50-302

H. Pratt Coanalysis of REactor Building Purge Valves

Rec'd w/ltr dtd 11-4-81

ACC.# 8111100419

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(A) Pressure v. Time Graph

(B) Pratt letter regarding additional information

(C) Customer/engineer response to request for information

(2) Valve Assembly Stress Report

(3) Operator Ratings

(4) Supplemental Torque Calculations

(5) General Arrangement and Cross-Section Drawings

8111100425

I. Introduction

This investigation has been made in response to a request by the customer/engineer for evaluation of containment isolation/purge valves during a faulted condition arising from a loss of coolant accident (LOCA).

The analysis of the structural and operational adequacy of the valve assembly under such conditions is based principally upon containment pressure vs. time data, system response (delay) time, piping geometry upstream of the valve, back pressure due to ventilation components downstream of the valve, valve orientation and direction of valve closure.

The above data as furnished by the customer/engineer forms the basis for the analysis. Worst case conditions have been applied in the absence of definitive input.

II. Considerations

The NRC guidelines for demonstration of operability of purge and vent valves dated 9/27/79, have been incorporated in this evaluation as follows:

A.1. Valve closure time during a LOCA will be less than or equal to the no-flow time demonstrated during shop tests, since fluid dynamic effects tend to close a butterfly valve. Valve closure rate vs. time is based on a sinusoidal function.

2. Flow direction through valve contributing to highest torque; namely, flow toward the hub side of disc if asymmetric, is used in this analysis. Pressure on upstream side of valve as furnished by customer/engineer is utilized in calculations. Downstream pressure vs. LOCA time is furnished by customer/engineer or assumed to be worst case.

3. Worst case is determined as a single valve closure of the inside containment valve, with the outside containment valve fixed at the fully open position.

4. Containment back pressure will have no effect on cylinder operation since the same back pressure will also be present at the inlet side of the cylinder and differential pressure will be the same during operation.

5. Purge valves supplied by Henry Pratt Company do not normally include accumulators. Accumulators, when used, are for opening the valve rather than closing.

6. Torque limiting devices apply only to electric motor operators. Based on findings in this report, motor operator torque switches should be bypassed or removed to eliminate motor lock-up during the LOCA closure cycle.

7&8. Drawings or written description of valve orientation with respect to piping immediately upstream, as well as direction of valve closure, are furnished by customer/engineer. In lieu of input, worst case conditions have been applied to the analysis; namely, 90° elbow (upstream) oriented 90° out-of-plane with respect to valve shaft, and leading edge of disc closing toward outer wall of elbow. Effects of downstream piping on system back pressure have been covered in paragraph A.2. (above).

B. This analysis consists of a static analysis of the valve components indicating if the stress levels under combined seismic and LOCA conditions are less than 90% of yield strength of the materials used.

A valve operator evaluation is presented based on the operators ability to resist the reaction of LOCA-induced fluid dynamic torques.

C. Sealing integrity can be evaluated as follows:

Decontamination chemicals have very little effect on EPT and stainless steel seats. Molded EPT seats are generically known to have a cumulative radiation resistance of 1 x 10^8 rads at a maximum incidence temperature of 350°F. It is recommended that seats be visually inspected every 18 months and be replaced periodically as required.

Valves at outside ambient temperatures below 0°F, if not properly adjusted, may have leakage due to thermal contraction of the elastomer, however, during a LOCA, the valve internal temperature would be expected to be higher than ambient which tends to increase sealing capability after valve closure. The presence of debris or damage to the seats would obviously impair sealing.

III. Method of Analysis

Determination of the structural and operational adequacy of the valve assembly is based on the calculation of LOCA-induced torque, valve stress analysis and operator evaluation.

A. Torque calculation

The torque of any open butterfly valve is the summation of fluid dynamic torque and bearing friction torque at any given disc angle.

Bearing friction torque is calculated from the following equation:

$$T_B = P \times A \times U \times \frac{d}{2}$$

where

P = pressure differential, psia

A = projected disc area normal to flow, in^2

U = bearing coefficient of friction

d = shaft diameter, in.

Fluid dynamic torque is calculated from the following equations:

For subsonic flow

$$\left[R_{CR} \geq \frac{P_1}{P_2} > 1.07 \text{ (approx.)} \right]$$

$$T_D = D^3 \times C_{T1} \times P_2 \times \sqrt{\frac{K}{1.4}} \times F_{RE}$$

For sonic flow

$$\left[\frac{P_1}{P_2} \geq R_{CR} \right]$$

$$T_D = D^3 \times C_{T2} \times P_2 \times \sqrt{\frac{K}{1.4}} \times F_{RE} \qquad (F_{RE} \geq 1)$$

Where

T_D = fluid dynamic torque, in-lbs.

F_{RE} = Reynold number factor

R_{CR} = critical pressure ratio, (f (α))

P_1 = upstream static pressure at flow condition, psia

P_2 = downstream static pressure at flow condition, psia

D = disc diameter, in.

C_{T1} = subsonic torque coefficient

C_{T2} = sonic torque coefficient

K = isentropic gas exponent ($\simeq$ 1.2 for air/steam mix)

α = disc angle, such that 90° = fully open; 0° = fully closed

Note that C_{T1} and C_{T2} are a function of disc angle, an exponential function of pressure ratio, and are adjusted to a 5" test model using a function of Reynolds number.

Torque coefficients and exponential factors are derived from analysis of experimental test data and correlated with analytically predicted behavior of airfoils in compressible media.

Empirical and analytical findings confirm that subsonic and sonic flow conditions across the valve disc have an unequal and opposite effect on dynamic torque. Specifically, increases in upstream pressure in the subsonic range result in higher torque values, while increasing P_1 in the sonic range results in lower torques. Therefore, the point of greatest concern is the condition of initial sonic flow, which occurs at a critical pressure ratio.

The effect of valve closure during the transition from subsonic to sonic flow is to greatly amplify the resulting torques. In fact, the maximum dynamic torque occurs when initial sonic flow occurs <u>coincident</u> with a disc angle of 72° (symmetric) or 68° (asymmetric) from the fully closed position.

