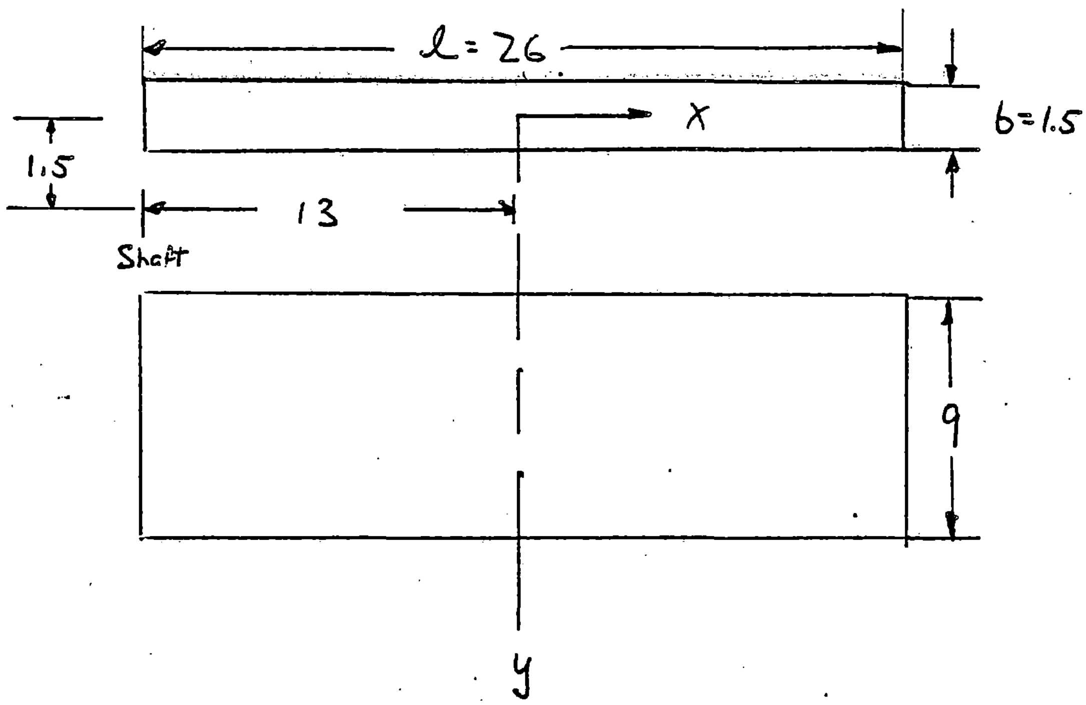
Simplified model

The mass moment of inertia of the lever arm will be found using this simplified model.

The mass moment of inertia of a rectangular parallelepiped about its center of gravity is given by

$$I_{c.G.} = \frac{m}{12}(b^2 + l^2)$$ where m = mass
b = depth
l = length

Mass Moment of Inertia of Section 1



Mass

$m = .283 \frac{lbm}{in^3} \times 9in \times 26in \times 1.5in$

$= \underline{\underline{99.33\ lbm}}$

$I_{c.G.} = \frac{99.33\ lbm}{12}(1.5^2 + 26^2)\ in^2$

$= \underline{\underline{5614\ lbm\text{-}in^2}}$

I_{yy} is the mass moment of inertia about the C.G. of the piece. We want the moment of inertia about the shaft

