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EVALUATION OF 42-INCH CONTAINMENT ISOLATION VALVES FOR ZION STATION, UNITS 1 & 2

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REVISION CONTROL SHEET

SUBJECT: EVALUATION OF 42 INCH CONTAINMENT ISOLATION VALVES FOR ZION STATION, UNITS 1 & 2

REPORT NUMBER: COM-0709-01

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EFFECTIVE PAGE(S)	REV	PREPARED	ACCURACY CHECK	CRITERIA CHECK	EFFECTIVE PAGE(S)	REV	PREPARED	ACCURACY CHECK	CRITERIA CHECK
ii-iv	0	DAG	TL	ABF	ii	1	DAG	T.L.	ABF
					iv	1	DAG	TL	ABF
1.1-1.2	0	DAG	TL	ABF	1.1	1	DAG	TL	ABF
					2.1	1	DAG	TL	ABF
2.1-2.2	0	DAG	TL	ABF	3.2	1	DAG	TL	ABF
3.1-3.10	0	DAG	TL	ABF	3.4	1	DAG	TL	ABF
					3.8	1	DAG	TL	ABF
4.1-4.2	0	DAG	TL	ABF	3.10	1	DAG	TL	ABF
5.1	0	DAG	TL	ABF	6.1	1	DAG	TL	ABF
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*As shown on individual page.

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ABSTRACT

A stress analysis was performed by NUTECH to evaluate stress level margins in critical components of the 42-inch butterfly valves manufactured by the Henry Pratt Company and used as containment purge isolation valves at Zion Nuclear Power Station, Units 1 & 2.

The purpose of this evaluation was to determine the worst case stress level margins existing in the critical load-carrying structural members of the valve during a closing event under design basis Loss of Coolant Accident conditions.

The evaluation consisted of an analysis of stresses in the valve shaft, pins, key and actuator arm. This analysis was performed using, as the loading condition, valve shaft torque values calculated from values in the Henry Pratt Company report, Reference 1. The results of the stress analysis indicate that the worst case stress level margins in the valve load-carrying structural members are acceptable.

It is concluded from this analysis effort that the critical internal components of the Pratt 42-inch butterfly valve will retain structural integrity if subjected to the flow induced loads resulting from a postulated design basis Loss of Coolant Accident when used as a containment purge isolation valve at Zion Station, Units 1 & 2.

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1.0 INTRODUCTION

In References 2, 3 and 4, the Nuclear Regulatory Commission requested Commonwealth Edison Company to respond to generic concerns regarding containment purging during normal plant operation and provided guidelines for operability of containment isolation valves used for purging. These operability guidelines included:

1. Demonstrating that the containment isolation valve actuators have sufficient torque capability to stroke the valves from full open to full closed within the technical specification time limit against design basis Loss of Coolant Accident containment pressure.

2. Ensuring that the valve structural elements have sufficient stress margins to withstand the concomitant loads imposed while closing.

The containment purge isolation valves at Zion Station, Units 1 & 2, are butterfly valves manufactured by the Henry Pratt Company. Based on the hydrodynamic torque results in Reference 1, it is apparent that these butterfly valves tend to close under the postulated flow conditions.

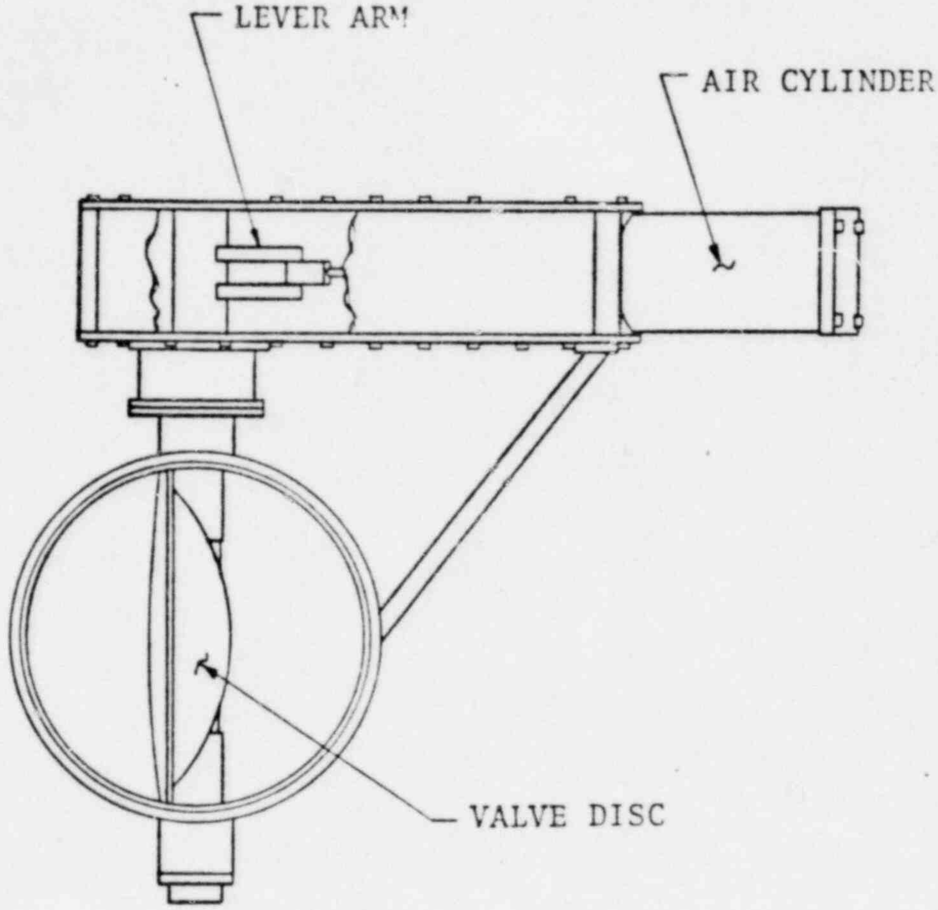
In order to address the second operability guideline, an analysis effort was conducted to evaluate the stress margins inherent in the valves under the postulated flow conditions. Analysis of the critical load-carrying components of the valve was performed utilizing as input the maximum torque values from the Reference 1 Henry Pratt Company report adjusted for postulated containment pressure. The analysis was comprised of a simplified stress analysis considering bending, shear and torsional shear loadings. Stress margins were calculated for the load bearing components utilizing standard stress allowables.

This report, prepared for Commonwealth Edison Company, presents the results of the stress analysis performed on the Zion Station, Units 1 & 2, 42-inch containment isolation valves and provides verification of acceptable stress level margins in their critical internal structural components under a postulated design basis Loss of Coolant Accident. The report summarizes the stress analysis design criteria, loading conditions, methods and results. APPENDIX A is the NUTECH stress and hydrodynamic torque analysis.

2.0 COMPONENT DESCRIPTION

The valves used at Zion Station, Units 1 & 2, for containment purging are 42-inch offset asymmetric disc butterfly valves with external pneumatic/spring (air to open/spring to close) valve actuators (shown on Figure 2-1). These Pratt 42-inch valves are constructed with a carbon steel body, a 4-1/4 inch diameter type 304 stainless steel shaft and sintered bronze bearings. The valve is mounted in horizontal runs of pipe with the shaft vertically oriented. The actuators are aligned horizontally and are attached to the valve shaft through a lever arm which is keyed to the shaft with a cold drawn steel key. The valve disc is attached to the valve shaft with two 1-1/2 inch tapered stainless steel pins.

Each plant uses two of these valves in series in the containment purge line. Of each pair, one valve is located inside containment and the other one is located outside containment. They serve the function of containment isolation valves. In the postulated event that these valves are open for purging and a design basis Loss of Coolant Accident occurs, these valves must be capable of closing within the technical specification time limits and provide containment integrity.



SKETCH OF BUTTERFLY VALVE

FIGURE 2-1

3.0 STRESS ANALYSIS DESIGN CRITERION, LOADING CONDITION AND ANALYTICAL METHODS

The stress analysis of the valve is presented in APPENDIX A. The design criterion, loadings and analytical methods used are presented below.

3.1 Design Criteria

The purpose of the analysis was to analytically determine that the stress levels in the valve load-carrying structural members were within limits that would preclude yielding for those members as the valve is closed against a flow rate generated by a postulated design basis Loss of Coolant Accident. This criterion would ensure that during closing, the active valve parts would not deform.