D-29254-1

JOB:FLOR.PWR;CRYST.RIV P2-VARIABLE SIZE ADJUSTED(REYNLDS NO.FNCTN!)
SAT.STEAM/AIR MIXTURE WITH 1.4 LBS STEAM PER 1-LBS AIR
SPEC.GR.= .738255 . MOL.WT.= 21.3872   KAPA(ISENT.EXP.)= 1.19775   R= 72.1972
GAS CONSTANT-CALC.
SONIC SPEED(MOVING MIXTR.)= 1316.65   FEET/SEC AT 225   DEG.

CRIT.CASE INLET VELOCITY IS 1.4606 TIMES HIGHER AS AIR CRIT.CASE INLET V1-OF 5 INCH MODEL

MAX. TORQUE IS AT THE CRITICAL PRESS.RATIO(.585-(5 IN)MODEL OR APPX .695051 (47.375   IN)WITH STMIX.)FIRST SONIC(@ 72 DEG.V.A.)

MAX.TORQUE INCLUDES SIZE EFFECT(REYNOLDS NO.ETC)APPX. X 1.37246   FOR 47.375 INCH BASIC LINE I.D.
ALL PRESSURES USED:STATIC(TAP)PRESS.-ABSOLUTE;P2 INCL.RECOVERY PRESS.
(TORQUE)CALC'S VALIDITY:P1/P2>1.07;

VALVE TYPE: 48"-R1A;1/6   CLASS 75
DISC SIZE: 46.718 INCHES   OFFSET ASYMMETRIC DISC
SHAFT DIA.:   4.75   INCHES
BRG. COEF. OF FRCTN.:   5.00000E-03
SEATING FACTOR: 15
INLET PRESS.VAR.MAX.:   50.7   PSIA
OUTLET PRESSURE(P6):   23.5   PSIA (72 DEG. ACTUAL PRESS.ONLY(VAR.))
MAX.ANG.FLOW RATE: 527196.   CFM; 645142.   SCFM; 35465.2   LB/MIN
CRIT.SONIC FLOW-90DG: 45093.9   LB/MIN AT 23.8381   INLET PSIA
VALVE INLET DENSITY: 6.72714E-02   LB/FT^3-MIN. .14391 LB/FT^3-MAX.
FULL OPEN DELTA P: 3.62416   PSI
SYSTEM CONDITIONS:
PIPE IN-PIPE-OUT -AND- AIR/STEAM MIXTURE SERVICE @ 225 DEG.F
MINIMUM 0.75 DIAM. PIPE DOWNSTREAM FROM CENT.LINE SHAFT.

P1 ABS. PRESSURE(ADJ.)FOLLOWS TIME/PRESS.TRANSIENT CURVE.
ABSOLUTE MAX.TORQUE IS DEPENDENT ON DELAY TIME AND 3.43 TO 2.15-TH POWER
OF (P1/P2)IN WORST RANGE X LINEAR CONSTANT X DWNSTR.PRESS. P6-ABS.(75-60DEG.)
IN SUBSONIC RANGE LIMITS-ONLY;SEE FORMULATIONS.-PER TESTS H.PRATT
THIS TQ. AT 72 DEG.SYMM. DISC (68=OFFSET SHAFT)CT=T/D^3/P2(ABS) ·

--5 IN.MODEL EQUIV.VALUES------ACTUAL SIZE VALUES-----

ANGLE APPRX.	P1 PSIA	P2 PSIA	DELP PSI	PRESS. RATIO	FLOW (SCFM)	FLOW (LB/MIN)	TD ----INCHLBS----	TB+TH ----INCHLBS----	TD-TB-TH	TIME(LOCA) SEC.
90	23.70	18.81	4.89	.794	645141	35465	59663	55	59608	1.00
85	27.28	19.45	7.83	.713	737320	40532	120605	111	120494	1.43
80	30.07	20.07	10.00	.667	783456	43068	171107	158	170949	1.86
75	32.51	20.25	12.26	.623	806358	44327	323497	299	323198	2.25
72	33.81	19.77	14.04	.585 CR	767504	42191	465275	430	464845	2.47
70	34.61	19.64	14.97	.567 CR	748916	41169	438358	405	437953	2.61
65	36.37	18.76	17.61	.516 CR	690090	37936	434405	401	434003	2.92
60	37.77	17.84	19.92	.472	601228	33051	336144	310	335833	3.17
55	38.78	16.89	21.89	.435	511847	28137	293011	270	292741	3.35
50	39.39	16.24	23.15	.412	421302	23160	217958	264	217693	3.46
45	39.60	15.76	23.84	.398	418401	23000	190308	295	190012	3.50
40	39.81	15.44	24.36	.388	294358	16181	142612	327	142284	3.54
35	40.42	15.12	25.30	.374	222007	12204	96397	354	96043	3.65
30	41.41	14.93	26.48	.361	168176	9245	59081	384	58696	3.83
25	42.72	14.82	27.89	.347	120845	6643	41362	433	40928	4.08
20	44.27	14.76	29.51	.333	75650	4158	31029	496	30533	4.39
15	45.99	14.71	31.28	.320	43501	2391	11965	583	11381	4.75
10	47.75	14.71	33.05	.308	22153	1217	6409	687	5721	5.14
5	49.42	14.70	34.72	.297	7269	399	4290	781	3508	5.57
0	50.70	14.70	36.00	.290	0	0	34398	732	33665	6.00

SEATING + BEARING + HUB SEAL TORQUE (M/M)=   34398 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 465275 IN-LBS @ 70 DEG.

B. Valve Stress Analysis

The Pratt butterfly valve furnished was specifically designed for the requirements of the original order which did not include specific LOCA conditions.

The valve stress analysis consists of two major sections: 1) the body analysis, and 2) all other components.

The body is analyzed per rules and equations given in paragraph NB 3545 of Section III of the ASME Boiler and Pressure Vessel Code. The other components are analyzed per a basic strength of materials type of approach. For each component of interest, tensile and shear stress levels are calculated. They are then combined using the formula:

$$S_{max} = \frac{1}{2}(T_1+T_2) + \frac{1}{2}\sqrt{(T_1+T_2)^2 + 4(S_1+S_2)^2}$$

where

S_{max} = maximum combined stress, psi

T_1 = direct tensile stress, psi

T_2 = tensile stress due to bending, psi

S_1 = direct shear stress, psi

S_2 = shear stress due to torsion, psi

The calculated maximum valve torque resulting from LOCA conditions is used in the seismic stress analysis, attachment #2, along with "G" loads per design specification. The calculated stress values are compared to code allowables if possible, or LOCA allowables of 90% of the yield strength of the material used.