$$I_{shaft} = I_{yy} + md^2$$

Where

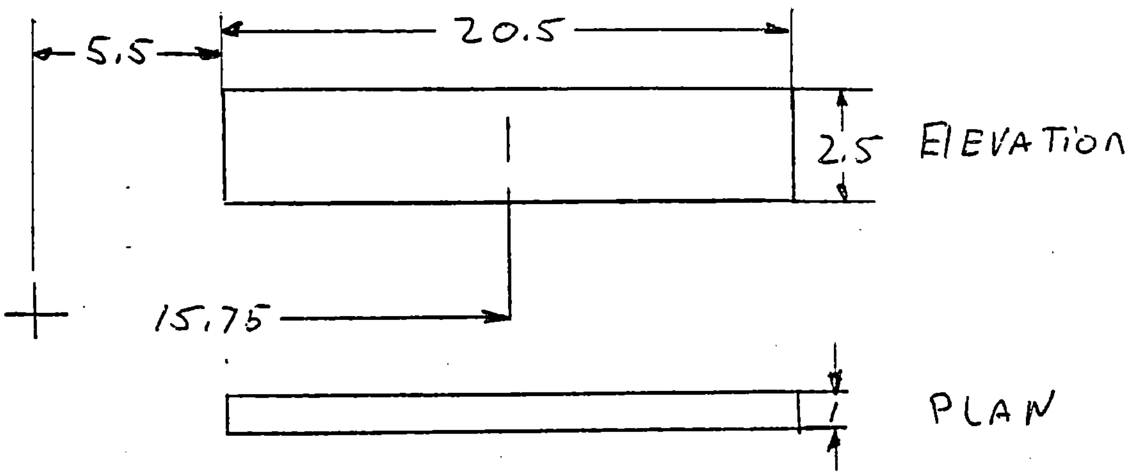
$$d^2 = 1.5^2 + 13^2$$

$$= \underline{\underline{171.25 \ in^2}}$$

$$I_{shaft} = 5614 + 99.33 \times 171.2$$

$$= 22624 \ lbm\text{-}in^2$$

Moment of Inertia of Section 2



Mass $= .283 \ \frac{lbm}{in^3} \times 1\,in \times 2.5\,in \times 20.5\,in$

$= \underline{14.5\ lbm}$

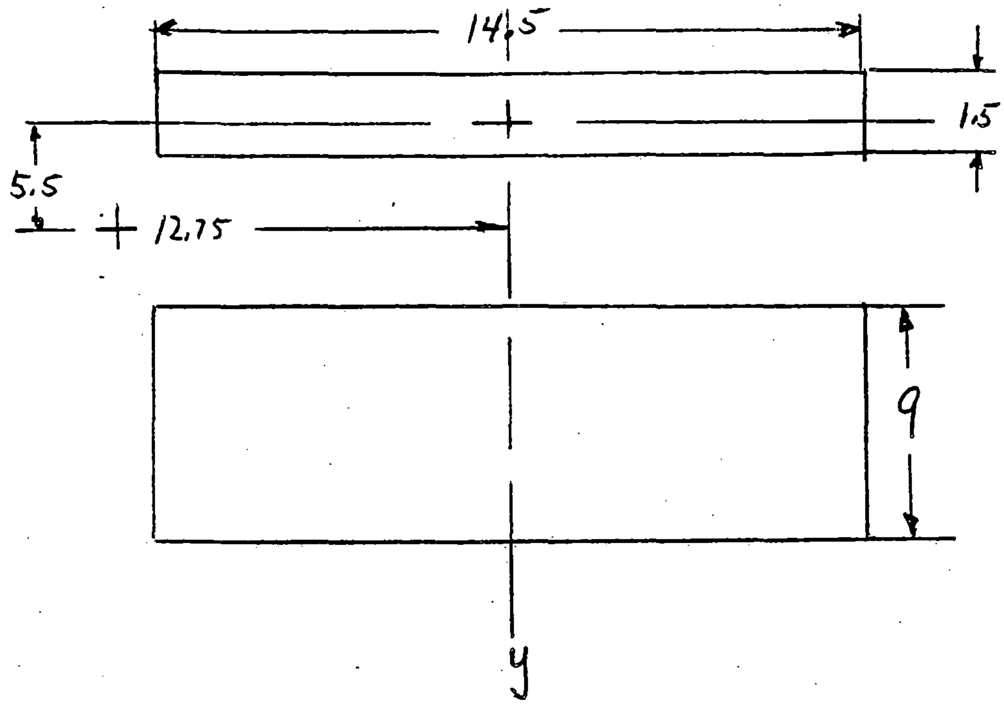
$I_{C.G.} = \frac{14.5}{12}(2.5^2 + 20.5^2)$

$= \underline{515\ lbm\text{-}in^2}$

$I_{2\ shaft} = 515.4 + 14.5(15.75^2 + 3.5^2)$

$= \underline{4289\ lbm\text{-}in^2}$

Mass Moment of Inertia of Section 3



$$Mass = .283 \frac{lbm}{in^3} \times 1.5 in \times 9 in \times 14.5 in$$

$$= \underline{\underline{55.4\ lbm}}$$

Moment of inertia about the centroid

$$I_{c.g.} = \frac{55.4\ lbm}{12}(1.5^2 + 14.5^2)$$

$$= \underline{\underline{980.9\ lbm\text{-}in^2}}$$

Moment of inertia about the shaft

$$I_{3\ shaft} = 980.9 + 55.4\,(12.75^2 + 5.5^2)$$

$$= 11662\ lbm\text{-}in^2$$

Total Mass Moment of Inertia of the disc Lever Arm

$$I = 22624 + 4289 + 11662$$
$$= 38576 \text{ lbm-in}^2$$

Total Mass Moment of Inertia of Disc and Lever Arm

$$I_{TOT} = 38576 + 154002$$
$$= 192578$$

$$\boxed{I_{TOT} = 192600 \text{ lbm-in}^2}$$

Total Mass

$$M_{TOT} = 545 + 99.33 + 14.5 + 55.4$$
$$= 714 \text{ lbm}$$

Appendix 6

Centrifugal and Inertia Loading of Disc Arm

The analysis of disc arm strength will parallel calculations made in Ref. 6 for the Farley main steam swing trip valve. The values for maximum allowable forces and moments for the Farley valve will be used for the Palisades valve disc arm, since the cross section and material of two valves are identical.