3.2 Loading Condition

The loading condition considered in the stress analysis of the butterfly valve included hydrodynamic torque and valve actuator restraining force. Dead weight and seismic forces were considered to be negligible.

Torque loading of the valve shaft was based on the torque values provided by the Henry Pratt Company in Reference 1, as reproduced in Table 3-1. These torque values were proportionally adjusted for containment pressure, as described in the following paragraph.

The Reference 1 hydrodynamic torque values were calculated using a containment pressure of 42.7 psia, the maximum postulated containment pressure at the end of the valve closing stroke. To be more realistic, the Zion Station containment pressure-time curve (Reference 5) was used to determine containment pressure versus valve disc angle during the valve stroke (The containment pressure-time characteristics are shown in Figure 3-1.). This method yields the relationship between hydrodynamic torque and valve disc angle represented on Figure 3-2. Figure 3-2 was determined by scaling torque values from the Reference 1 values on the basis of containment pressure and Mach number as follows:

$$\text{Torque} = C_T \rho V^2 D^3 \qquad \text{(Reference 6)}$$

with C_T = Torque coefficient

ρ = Density of air

V = Velocity of air at minimum area in valve

D = Diameter of valve opening

Assuming that air is a perfect gas gives:

$$\text{Torque} = C_T \frac{P}{RT} (M\sqrt{\gamma RT}\,)^2 D^3 = C_T\, PM^2 \gamma D^3.$$

γ = Ratio of specific heats

P = Containment pressure

R = Universal gas constant

T = Air temperature

M ... number

Ratioing the Reference 1 torque values (subscript P) to the actual torque values (subscript a) yields:

$$\frac{(\text{Torque})_P}{(\text{Torque})_a} = \frac{(C_T\ \gamma\ D^3)_P}{(C_T\ \gamma\ D^3)_a} \times \frac{(PM^2)_P}{(PM^2)_a}$$

Since C_T, γ and D are constant for this comparison:

$$(\text{Torque})_a = \frac{(PM^2)_a}{(PM^2)_P} \times (\text{Torque})_P$$

For the maximum hydrodynamic torque case, at a valve disc angle of 15° from full open, the postulated containment pressure is 28.8 psia, which yields a Mach number of 1.0 at the valve disc edge. Therefore, the maximum hydrodynamic torque is determined as follows:

P_P = 42.7 psia $\qquad$ P_a = 28.8 psia

M_P = 1.0 $\qquad$ M_a = 1.0

$(\text{Torque})_P$ = 166,710 inch-pounds (Reference 1)

$$(\text{Torque})_a = \frac{(28.8 \times 1.0^2)}{(42.7 \times 1.0^2)} (166{,}170) = 112{,}077 \text{ in-pounds.}$$
(tending to close valve)

Conservatism was introduced by analyzing the upstream valve for flow forces while assuming the downstream valve and piping were not connected.

Valve actuator force was conservatively calculated based on the assumption that the valve actuator torque balanced the flow induced hydrodynamic torque generated in the valve at each valve disc angle. The valve actuator is comprised of an air cylinder/spring combination. The spring is attached to a piston inside the air cylinder and forces the valve closed when there is no air in the cylinder. To open the valve, the cylinder is pressurized with air such that the air pressure on the piston overcomes the spring force. Upon a containment isolation signal, the air in the cylinder is bled out through an orifice thus permitting the spring to gradually close the valve. If the flow induced hydrodynamic torque is positive this closing torque. If the flow induced hydrodynamic torque is negative (to open the valve), the effect is to compress the spring. The lower curve in Figure 3-3 represents the torque available from the spring force which can resist this opening torque. Since the upper and lower curves envelope the calculated hydrodynamic torque, during a postulated design basis Loss of Coolant Accident, the valve will close at the normal operating rate governed by the initial air pressure, orifice size and spring constant.

3.3 Analytical Methods

The butterfly valve was analyzed to determine the stress level margins in the valve load-carrying active components during the postulated flow condition.

The analysis consisted of determination of bending, torsion and shear loads on the valve shaft, key, pins and actuator arm at the critical valve disc angle of 15° from full open. Bending and torsional moments and shear forces were calculated at the actuator arm attachment, upper and lower bearings and the pins.

The maximum shear stress due to combined bending, torsion and shear was then calculated and compared either to an allowable of 1/2 yield strength, or to a value developed from the maximum distortion energy theory (Reference 7), to generate safety factor values.

TABLE 3-1

HENRY PRATT COMPANY TORQUE VS. VALVE DISC ANGLE RESULTS

VALVE DISC ANGLE (Degrees from full open)	DYNAMIC TORQUE (inch-pounds closing valve)
0	82,968
5	124,156
10	148,095
15	166,170
20	148,392
25	134,665
30	125,752
35	111,806
40	98,875
45	81,055
50	62,171
55	44,218
60	28,230
65	16,611
70	11,147
75	5,638
80	3,347
85	1,181
90	- 42,025