C. Operator Analysis

This analysis specifically evaluates the worst case, inside containment, valve with Limitorque operator. The rating of the outside Bettis operated valve is included for informational purposes only.

Model:	Limitorque SMB 1-40/H3BC
Rating:	67800 in-lbs.
Max. Valve Torque:	465275 in-lbs.

Model:	Bettis T520-SR2
Rating:	225,000 in-lbs. (at full open and closed positions only)

The operators furnished were specifically designed for the requirements of the original order which did not include specific LOCA conditions.

The maximum torque generated during a LOCA induces reactive forces in the load carrying components of the actuator.

The Limitorque model furnished has a rating which exceeds the calculated valve torque for the following valve angles:

40 degrees open to 0 degrees (fully closed)

The Bettis model furnished has a rating which exceeds the calculated valve torque for the following valve angles:

55 degrees open to 0 degrees (fully closed)

Listed in the attachments section of this report are the following documents used in evaluating the structural and operational adequacy of the actuators.

-Operator Rating (Attachment #3)

-Supplemental Torque Calculations (Attachment #4)

IV. Conclusions

It is concluded that neither the valve structure (with present materials) nor the valve actuator are adequate to withstand the defined LOCA-induced loads based on the calculated torques developed in this analysis except for restricted valve opening as described below:

Specifically, the valve top stub shaft and top disc hub blocks are shown to be overstressed except at valve disc angles of 60° or less (see attachments 2 and 4).

In addition, the calculated torques exceed the manufacturers rating for the actuator except at valve disc angles of 40° or less (Limitorque Operator) and 55° or less (Bettis Operator). (See attachments 3 and 4)

V. Additional Information

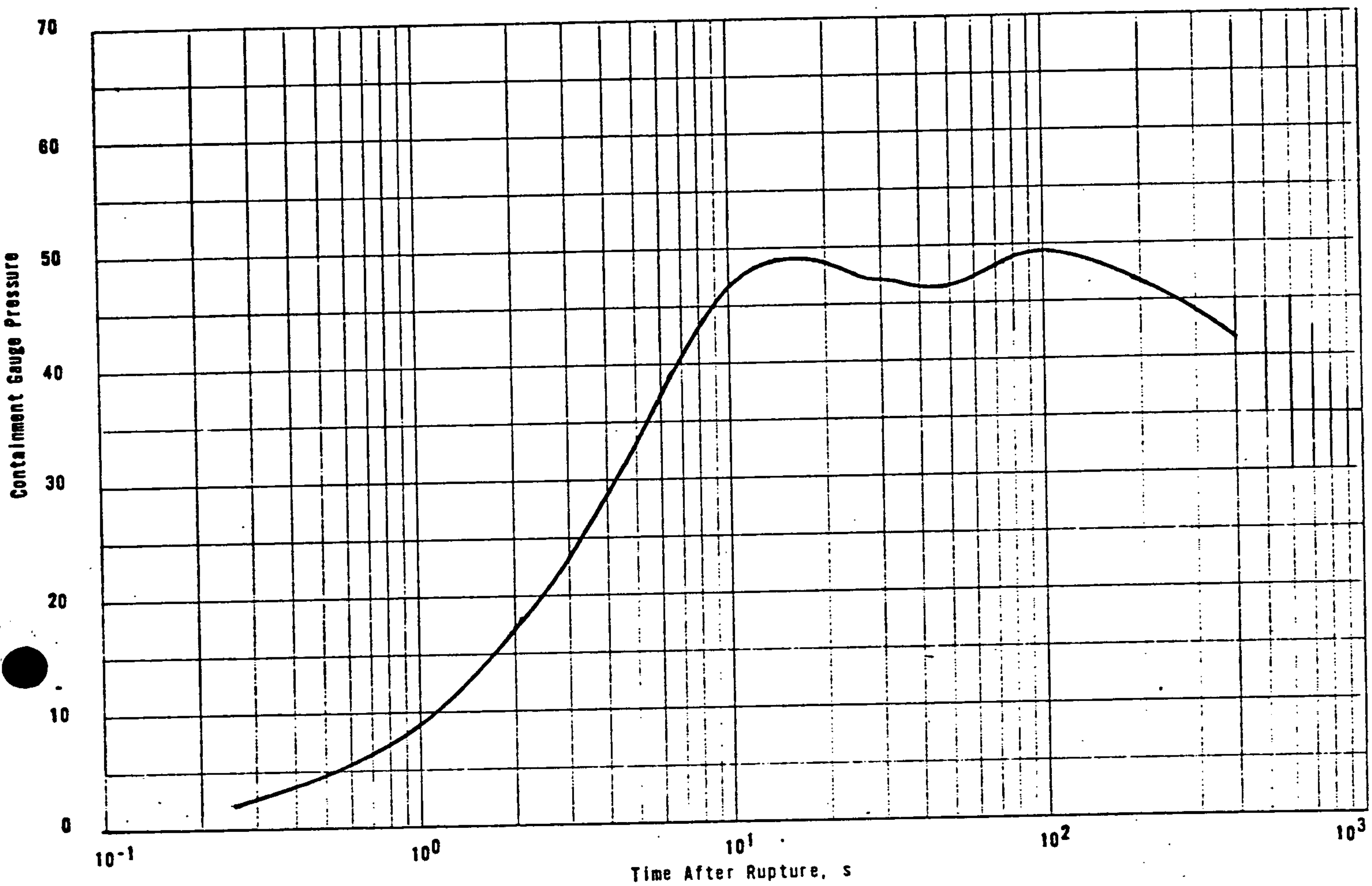
The following items are presented to describe how system factors affect torques developed in this analysis for your consideration and are informational only.

Further analysis by the customer/engineer is recommended prior to implementation.