This report will calculate the actual forces and moments for comparison to the Farley allowables, using the method described in the Farley report.

1. Loads on the Disc-Arm Assembly

All loads acting on the disc arm are due to the pressure differential across the disc, causing rotary motion of the assembly. The forces applied to the disc arm by the disc are as follows

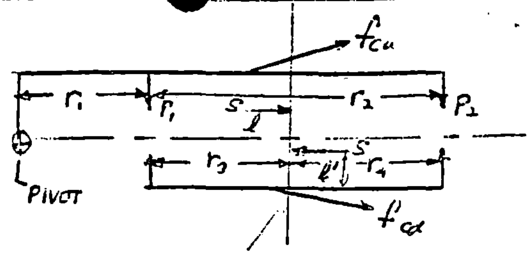
P_1, P_2 = bolt loads

S = bearing load of disc post on arm

In order to calculate P_1 and P_2, it is first necessary to compute the bearing load S:

$$S = m_d r \omega^2 \cos\theta + m_d r \alpha \sin\theta$$

Where



$m_d = 1.410 \ \frac{lb\text{-}sec^2}{in}$

$\omega = 92.89$ rad/sec

$\alpha = 11{,}800$ rad/sec^2

$r = 15.5''$

$\theta = \tan^{-1} \frac{1.173}{15.5} = 4.328°$

Substituting the values above

$$S = (1.410)(15.5)(92.89)^2 \cos 4.328° + (1.410)(15.5)(11{,}800) \sin 4.328$$

$$= 207{,}501$$

P_1 and P_2 can be calculated using the equation

$I_{pa}\alpha = P_1 r_1 + P_2 r_2 + S l$ ARM EQN

$I_{do}\alpha = P_1 r_3 - P_2 r_4 - S l'$ DISC EQN

where

$r_1 = 6.062''$	$r_3 = 9.25''$	$l = .75''$
$r_2 = 23.562''$	$r_4 = 8.25''$	$l' = 1.923''$

So

$$99.834(11{,}800) = P_1(6.062) + P_2(23.562) + (.75)\,207{,}501$$

$$57.943(11{,}800) = P_1(9.25) - P_2(8.25) - (1.923)\,207{,}501$$

This Results in

$P_1 = 126{,}686$ lbs

$P_2 = 10{,}799$ lbs

2. Moment Calculations

a. Moment Distribution due to Rotation

The moment equation for rotating arm pivot due to inertia is

$$M_\alpha = I_{pa}\alpha - \alpha\int_0^x C\eta^2 d\eta$$

where C is the shape factor.

Since the right hand end of the disc arm is free, there is no moment (at $x = 23.562''$, $M_\alpha = 0$)

Using these boundary conditions, the equation can be simplified to:

$$M_\alpha = I_{pa}\alpha - \frac{\alpha C}{3}x^3$$

or

$$= I_{pa}\alpha - a x^3$$

Substituting, $x = 23.562$ — $M_\alpha = 0$

$$0 = (99.834)(11{,}800) - a(23.562)^3$$

$$a = 90.058$$

The inertia moment is, therefore

$$\boxed{M_\alpha = 1{,}178{,}041 - 90.058\,x^3}$$

b. Centrifugal Force Distribution

The centrifugal load at x varies linearly with the distance from the C.G.:

$$F_\omega(x) = f_{ca} - \frac{f_{ca}}{23.562}x \qquad f = m_a r \omega^2$$

$$F_\omega(x) = 51,273 - \frac{51,273}{23.562}x \qquad = .438\sqrt{(3.17)^2+(3.25)^2}\,(92.89)^2$$

$$= 51,273 \text{ lbs}$$

The moment arm $u(x)$ is given by $u = V \sin\alpha$ where $\alpha = \tan^{-1}\frac{3.25}{(23.562-x)/2}$

$$\text{or} \quad u = \frac{23.562-x}{2}\sin\left(\tan^{-1}\frac{6.5}{23.562+x}\right) \qquad V = \frac{23.562+x}{2} - x$$