TABLE 3-1

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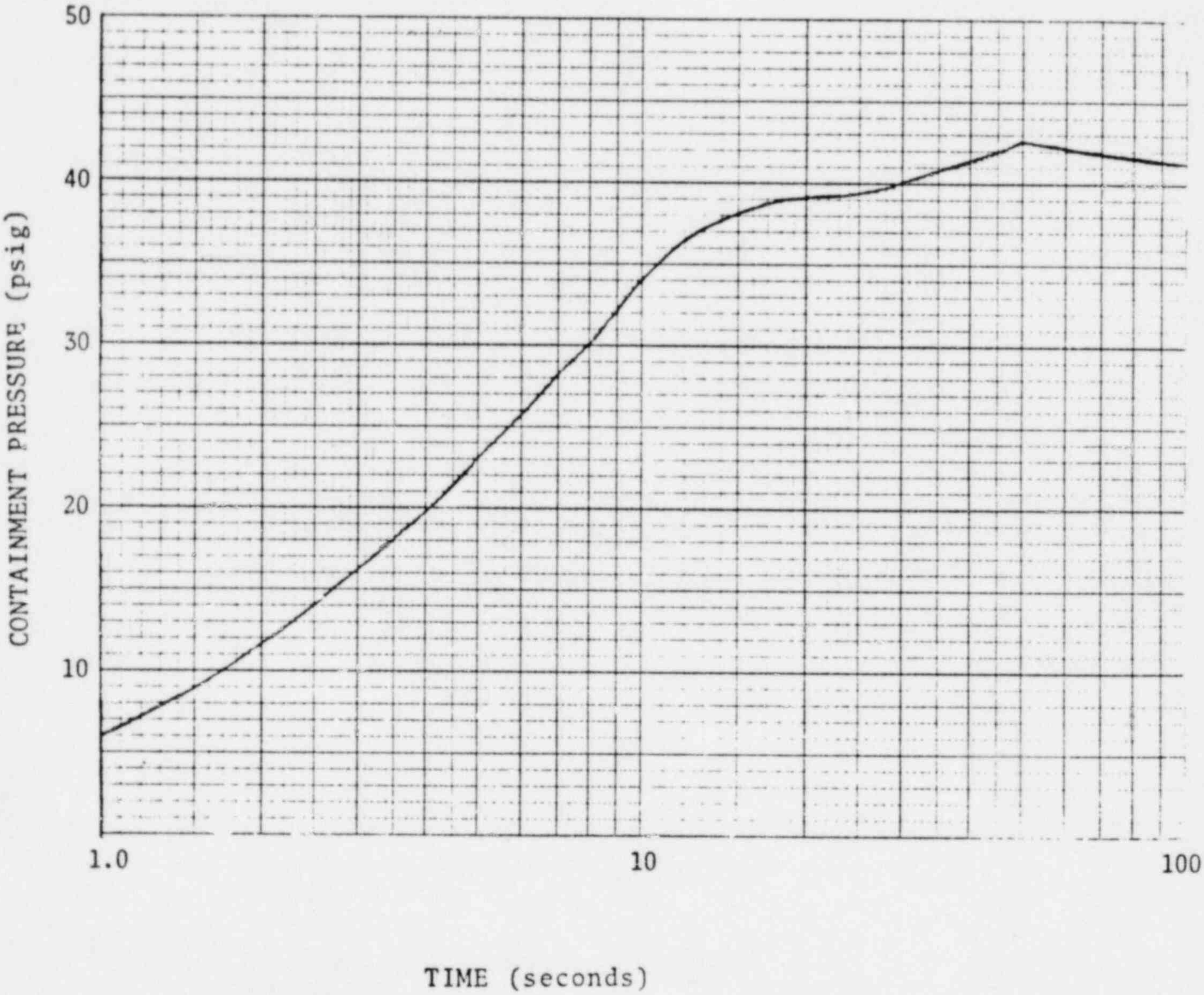
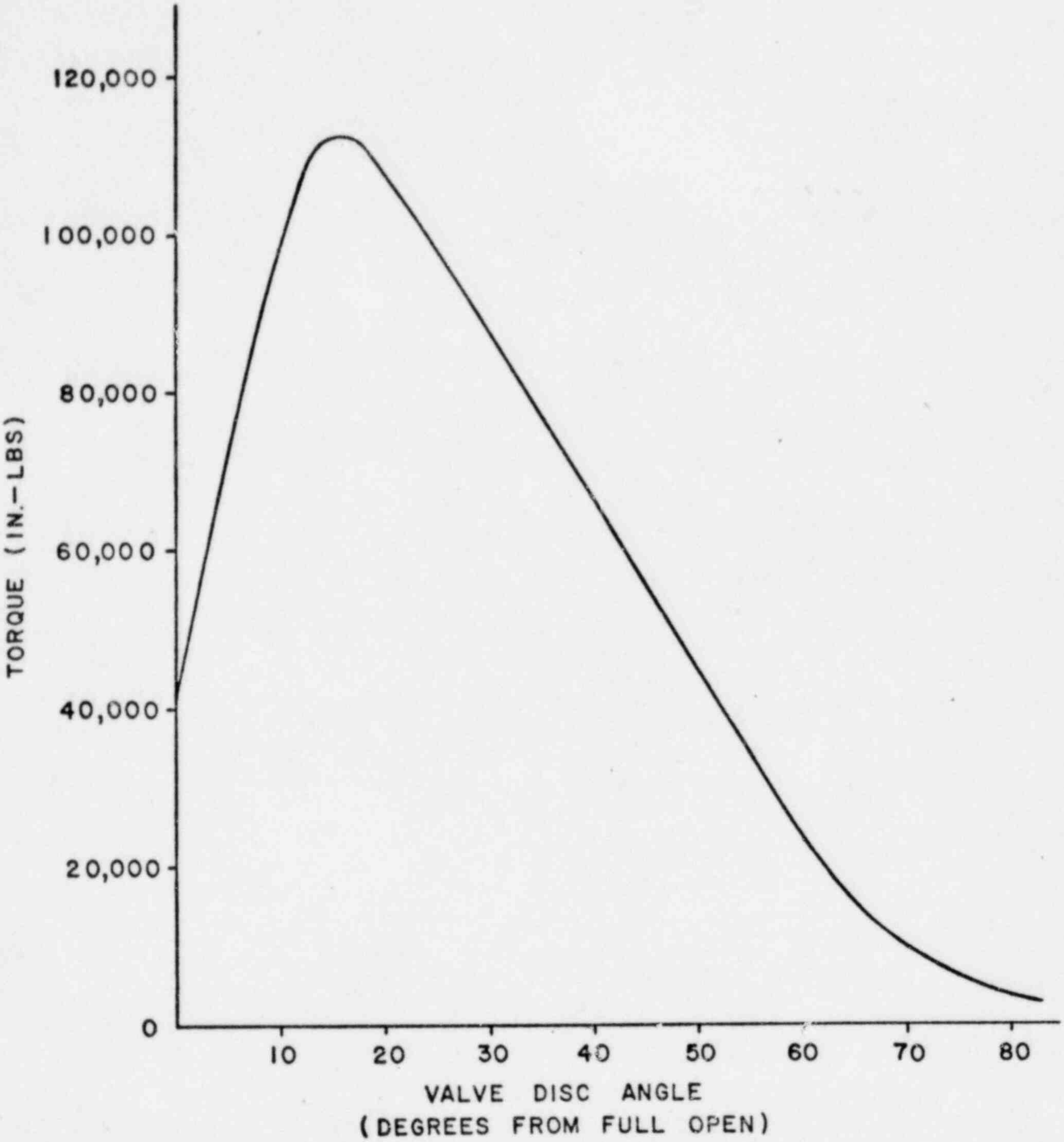


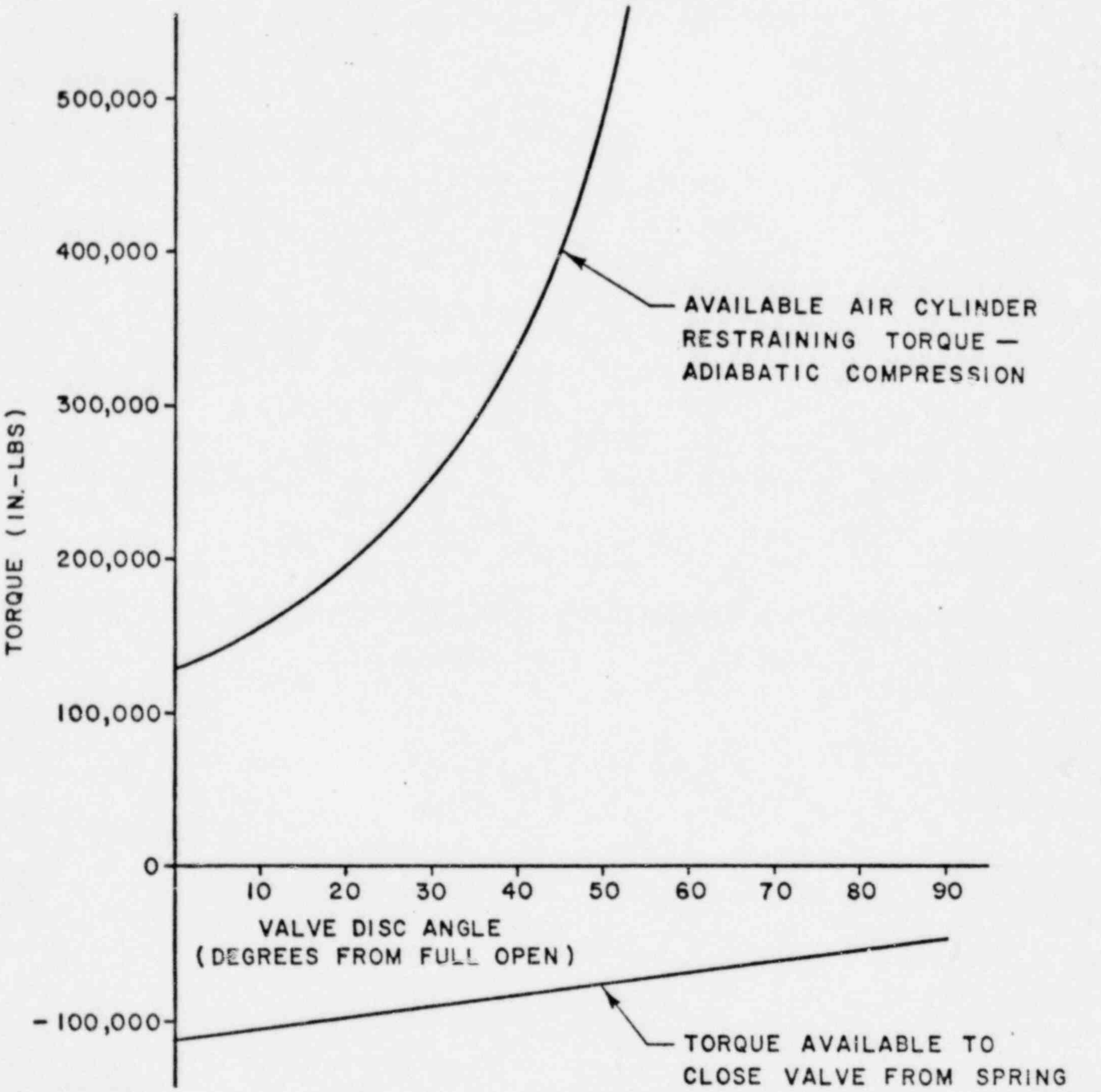
CONTAINMENT PRESSURE VS. TIME

FIGURE 3-1



HYDRODYNAMIC TORQUE VS. VALVE DISC ANGLE

FIGURE 3-2



AVAILABLE TORQUE IN SYSTEM

FIGURE 3-3

4.0 STRESS ANALYSIS RESULTS

The stress and safety factor values from the stress analysis, Appendix A, are presented in Table 4-1. The critical location in the valve was determined to be the shaft at the upper valve disc-to-shaft pin, where the stress level was conservatively calculated to be 97% of the allowable.