1. An important factor governing the magnitude of the dynamic torque is delay time from the start of a LOCA incident to activation of the pressure sensing mechanism, which in turn initiates valve closure. Careful re-evaluation by the customer/engineer of the pressure sensing/timing sequence may render the present valve assembly functional through a significantly greater range of angles.
2. Installation of a convergent-divergent section downstream of the outside containment valve with a throat area sufficient to allow unrestricted ventilation during normal operation, but which will choke LOCA-induced flow while the valve is closing, through the critical range of 80°-60° open, could resultantly reduce the flow through the valve to subsonic levels.
3. An orifice plate installed similar to #2 above can also choke the system downstream and reduce flow through the valve to subsonic levels.
4. Mechanically restrict or block the valve disc to a maximum disc opening angle. (See attachment #4 for further illustration).

ATTACHMENT 1A

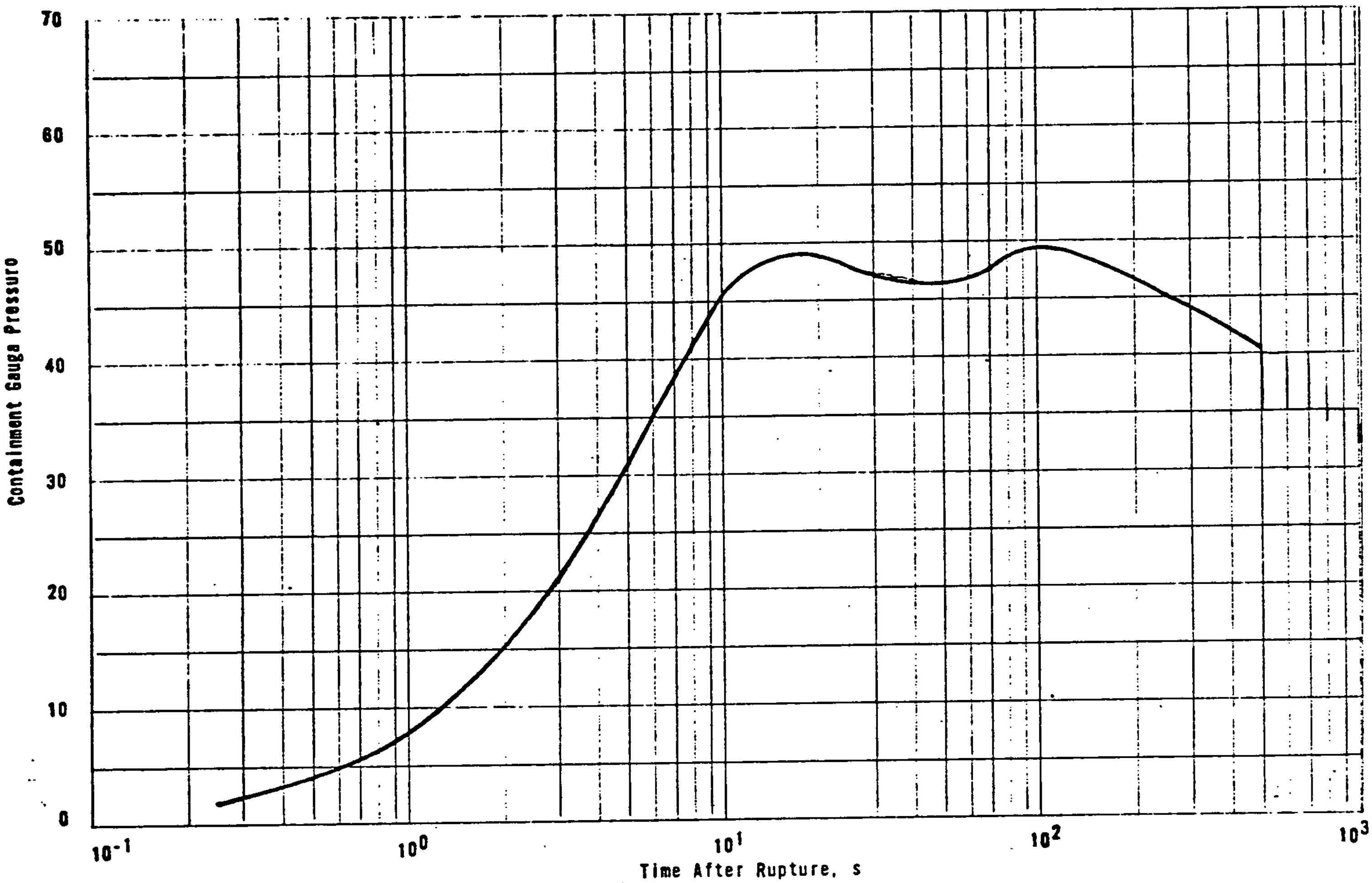
PRESSURE vs. TIME GRAPHS



REACTOR BUILDING PRESSURE
VERSUS TIME FOR 14.1 FT^2
HOT LEG BREAK

CRYSTAL RIVER UNIT 3

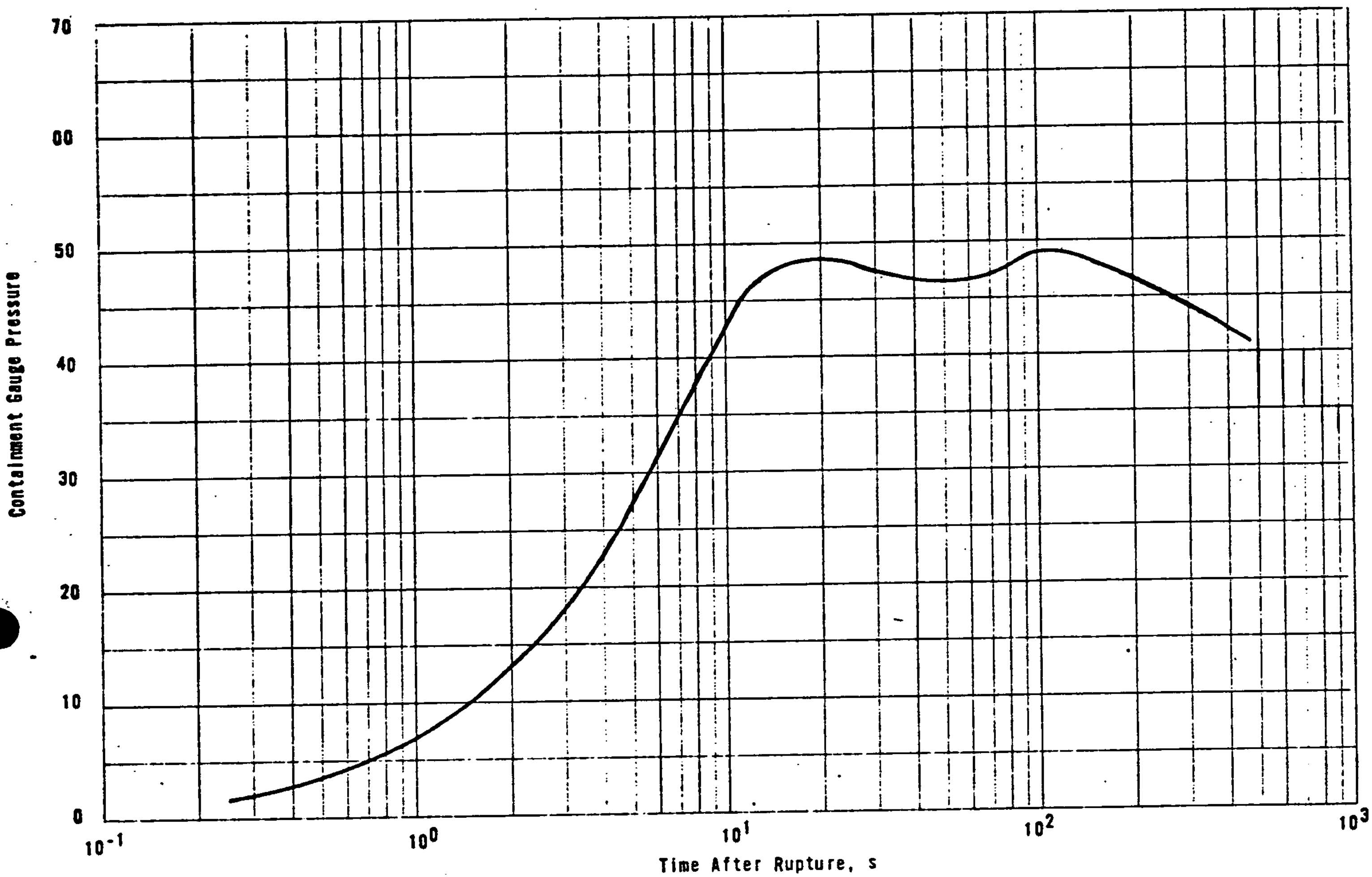
FLORIDA POWER CORPORATION FIGURE 14–72B
(AM. 27 6-29-73)



REACTOR BUILDING PRESSURE
VERSUS TIME FOR 11.0 FT2
HOT LEG BREAK

CRYSTAL RIVER UNIT 3

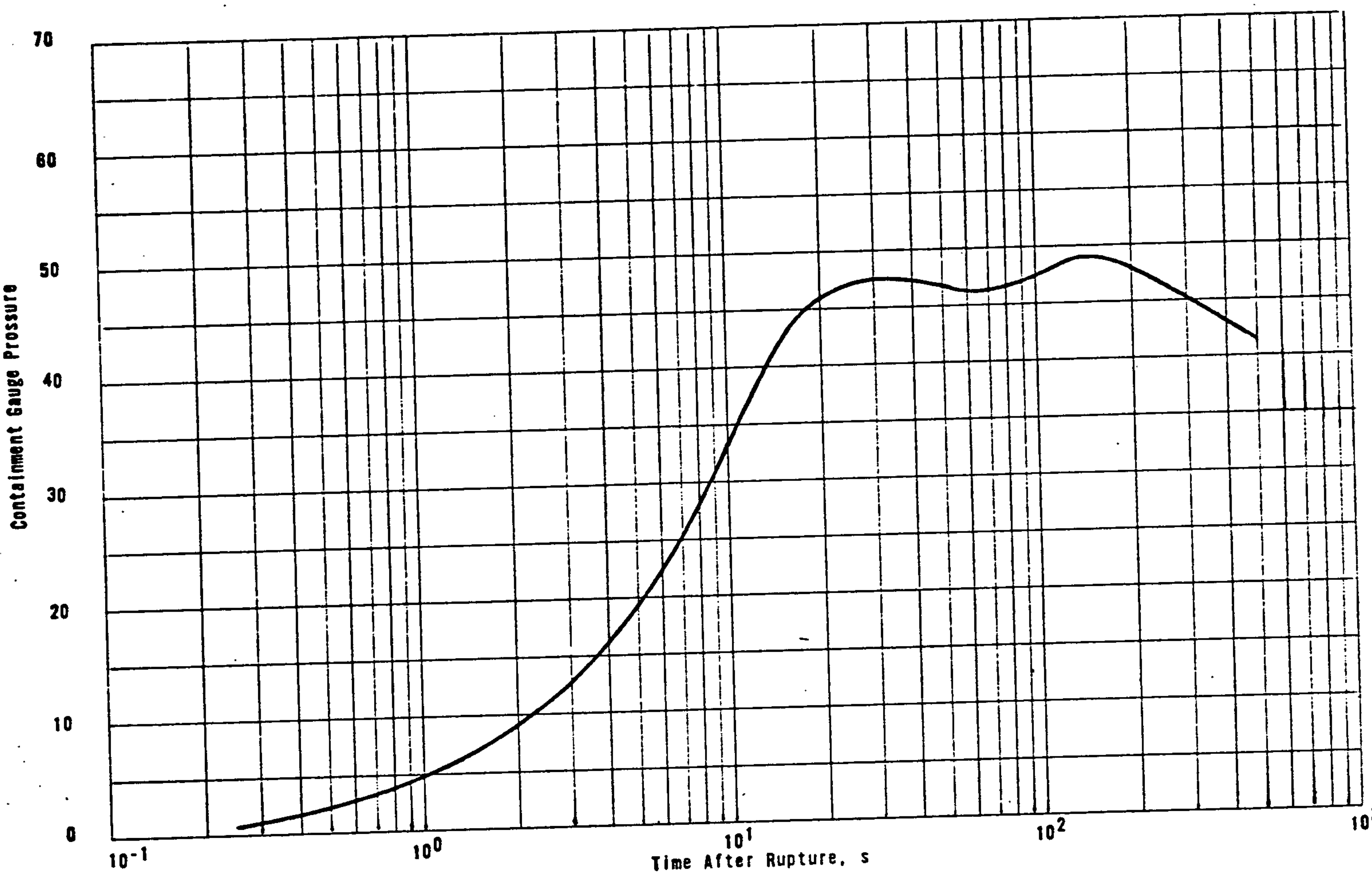
FLORIDA POWER CORPORATION FIGURE 14–72C
(AM. 27 6-29-73)