This results in

$$M_\omega(x) = F_\omega(x)\,u(x)$$

$$= 51,273\left(\frac{23.562-x}{23.562}\right)\left(\frac{23.562-x}{2}\right)\sin\left(\tan^{-1}\frac{6.5}{23.562+x}\right)$$

$$\boxed{M_\omega(x) = 51,273 \quad \frac{(23.562-x)^2}{47.136}\sin\left[\tan^{-1}\left(\frac{6.5}{23.562+x}\right)\right]}$$

c. Total Moment

Adding the various moments applied to the arm,

$$M_T = M_\alpha(x) + M_\omega(x) + 2.55 - P_2(23.562-x)$$

The maximum moment is at the first bolt, at $x = 6.062''$

This yields

$$M_{max}\ 1{,}178{,}041 - 90.058(6.062)^3 +$$
$$51{,}273\left(\frac{(23.562-x)^2}{47.136}\right)\sin\left(\tan^{-1}\frac{6.5}{23.562+6.062}\right)$$
$$+\ 2.5\,(207{,}501) - 10{,}799\,(23.562-6.062)$$

or

$$M_{max} = 1{,}157{,}979 + 71{,}393 + 518{,}753 - 188{,}983$$
$$= 1{,}559{,}143 \text{ in-lbs}$$

3. Axial Load

To check the load conditions of the arm, the axial load and total moment must be compared to the maximum allowable values given in the Farley report these values are

$$F_{max} = 301{,}875 \text{ lbs}$$
$$M_{max} = 1{,}817{,}140 \text{ in-lbs.}$$

The axial force can be calculated from

$$F_{ax} = S + m_a r \omega^2 \cos\phi$$
$$- \frac{I_{pa}\alpha}{r}\sin\phi \qquad \phi = \tan^{-1}\frac{3\,1/4}{13.188}$$
$$= 13.84°$$

$$= 207{,}501 + 51{,}273\cos 13.84°$$
$$- \frac{(99.834)(13{,}656)}{13.188}\sin 13.84°$$

$$= 207{,}501 + 49{,}784 - 24{,}729$$

$$F_{ax} = 232{,}557 \text{ lbs}$$

The "collapse" loads that the arm must carry are 1/.9 times the actual, yielding in this case

$$M_c = 1,559,143 / .9 = 1,732,381 \quad \text{in-lbs}$$

$$F_c = 232,557 / .9 = 258,396 \text{ lbs}$$

Both of these are below the allowable, as calculated by Farley. Since the cross sections of the two disc arms are identical, it can be concluded that the Palisades disc arm is not overstressed.

4. Bolt and Tab Loads

The bolt load as determined earlier is

$P_1 = 126,686$ lbs
$P_2 = 10,799$ lbs
} each carried by two bolts

So

$B_1 = P_1/2 = 63,343$ lbs bolts closest to pivot

$B_2 = P_2/2 = 5400$ lbs bolts furthest from pivot

The moments exerted on the tabs by the bolts is (moment arms are 1.5" for B_1 and 3" for B_2)

$M_1 = 1.5 \times B_1 = 95,015$

$M_2 = 3 \times B_2 = 16,200$ in-lbs

The collapse moments are

$M_{1c} = M_1/.9 = 105,572$ in-lbs

$M_{2c} = M_2/.9 = 18,000$ in-lbs

The moment carrying capacity of the tabs is given by

Tab 1: $M = 2(40,250 \times 4.25 \times .75 \times .75/2)$

(4.25" × 1.5" tab) = 96,200 in-lbs

Tab 2: 3" wide × 1.5" high
$M_2 =$ 67,900 in-lbs

Both tabs are considered acceptable.

5. "T" sections

The reduced sections of the disc arm nearest the pivot and at the opposite end of the arm are analyzed in Ref. 6, for the Farley Valve. Since the Axial Load and Moment as calculated in this report are both less than the Farley values, it can be concluded that the "T" sections of the Palisades arm are of sufficient size to safely carry the imposed loads.

Appendix H

Shaft Load

The load imposed on the valve shaft is due to the angular rotation of the disc-disc arm assembly. The maximum axial load calulated in Appendix G is 232,557 lbs. Since this is lower than the Farley load and since the shafts are identical in the two valves, the Palisades valve can be considered adequate. The stress in the Palisades shaft will be 80% of the stress in the Farley shaft.

Disc Post

The maximum load on the disc post as calculated in Appendix G of this report is S = 207,501 lbs. The disc post load on the Farley valve is S = 232,211 lbs. Since the disc post size is the same on both valves, the stresses in the Palisades post must be less than those in the Farley valve and, therefore, acceptable