TABLE 4-1

SUMMARY OF STRESS RESULTS

	STRESS (ksi)	PERCENTAGE OF ALLOWABLE
SHAFT:		
SHEAR STRESS AT KEY	7.63	51%
SHEAR STRESS AT UPPER BEARING	12.47	83%
SHEAR STRESS AT PINS	16.79	97%*
KEY:		
SHEAR STRESS	9.37	62%
COMPRESSIVE STRESS	14.99	50%
PINS:		
SHEAR STRESS	7.13	48%
ACTUATOR ARM:		
BENDING STRESS	5.25	18%

* BASED ON MAXIMUM DISTORTION ENERGY THEORY SHEAR STRESS ALLOWABLE OF 17.31 KSI.

5.0 CONCLUSIONS

The stress analysis of the Pratt 42-inch butterfly valve demonstrates that the loads and stresses imposed upon the active load-carrying components during a valve closure under a postulated design basis Loss of Coolant Accident event are within acceptable limits. The stress margins are sufficient to ensure no significant deformation of the active valve parts will occur when the valve is used as a containment isolation valve in the Zion Station, Units 1 and 2. The loads used in the stress analysis were based upon results from the Henry Pratt Company.

6.0 REFERENCES

1. J. E. Sirovatka (Henry Pratt Company) letter to Commonwealth Edison Company, "Commonwealth Edison, Zion Nuclear Plant, P. O. 241329, HPCo Ref. D-28504", dated April 18, 1980.

2. Nuclear Regulatory Commission letter from Mr. A. Schwencer to Mr. Cordell Reed (Commonwealth Edison Company) "Containment Purging During Normal Plant Operation", dated November 29, 1978.

3. Nuclear Regulatory Commission letter from Mr. Darrell G. Eisenhut to All Light Water Reactors, "Containment Purging and Venting During Normal Operation - Guidelines for Valve Operability" dated September 27, 1979.

4. Nuclear Regulatory Commission letter from Mr. A. Schwencer to Mr. D. Louis Peoples (Commonwealth Edison Company), "Containment Purging and Venting During Normal Operation", dated October 23, 1979.

5. "Figure 14.3.4-6, Pressure-Temperature Curve, Zion Nuclear Power Station, Final Safety Analysis Report".

6. T. Sarpkaya, "Torque and Cavitation Characteristics of Butterfly Valves," Journal of Applied Mechanics, Transactions of the ASME, Number 60-WA-105, December 1961, pp. 511 - 518.

7. Robert C. Juvinall, <u>Engineering Considerations of Stress, Strain and Strength</u>, McGraw-Hill Book Company, 19 Edition pp. 85-89.

APPENDIX A

N TECH Stress Analysis of 42-Inch

Butterfly Valve

nutech
San Jose, California

Project ZION Nuclear Power Station File No. 64.802.0006
Owner Commonwealth Edison Company
Client Commonwealth Edison Company

STRESS ANALYSIS OF PRATT 42"

BUTTERFLY VALVE

SHAFT:

LOADS ON VALVE SHAFT

1. THE MAXIMUM HYDRODYNAMIC TORQUE (T_h) TENDING TO CLOSE THE VALVE, CALCULATED FROM RESULTS GENERATED IN THE VALVE TEST (REF. 1, PAGE 18) IS APPROXIMATELY 112^{IN-K} WHEN THE VALVE IS 75° FROM THE CLOSED POSITION (FIG. 1). THE ACTUATOR WAS LOCATED 75° FROM THE FLOW DIRECTION. (REF. 4).
2. RESTRAINING TORQUE BY ACTUATOR (T_a) = 112^{IN-K}
3. PISTON FORCE GENERATED RESTRAINING TORQUE (F_a)

$$F_a = T_a / L_a = 112/10.15 = 11.03^{K}$$

WHERE

L_a = ACTUATOR ARM TO SHAFT LENGTH (PAGE 16)

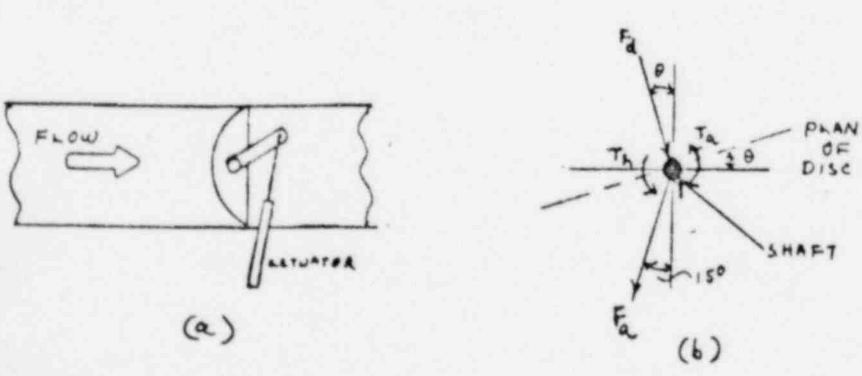


FIG. 1

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4. CENTER OF PRESSURE OF HYDRODYNAMIC FORCE (F_α) ON DISC FROM RESULTS GENERATED IN VALVE TEST (REF. 1).

AT 15° FROM FULL OPEN, PRATT DATA SHOWS:

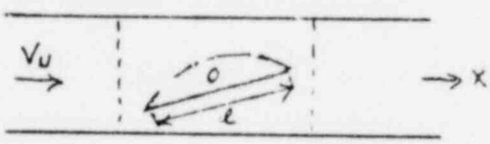
TORQUE = 166,283 in-lbs

PRESSURE UPSTREAM (P_U) = 42.7 psia.

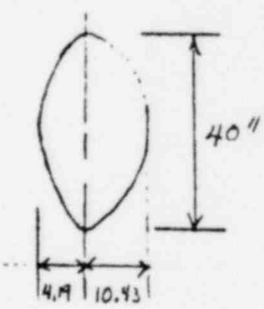
ΔPRESSURE ACROSS DISC (ΔP_{DISC}) = 20.06 psi

DENSITY UPSTREAM (ρ_U) = 0.155 lb/ft³

FLOW RATE = 55,544 lb/min



PROJECTED AREA OF DISC IN X DIRECTION IS ELLIPSE SHAPED



$$A_D = \frac{\pi(4.19)(20)}{2} + \frac{\pi(10.43)(20)}{2}$$

$$= 459.30 \text{ in}^2$$

$$\text{FLOW AREA } (A_F) = \frac{\pi}{4}\ell^2 - A_D$$

$$= \frac{\pi}{4}(40)^2 - 459.3$$

$$= 1256.64 - 459.3 = 797.34 \text{ in}^2$$

FROM MOMENTUM EQUATION

$$F_X = \iint_{C.S.} \rho\vec{V}(\vec{V}\cdot d\vec{A}) = \rho V^2 A_D$$

FOR VELOCITY = UPSTREAM VELOCITY

$$\rho_U V_U A_D = \rho_T V_{max} A_f = 55{,}544 \text{ lb/MIN}$$

where

ρ_U = density upstream

V_U = Velocity upstream

ρ_T = density AT THROAT

SINCE

$$A_D/A_F = 459.3/797.34 = 0.58$$

$$\therefore \rho_U V_U A_D = \rho V A_f (A_D/A_f) = \rho V A_D$$

$$= 55{,}544 (0.58) = 32215.5 \text{ lb/MIN}$$

$$V_U = \frac{Q}{\rho A} = \frac{55{,}544 (144 \text{ in}^2/\text{ft}^2)}{(60 \text{ s/m})(1256.64 \text{ in}^2)(.155 \text{ lbm/ft}^3)}$$

$$= 684.39 \text{ ft/sec}$$

$$\therefore F_U = (\rho V A_D)(V_U) = \frac{(32215.5)(684.39)}{(60 \text{ s/min})(32.2 \frac{\text{lbm-ft}}{\text{lbf-sec}^2})}$$

$$= \underline{11412 \text{ lbf}} \text{ for PRATT CONDITIONS}$$

$$\text{CENTER OF PRESSURE} = \frac{166{,}283 \text{ in-lbf}}{11412 \text{ lbf}} = 14.57 \text{ in. from sh-ft}$$