REACTOR BUILDING PRESSURE
VERSUS TIME FOR 8.55 FT^2
HOT LEG BREAK

CRYSTAL RIVER UNIT 3

FLORIDA POWER CORPORATION FIGURE 14–72D
(AM. 27 6-29-73)



REACTOR BUILDING PRESSURE
VERSUS TIME FOR 5.0 FT^2
HOT LEG BREAK

CRYSTAL RIVER UNIT 3

FLORIDA POWER CORPORATION

FIGURE 14–72E
(AM. 27 6-29-73)

ATTACHMENT 1B

PRATT LETTER REGARDING
ADDITIONAL INFORMATION

PRATT

HENRY PRATT COMPANY

creative engineering for fluid systems

401 SOUTH HIGHLAND AVENUE · AURORA, ILLINOIS 60507

December 3, 1980

Florida Power and Light Co.
P.O. Box 14042
St. Petersburg, FL 33733

Attention: Mr. K.M. Elder
Project Engineer

Subject: Crystal River - Unit #3
48" Purge Valve Analysis
A-49210Q

Dear Mr. Elder:

Recent findings in the general analysis of purge valves subjected to LOCA conditions have necessitated a request for additional technical data from the customer/engineer.

Delay time, system back pressure and valve orientation have a significant impact upon maximum torque and resultant stresses in the valve assembly. To properly complete the purge valve analysis referenced above, the following information is required:

1. The combined resistance coefficient for all ventilation system components downstream of the valve (one for each valve size) or

   A graph of back pressure vs. LOCA time at a distance 10-12 diameters downstream of the valve. Consider also the capacity of the piping, filter and duct work to resist increases in back pressure.

2. Maximum and minimum delay times from LOCA to initiation of valve rotation.

Mr. Elder
Page 2
December 3, 1980

3. Drawings or written description of valve orientation with respect to elbow immediately upstream of valve (within 6 diameters), as well as direction of valve closure (clockwise or counterclockwise) as viewed from operator end.

In the absence of the above information, the following assumptions will apply to the purge valve analysis;

1. Back pressure of 19.7 psia throughout valve closing cycle. ? Higher back pressure increases maximum dynamic torque and valve stresses.

2. Delay time from LOCA to initiation of valve rotation shall be chosen to permit initial sonic flow condition and critical valve disc angle to coincide, resulting in maximum possible dynamic torque.

3. 90° elbow immediately upstream, oriented 90° out-of-plane with respect to valve shaft, with leading edge of disc closing away from outside radius of elbow. Such orientation and closure will increase torque values by 20% or more.

Your prompt response within 30 days would be appreciated.

Very truly yours,

HENRY PRATT COMPANY

T. J. Wrona

T.J. Wrona, Manager
Contract and Proposal Engineering

/sw

CC: R.D. Nelson

ATTACHMENT 1C

CUSTOMER/ENGINEER RESPONSE
TO REQUEST FOR INFORMATION

Florida Power
CORPORATION

February 4, 1981

Henry Pratt Company
401 S. Highland Avenue
Aurora, IL 60507

Attention: Mr. T. J. Wrona

Subject: Crystal River Unit #3
48" Purge Valve Analysis (AHV-1A,B,C,D)
FPC P.O. No. A-49210Q

Reference: Henry Pratt Company ltr. Wrona to Elder dtd Dec. 3, 1980

Dear Mr. Wrona:

In accordance with your request for additional information per the referenced letter, and our telephone conversation of February 2, please find the attached drawings which depict the arrangement of ductwork and components associated with the subject valves. Hopefully this will provide enough data for Pratt to calculate the required resistance coefficients necessary for the analysis.

Also, please find the attached graphs taken from CR-3's FSAR depicting Reactor Building pressure versus time following various LOCA break sizes. Actuation of the valves will take place after the Reactor Building pressure reaches 4 psig (about 0.5 to 1 second depending upon break size.). In addition, there will be a delay of approximately 0.5 second for the ES signal to reach the actuator. (Reference B&W ltr FPC-80-016 dtd 6/30/80 and GAI ltr. FCS-1442 dtd 1/8/81.) The orientation of the valves and the direction of closure is shown on the attached Pratt drawing.

It is understood that this analysis can be completed 30 days after receipt of this information, provided it is sufficient. If further delays are anticipated or if the information which we are sending you is insufficient, please contact me immediately at (813) 866-4419. Thank you for your assistance.