FOR VELOCITY = SONIC VELOCITY

$$V_{MAX} = M\sqrt{\gamma g R T}$$

where $M = 1.0$

$$T = 283°F = 743°R$$

$$V_{MAX} = \sqrt{(1.4)(32.2)(53.3)(743)} = 1336.13 \text{ ft/sec.}$$

$$F = (\rho V A_D) V_{MAX}$$

$$= \frac{(32215.5 \frac{lbm}{min})(1336.13 \text{ ft/s})}{(60 \text{ s/min})(32.2 \text{ lbm-ft/lbf-sec})} = 22279.62 \text{ lbf.}$$

$$\text{CENTER OF PRESSURE} = \frac{166{,}283}{22279.62} = 7.46 \text{ in.}$$

SINCE THE CASE WE ARE INVESTIGATING AND THE CASE PRATT INVESTIGATED BOTH EXHIBIT SONIC FLOW AT THE THROAT AND HAVE THE SAME FLOW PATH GEOMETRY, IT IS REASONABLE TO ASSUME THAT, (1) PRESSURE AND VELOCITY PROFILES ARE SIMILIAR & (2) THE CENTER OF PRESSURE WILL EXIST IN THE SAME LOCATION ON THE DISC. ALSO, IT IS REASONABLE TO ASSUME THE CENTER OF PRESSURE WILL LIE BETWEEN THE CENTERS CALCULATED FOR THE MINIMUM UPSTREAM VELOCITY AND THE MAXIMUM (SONIC) VELOCITY. THEREFORE, STRESSES AND FORCES WILL BE CALCULATED FOR THESE BOUNDING LOCATIONS FOR THE CASE WHERE THE HYDRODYNAMIC TORQUE IS 112 $^{K\text{-}IN}$.

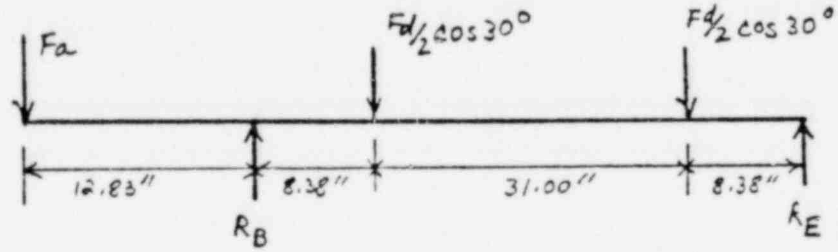
$$F_{d_1} = 112^{K\text{-}IN} / 14.57 \text{ IN} = 7.69^{K}$$

$$F_{d_2} = 112^{K\text{-}IN} / 7.46 \text{ IN} = 15.01^{K}$$

5. Maximum Moments & Shears

@ 75° from closed, forces in F_a plane (Fig. from Ref. 4)

CASE I. $F_d = 7.69^K$

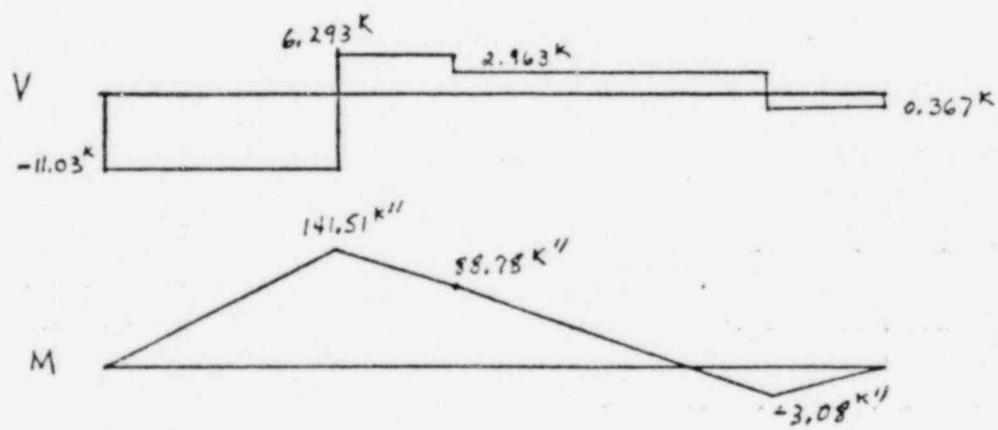


$F_d/2 \cos 30° = 3.33^K$

$\circlearrowleft \Sigma M_B = 0$

$R_E = 3.33 - 11.03\left(\frac{12.83}{47.76}\right) = 0.367^K$

$R_B = 11.03 + 6.66 - 0.367 = 17.323^K$

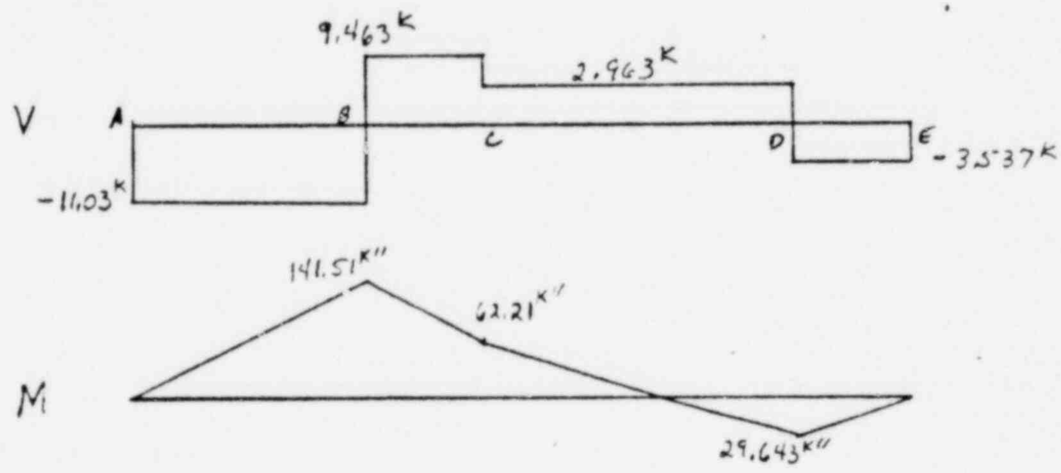


CASE II. $F_d = 15.01^K$

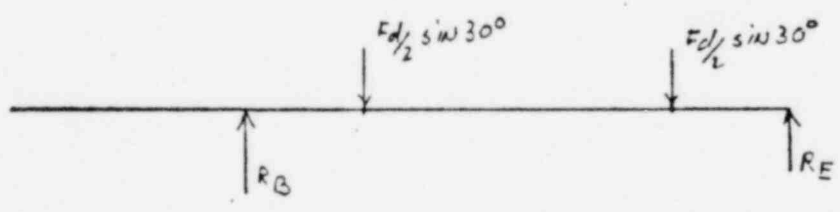
$F_d/_2 \cos 30° = 6.5^K$

$R_E = 6.5 - 2.963 = 3.537^K$

$R_B = 11.03 + 13.0 - 3.537 = 20.493^K$



@ 75° from CLOSED, FORCES PERPENDICULAR TO F_a



CASE I $F_d/_2 \sin 30° = 1.92^K$

CASE II $F_d/_2 \sin 30° = 3.75^K$

CASE I

$V_B = V_C = V_E = 1.92^K$

$M_B = 0$

$M_C = 16.09^{K''}$

CASE II

$V_B = V_C = V_E = 3.85^K$

$M_C = 32.22^{K''}$

MAX. MOMENTS, SHEARS, TORQUES

$M_B = 141.51^{K''}$

$T_B = 112^{''K}$

$V_B = 11.03^K$

$M_C = \sqrt{(16.09)^2 + (88.78)^2} = 90.23^{K''}$

$T_C = 112^{''K}$

$V_C = \sqrt{(2.963)^2 + (3.75)^2} = 4.78^K$

6. MAXIMUM SHEAR STRESS @ BEARING (B)

$$\sigma_B = \frac{M_c}{I} = \frac{32M}{\pi D^3} = \frac{32(141.51)}{\pi(4.25)^3} = 18.777 \text{ ksi}$$

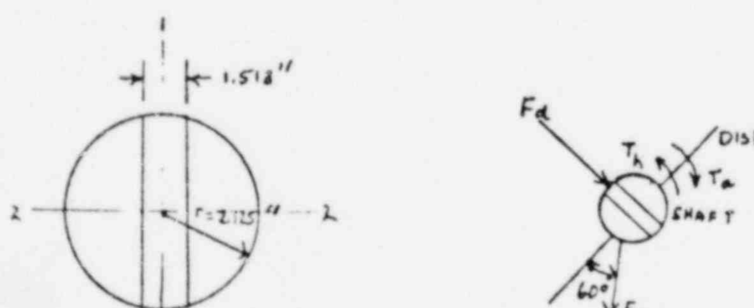
$$\tau_T = \frac{16T}{\pi D^3} = \frac{16(112)}{\pi(4.25)^3} = 7.431 \text{ ksi}$$

$$\tau_V = \frac{V}{A} = \frac{11.03}{\pi/4(4.25)^2} = 0.78 \text{ ksi}$$

$$S = \sqrt{\left(\frac{\sigma}{2}\right)^2 + (\tau_T + \tau_V)^2} \quad \text{(REF. 5)}$$

$$= \sqrt{\left(\frac{18.777}{2}\right)^2 + (7.431 + 0.78)^2} = \underline{12.47} \text{ ksi} < \sigma_y/2 = 15 \text{ ksi}$$

7. MAXIMUM SHEAR STRESS @ PINS (C)



(a) SHAFT AT C.

(b) FORCES

(FIG. 2)

$$I_{11} = \pi r^4 - \frac{bh^3}{12}$$

$$= \pi(2.125)^4 - \frac{4.25(1.513)^3}{12}$$

$$= 14.788 \text{ in}^4$$

$$I_{22} = \pi(2.125)^4 - \frac{1.513(4.25)^3}{12}$$

$$= 6.336 \text{ in}^4$$

$$A = \pi r^2 - bh = 7.756 \text{ in}^2$$

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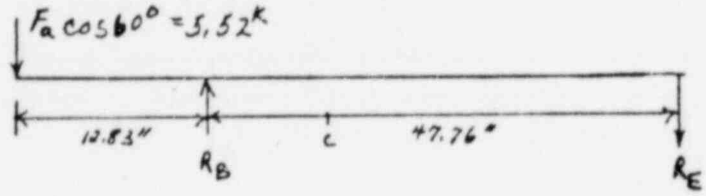
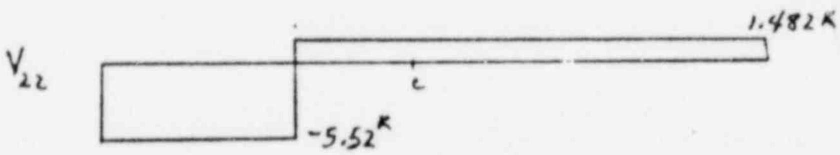
PERPENDICULAR TO HOLE (// to DISC FACE) (SEE FIG. 2)

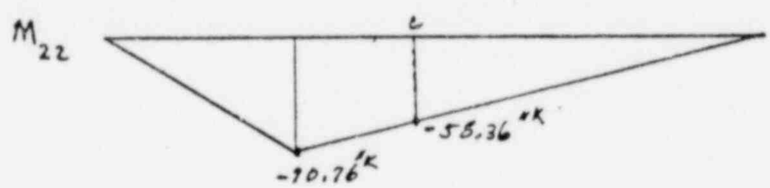


$\Sigma M_B = 0$

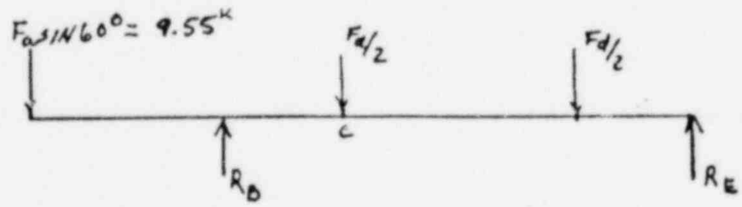
$$R_E = F_a \cos 60° (12.83")/47.76" = \frac{11.03(.50)(12.83)}{47.76} = 1.482^K$$

$$R_B = 6.997^K$$





PARALLEL TO HOLE (⊥ TO DISC FACE)



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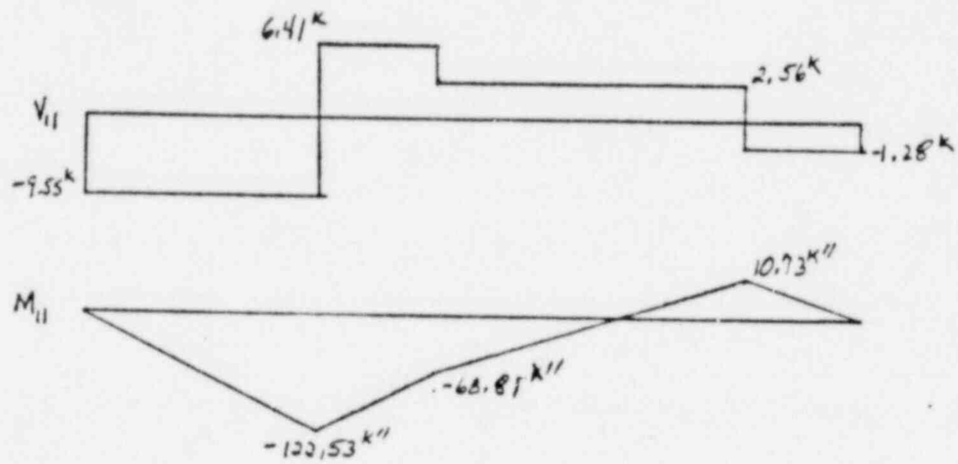
$F_a \sin 60° = 11.03 \sin 60° = 9.55^k$

$F_d/2 = 7.69/2 = 3.85^k$

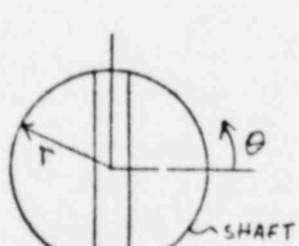
$R_E = 3.85 - \left[\frac{9.55(12.83)}{47.76}\right] = 1.28^k$

$R_B = 9.55 + 7.69 - 1.28 = 15.96^k$



Check MOMENT

$\left[(68.81)^2 + (58.36)^2\right]^{1/2} = 90.23^{k''}$



@ MAX. STRESS POINT

$$\frac{M_{22}\, r\cos\theta}{I_{11}} + \frac{M_{11}\, r\sin\theta}{I_{22}} = \sigma_b$$

where $r = 2.125''$

$I_{11} = 14.788 \text{ in}^4$

$I_{22} = 6.336 \text{ in}^4$

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TABLE 1

θ (DEG)	$\frac{N_{22} r \cos\theta}{I_{11}}$	$\frac{M_{11} r \sin\theta}{I_{22}}$	σ_b (KSI)
0	8.39	0	8.39
15	8.10	5.97	14.07
30	7.26	11.54	18.80
45	5.93	16.32	22.25
60	4.19	19.99	24.18
75	2.17	22.29	24.46
90	0	23.08	23.08

FROM TABLE 1

$$\sigma_{b\,max} = 24.46^{\,ksi}$$

$$\tau_T = \frac{T}{\pi D^3/16 - d D^2/6} = 10.6464^{\,ksi} \quad \text{(REF. 6, page 244)}$$

where $d = 1.513''$

$D = 4.25''$

$T = 112^{\,"k}$

$$\tau_V = \frac{V}{A} = \left[6.41^2 + 1.48^2\right]^{1/2} / 7.756 = 0.85 \text{ ksi}$$

$$S = \sqrt{\left(\frac{\sigma_b}{2}\right)^2 + (\tau_T + \tau_V)^2} = \underline{\underline{16.79}}^{\,ksi}$$