Sincerely,

FLORIDA POWER CORPORATION

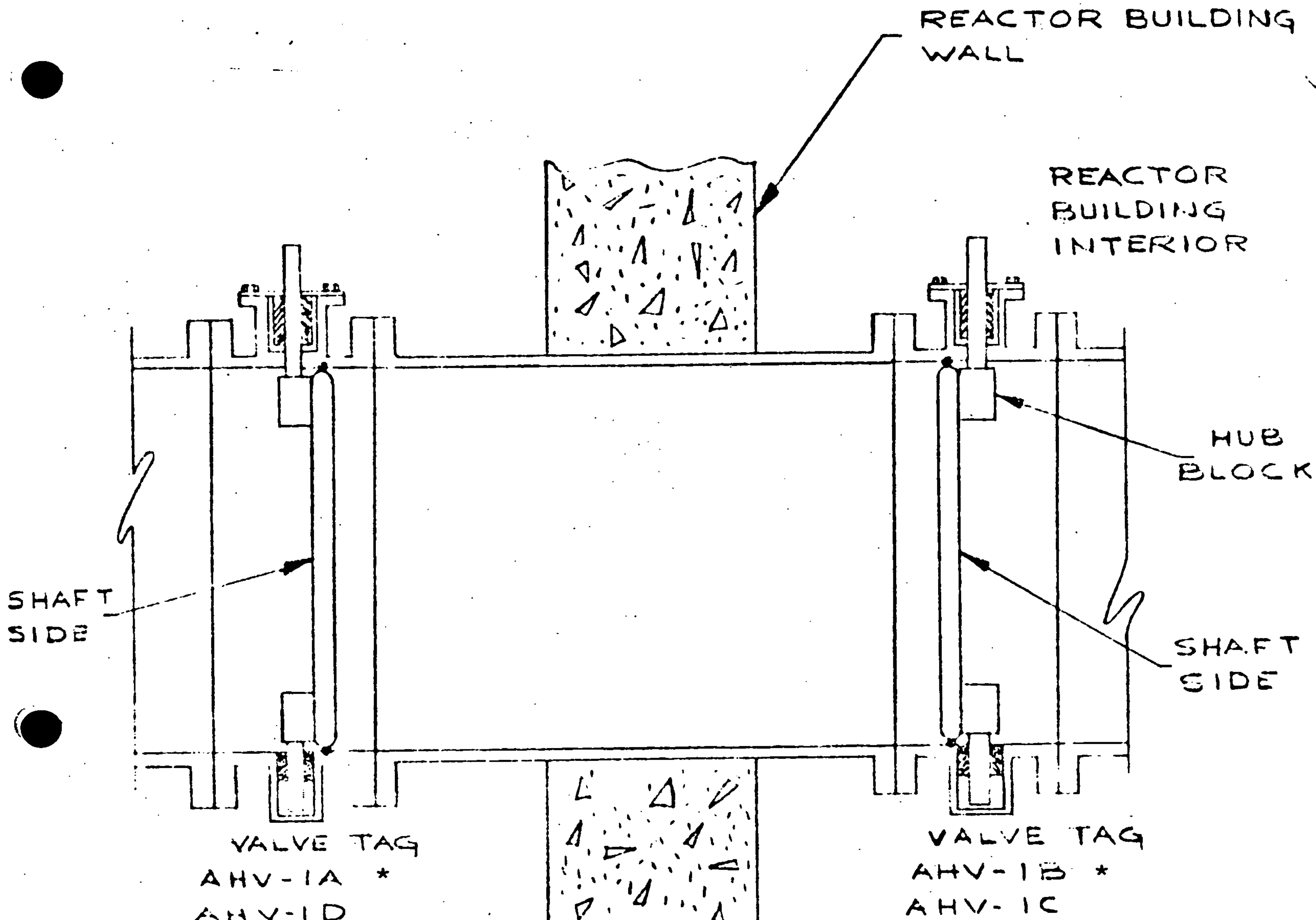
G A Becker

G.A. Becker
Supervisor, Mech./Struct. Engineering

GAB/jw

enclosure

cc: E.C. Simpson
T.C. Lutkehaus
Readers
P.Y. Baynard
F.J. Tomazic (GAI)
File: EQ 3-5-31 w/attach



(Bettis spring opposed air operated)

(Limitorque Operators)

REVISION "O"

Valve AHV-1A & 1B have operator mounted on bottom.

MACHINING TOLERANCES
UNLESS OTHERWISE SPECIFIED. FRACTIONAL DIMENSIONS ± 1/64 DECIMAL DIMENSIONS ± .005 ALL ANGLES ± 1° DIAMETERS ON COMMON CENTERLINE ARE TO BE CONCENTRIC WITHIN .010 T.I.R. MAX. SURFACE ROUGHNESS 125 R.M.S. REMOVE ALL BURRS AND BREAK ALL SHARP CORNERS

PRATT — HENRY PRATT COMPANY, AURORA ILL.

VALVE INSTALLATION (OPERATORS NOT SHOWN FOR CLARITY)

DRN BY R.C.S. CHKD BY

SCALE NONE DATE 14 JULY 72

MATERIAL SPECIFICATIONS					
3			6		
2			5		
1			4		
REV. DATE	BY	APP.	REV. DATE	BY	APP.

APPROVED

PART NO.

FIG. 1

ATTACHMENT 2

Nuclear Purge Valve

Stress Analysis

J.H. 5-71

SEISMIC ANALYSIS

for 48 inch

Nuclear Purge Valve

Prepared by L. Powell 12-30-70
Approved by [illegible]

PROJECT - Gilbert Associates - Florida Power Corp.
PRATT ORDER NO. 7-3915
CUSTOMER ORDER NO. PR3-1783 Q

VALVE SIZE 48 inch
SEISMIC ACCELERATIONS 5 g's simultaneously applied in each of three perpendicular directions.

Summarized in the following two tables are the stress intensities of primary concern. Table I identifies body stresses and how they relate to the "Draft ASME Code for Pumps and Valves for Nuclear Power" dated Nov., 1968. Table II identifies stresses in other elements of the butterfly valve assembly, for which the pump and valve code provides no specific analysis procedure. All allowable stress levels are as specified in Table A-1 of the code.

TABLE I - Body Stress Levels

Stress Name	Code Ref. Par.	Code Sym.	Analysis Ref. Pg.	Stress Level	Allowable Stress
Primary Membrane Stress Insity	452.3	Pm	5	1,025	Sm 18,900
Primary & Secondary Stresses due to flange, pressure, and seismic loads.	452.4a	Qp	5	5,177	1.5 Sm 28,350
Secondary Stresses Due to Pipe Reaction	452.4b	Ped Peb Pet	6	3,317 6,896 555	1.5 Sm 28,350
Valve Body Secondary Stresses	452.4	Sn	7	12,165	3 Sm 56,700
Fatigue Stress ($N_a \geq 2{,}000$)	452.5	Sp	7	8,967	65,000

Notes: 1. Body material is carbon steel per ASTM A-516, Gr. 60.