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8. MAXIMUM SHEAR STRESS AT KEYWAY (A)

$\sigma_b = 0$

$\tau_T = \frac{Tc}{J} = \frac{16(112)}{\pi (4.25)^3} = 7.43^{ksi}$

$\tau_V = V/A = 11.03 / [\pi (4.25)^2 - (1.125)(0.703)] = 0.20^{ksi}$

$S = 7.63^{ksi}$

STRESS ON TWO TAPER PINS @ C S.S. 18-8 TYPE 304

$S_y = 30^{ksi}$

UNIT WORKING STRESS ON PINS IN SHEAR (REF. 7)

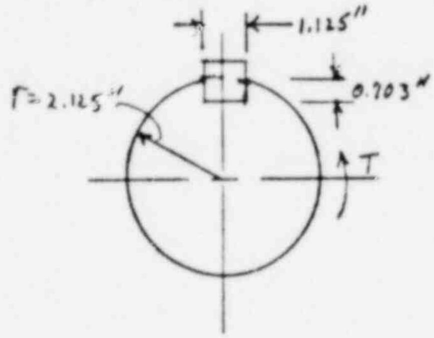
FOR #14 PINS

$S_U = \frac{1.27T}{2D \cdot d^2} = \frac{1.27(112)}{2(4.25)(1.532)^2} = 7.13^{ksi/PIN}$

where D = dia. of shaft

d = dia. of pin

STRESS ON KEY @ A AISI COLD DRAWN STEEL



$T = 112^{KIN}$

$F = T/r = \frac{112}{2.125} = 52.71^{K}$

By distortion-energy theory, shear strength is

$S_{sy} = 0.577\, S_y$ (Ref. 8)

$= 0.577(30) = 17.31^{ksi}$

SHEAR ACROSS KEY (REF. 8)

$\tau = F/tl = 52.71/1.125(5") = 9.37^{ksi}$

CRUSHING STRESS (REF. 8)

$A_s = 0.703(5") = 3.515\ in^2$

$S_c = \frac{F}{A_s} = 14.99^{ksi} < \sigma_y = 30^{ksi}$

SUMMARY OF RESULTS

PART	MAX. SHEAR STRESS (KSI)	MAX. BENDING STRESS (KSI)	MAX. COMPRESSIVE STRESS (KSI)	PERCENTAGE OF ALLOWABLE
SHAFT				
@ PINS	16.97	—	—	97%*
@ KEYWAY	7.63	—	—	51%
@ BEARING	12.47	—	—	83%
TAPER PINS	7.13	—	—	48%
KEY	9.37	—	14.99	62%
LEVER	—	5.25	—	18%

MAX. SHEAR STRESS FOR SHAFT = $\sqrt{(\sigma_y/2)^2 + \tau^2}$

MAX. SHEAR STRESS FOR TAPER PINS = $\tau_{TORSION}$

MAX. SHEAR STRESS FOR KEY = τ_{SHEAR}

MAX. COMPRESSIVE STRESS = σ_{AXIAL}

STRESS ALLOWABLE = $\sigma_y/2$ = 15 ksi FOR SHEAR

= σ_y = 30 ksi FOR COMPRESSION

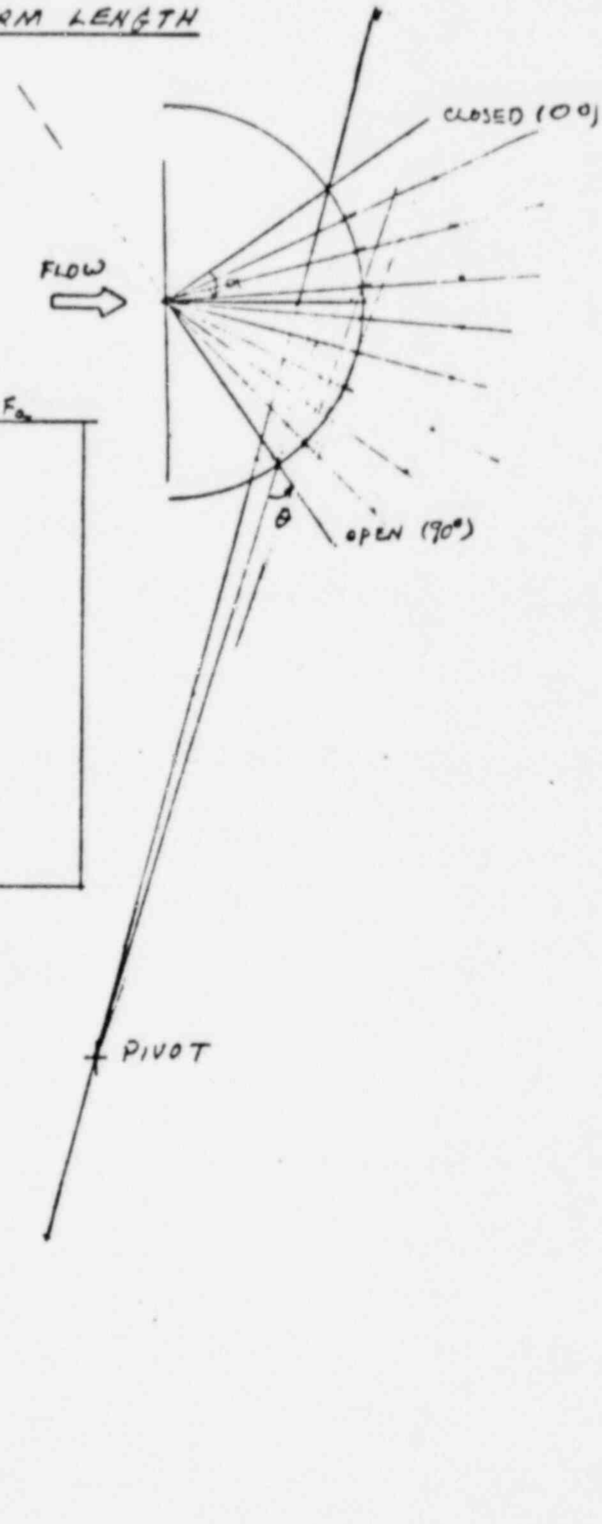
*BASED ON MAXIMUM DISTORTION ENERGY THEORY SHEAR STRESS ALLOWABLE OF 17.31 ksi (Ref. 6, p. 85-89)

CALCULATION OF EFFECTIVE LEVER ARM LENGTH

ASSUME:

FULLY COMPRESSED SPRING
IN OPEN POSITION

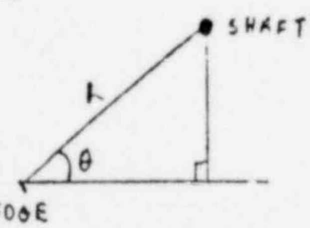
MAX EXTENSION = 20" @ CLOSED



α_{SHAFT}	EXTENSION	θ_{Fa}	L'	F_a
0°	20"	50°	9.74	
10	|	51	9.88	
20	|	52	10.02	
30	|	54	10.29	
40	|	54	10.29	
50	|	55	10.42	
60	|	55	10.42	
70	|	54	10.29	
80	|	52	10.02	
90	0	51	9.88	

$L' = L \sin\theta$

$L = 12.7188''$



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SEISMIC FORCES

A. NATURAL FREQUENCY

1. VOL. + MASS OF DISC (DWG B-7669, PRATT Co.)

$\text{VOL. DISC PLATE} = \pi r^2 t = \pi\left(\frac{36.5}{2}\right)^2(1.25) = 1307.93\ in^3$

$m = (.284\ \#/in^3)(1307.93\ in^3) \div (386.40) = 0.961\ \#s^2/in$

2. VOL. + MASS OF DISC EDGE (DRWG B-7659 PRATT CO)

$VOL = \frac{1}{2}''(119.375'')(2.5'') = 149.22\ in^3$

$m = .284(149.22) \div 386.40 = 0.116\ \#s^2/in$

3. VOL. + MASS OF DEEP DISC PLATE (DWG B-7673, PRATT CO.)

$VOL = \frac{2}{3}\pi\left(R^2H - r^2h\right) = \frac{2}{3}\pi\left(26.25^2 - 26^2\right)(7.438) = 203.49\ in^3$

$m = .284(203.49) \div 386.40 = 0.150\ \#s^2/in$

4. VOL. + MASS OF TOP HUB BLOCK (DWG B-7656, PRATT CO.)

$V = (7.0)(6.5)(8.625) - (2.25)(6.5)(1) - .7854(3.875)^2(7.75)$

$+ \frac{\pi}{4}(4.687)^2(.75) = 299.35\ in^3$

$m = (.284)(299.35) \div 386.4 = 0.220\ \#s^2/in$

5. VOL. + MASS OF BOTTOM HUB BLOCK (DWG. A-4360, PRATT)

$V = (6.375)(6.5)(4.25) + \frac{\pi}{4}(4.687)^2(.75) - .7854(3.875)^2(5)$

$= 130.08\ in^3$

$m = .284(130.08) \div 386.4 = 0.096\ \#s^2/in$

6. TOTAL DISC MASS $= 0.961 + .110 + .15 + .220 + .096 = 1.537\ \#s^2/in$

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7. VOL AND MASS OF SHAFT

$VOL = .7854 D^2 L = .7854 (4.25)^2 (47) = 666.76 \ in^3$

$m = .284 (666.76) \div 386.4 = 0.49 \ \# s^2/in$

UNIFORM MASS/IN $= 0.49 \div 47 = 0.010 \ \# s^2/in/in$

8. INERTIA OF SHAFT

$I = \pi R^4/4 = \pi \left(\frac{4.25}{2}\right)^4/4 = 16.015 \ in^4$

9. NATURAL FREQUENCY (FROM ROARK - FIFTH EDITION) CASE 1-C

$$f_1 = \frac{6.93}{2\pi} \sqrt{\frac{EI}{m_D l^3 + .383 m_s l^4}}$$

$$= \frac{6.93}{2\pi} \sqrt{\frac{(29 \times 10^6)(16.015)}{(1.537)(47)^3 + .383(.01)(47)^4}} = 56.30 \ cps.$$

period = 0.018 sec

10. FROM REF. 10, THE acceleration @ .035 sec (30 Hz) = 0.4g

FORCE = $ma = (1.537 + .49)(.4)(386.4) = \underline{313.29}$ lbs

To be conservative, A 1.5 factor will be applied to account for the other modes. Also, since response DATA is NOT AVAILABLE at 56 Hz., which is close to the ZPA, this would be extremely conservative.

FORCE = $1.5(313.29) = 470$ lbs

THE SEISMIC FORCE = 6% of CASE I (pg. 5) AND IS EQUAL TO 3% of CASE II (pg. 6) therefore can be considered insignificant.

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INSERT A

CALCULATION OF HYDRODYNAMIC TORQUE

THE HYDRODYNAMIC TORQUE VALUES CALCULATED IN REFERENCE 1 WERE PREDICATED ON A CONTAINMENT PRESSURE OF 42.7 PSIA. TAKING THE POSTULATED PRESSURE-TIME CURVE FOR THE ZION NUCLEAR STATION (REFERENCE 9) INTO ACCOUNT RESULTS IN LOWER HYDRODYNAMIC TORQUE VALUES AS FOLLOWS:

1. PER THE PRATT VALVE CATALOGUE, THESE BUTTERFLY VALVES CLOSE 90% IN 3 SECONDS OR

$$\frac{3 \text{ SECONDS}}{81 \text{ DEGREES}} = .037 \frac{\text{SEC}}{\text{DEG}}$$

2. ASSUME A 2.0 SECOND INSTRUMENT LAG TIME BETWEEN THE TIME A POSTULATED LOCA OCCURS AND THE TIME THE VALVE RECEIVES A CONTAINMENT ISOLATION SIGNAL.

TIME AFTER LOCA VS. VALVE DISC ANGLE (α):

$$\text{TIME} = 2.0 + (.037)(\alpha) \qquad 0^\circ < \alpha \le 81^\circ$$

3. TORQUE $= C_T \rho V^2 D^3$ (REF. 2)

ASSUMING AIR TO BE A PERFECT GAS

$$\text{TORQUE} = C_T \frac{P_{CONT.}}{RT} \left(M\sqrt{\gamma R T}\right)^2 D^3$$

$$= (C_T \gamma D^3)(P_{CONT.}) M^2$$

- FOR $P_{CONT.} > \frac{14.7}{(.528)} = 27.8$ PSIA, $M = 1.0$

$$\therefore \frac{(\text{TORQUE})_{PRATT}}{(\text{TORQUE})_{POSTULATED}} = \frac{(C_T \gamma D^3 M^2)(P_{CONT.})_{PRATT}}{(C_T \gamma D^3 M^2)(P_{CONT.})_{POSTULATED}}$$

SINCE $(C_T \gamma D^3 M^2)$ IS CONSTANT

$$(\text{TORQUE})_{PRATT} = \frac{(P_{CONT.})_{PRATT}}{(P_{CONT.})_{POSTULATED}} (\text{TORQUE})_{POSTULATED}$$

- FOR $P_{CONT.} < 27.8$ PSIA :

$$(\text{TORQUE})_{PRATT} = \frac{(P_{CONT})_{PRATT}}{(P_{CONT} M^2)_{POSTULATED}} (\text{TORQUE})_{POSTULATED}$$

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4. ASSUME VALVE CLOSES COMPLETELY 5.0 SECONDS FOLLOWING ISOLATION SIGNAL.

α (degrees)	Time (secs)		$P_{CONTAINMENT}$ (psia)	Torque (in-lbs)
0	2.0	($M=.91$)	25.2	40,543
10	2.37		28.2	97,505
15	2.56		28.8	112,077
20	2.74		29.5	102,519
30	3.11		30.1	88,645
40	3.48		32.7	75,719
50	3.85		33.9	49,353
60	4.22		35.2	23,272
70	4.59		36.5	9,528
80	4.96		37.7	2,955
90	7.00		42.7	-42,025

ALSO $TORQUE = C_T P M^2 \gamma D^3$

FOR SUBSONIC POSTULATED FLOW:

$$\frac{TORQUE_{PRATT}}{TORQUE_{POSTULATED}} = \frac{(PM^2)_{PRATT}}{(PM^2)_{POST.}} = \frac{(42.7)(1.0)^2}{(25.2)(.91)^2} = \underline{2.05}$$

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4. HENRY PRATT CO., "42 INCH BUTTERFLY DRAWINGS & SPECIFICATIONS" (SEE PAGE 16.8.)
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8. SHIGLEY, JOSEPH EDWARD, "Mechanical Engineering Design," McGraw-Hill Book Co., 1972, Pg. 333
9. FIGURE 14.3.~~4-6~~, PRESSURE-TEMPERATURE CURVE, ZION NUCLEAR POWER STATION, FINAL SAFETY ANALYSIS REPORT
10. P SPECTRA(ME-164) ARS DIGITATION, ZION STATION FROM STONE & WEBSTER 11/Oct.

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REFERENCES

DRAWINGS, HENRY PRATT CO., AURORA, ILLINOIS

A-4452 ROLLER BEARING

B-982 STANDARD SQUARE KEYS
B-4381 STANDARD TAPER PIN BLANKS
B-7980 FIN. FAB. LEVER ASSY. FOR #4x22 CYL. OPER.
B-7733 CONTROL ASSY.
B-7669 FINISHED DISC PLATE
B-7672 FINISHED DEEP DISC PLATE
B-7673 ROUGH DISC PLATE
B-7758 CYLINDER OPERATOR ASSEMBLY
B-7658 HUB ASSEMBLY FOR OFFSET DISC
B-7756 4½" Dia. SHAFT FOR 41" VALVE

C-961 41" ROUGH FAB. OFFSET DISC WELDING ASS'Y.
C-963 41" DISC ASSEMBLY FOR OFFSET DISC
C-974 41" ROLLER BEARING, WELD END, OFFSET DISC FABRICATED ASS'Y.
C-962 41" FINISHED FAB. OFFSET DISC
C-418 SPRING ASSEMBLY WITH CYL.

F-232 41" RIA SPECIAL ROLLER BEARING, SPECIAL WELD END FABRICATED BODY

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