2. Allowable stresses are for 300°F.

3. Valve Tag No.'s are: AHV 1A
AHV 1B
AHV 1C
AHV 1D

TABLE II - Non-Codified Stress Levels

Valve Component	Stress Name	Analysis Ref. Pg.	Material	Stress Level*	Allowable Stress
Disc	Maximum Disc Stress	8	ASTM A-516 Gr. 60	8,064	Sm 18,900
Shaft	Maximum Shaft Stress	9	ASTM A-479 Type 304	38,937	.9Sy = 27,000
Shaft Retainer Assembly	Retainer Shear Stress	10	ASTM A-240 Type 304	6,500	.5 Sm 9,900
	Retainer Bearing Stress	10	ASTM A-240 Type 304	13,200	Sm 19,800
	Bolt Tensile Stress	10	ASTM A-540 CL.1,Gr.B21	38,700	Sm 46,200
	Shaft Groove Shear Stress	10	ASTM A-479 Type 304	3,400	.5 Sm 9,900
Hub Block Assembly	Keyway Bearing Stress	11	ASTM A-350 Gr.LF-1	60,870	.9 Sy = 27,000
	Max. Combined Bolt Stress	11	ASTM A-540 CL.1,Gr.B21	38,640	Sm 46,200
Thrust Bearing Assembly	T. Washer Normal Bearing Stress	13	Silicon Lub Bronze	415	* 1,200
	T. Washer Seismic Bearing Stress	13	Silicon Lub Bronze	2,075	* 8,000
	Adjusting Screw Shear Stress	13	ASTM A-479 Type 316	6,150	.5 Sm 10,000
	Adjusting Screw Tensile Stress	13	ASTM A-479 Type 316	11,200	Sm 20,000
	Retaining Screw Tensile Stress	13	ASTM A-540 CL.1,Gr.B21	21,100	Sm 46,200
	Cover Shear Stress	13	ASTM A-285 Gr.C	2,300	.5 Sm 8,850

*Not specified in pump and valve code.

Note: Allowable stresses are for 300° F.

* CALCULATION VALVE TORQUE = 435,000 IN-LBS.

HENRY PRATT CO.
AURORA, ILL.

PREPARED	DATE
CHECKED	DATE

REFERENCE NO.

FILE NO.

SH OF

NUCLEAR PURGE VALVE STRESS ANALYSIS

Prepared by J. Purcell 9-9-70
Approved by

INTRODUCTION

Described briefly in the following pages is the analysis used in verifying the structural adequacy of the main elements of the butterfly valve. Each element is described separately in its own, appropriately titled, section.

Seismic loads were made an integral part of this analysis by the inclusion of the acceleration constants g_x, g_y, g_z. Should they not be present in any of the directions of interest, simply set the appropriate value of g_i to zero.

The symbols g_x, g_y, and g_z represent accelerations in the x, y, and z directions respectively. These directions are defined with respect to the valve body centered coordinate system illustrated in the figure 1. Specifically x is along the pipe axis. z is along the shaft axis. y is perpendicular to x & y and in the direction forming a right hand triad with them.

Valve orientation with respect to gravity is taken into account by adding the appropriate quantity to the seismic loads. The justification for doing this is that a gravitational load is completely equivalent to a 1 "g" seismic load.

As an example of including gravitational loads, consider a valve oriented so that z is vertical and subjected to seismic loads G_x, G_y, and G_z. The appropriate valves for g_x, g_y, and g_z would be:

$$g_x = G_x$$

$$g_y = G_y$$

$$g_z = 1 + G_z$$

Throughout the analysis, reference is made to a "banjo" assembly. This is the assembly consisting of the disc, the stub shafts, the hub blocks, and the mounting hardware. It is termed a "banjo" assembly simply because it resembles a banjo in appearance, and this is an easy way to refer to it. The main elements of the banjo assembly are identified in figure 2.

HENRY PRATT CO.
AURORA, ILL.

PREPARED DATE

CHECKED DATE

REFERENCE NO

FILE NO.

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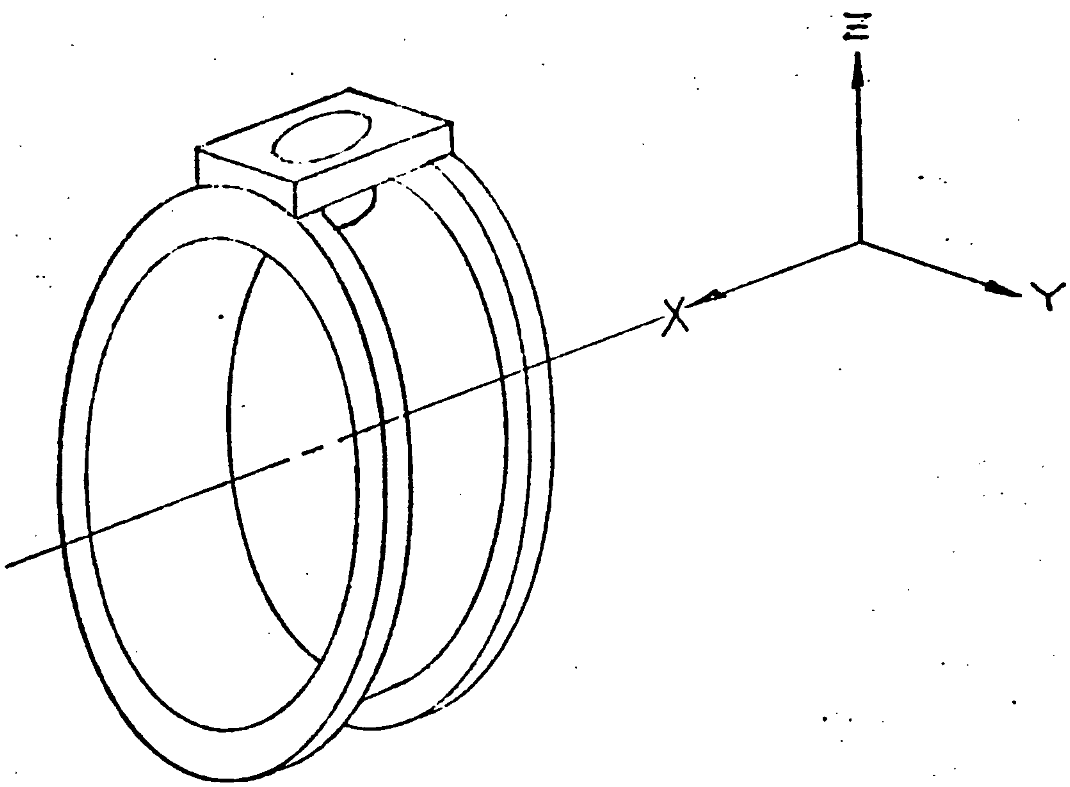
FIGURE 1 - VALVE BODY CENTERED CO-ORDINATE SYSTEM FOR DEFINING ACCELERATION DIRECTIONS



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PREPARED	DATE
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Figure 2 - Essential Features Of Banjo Assembly

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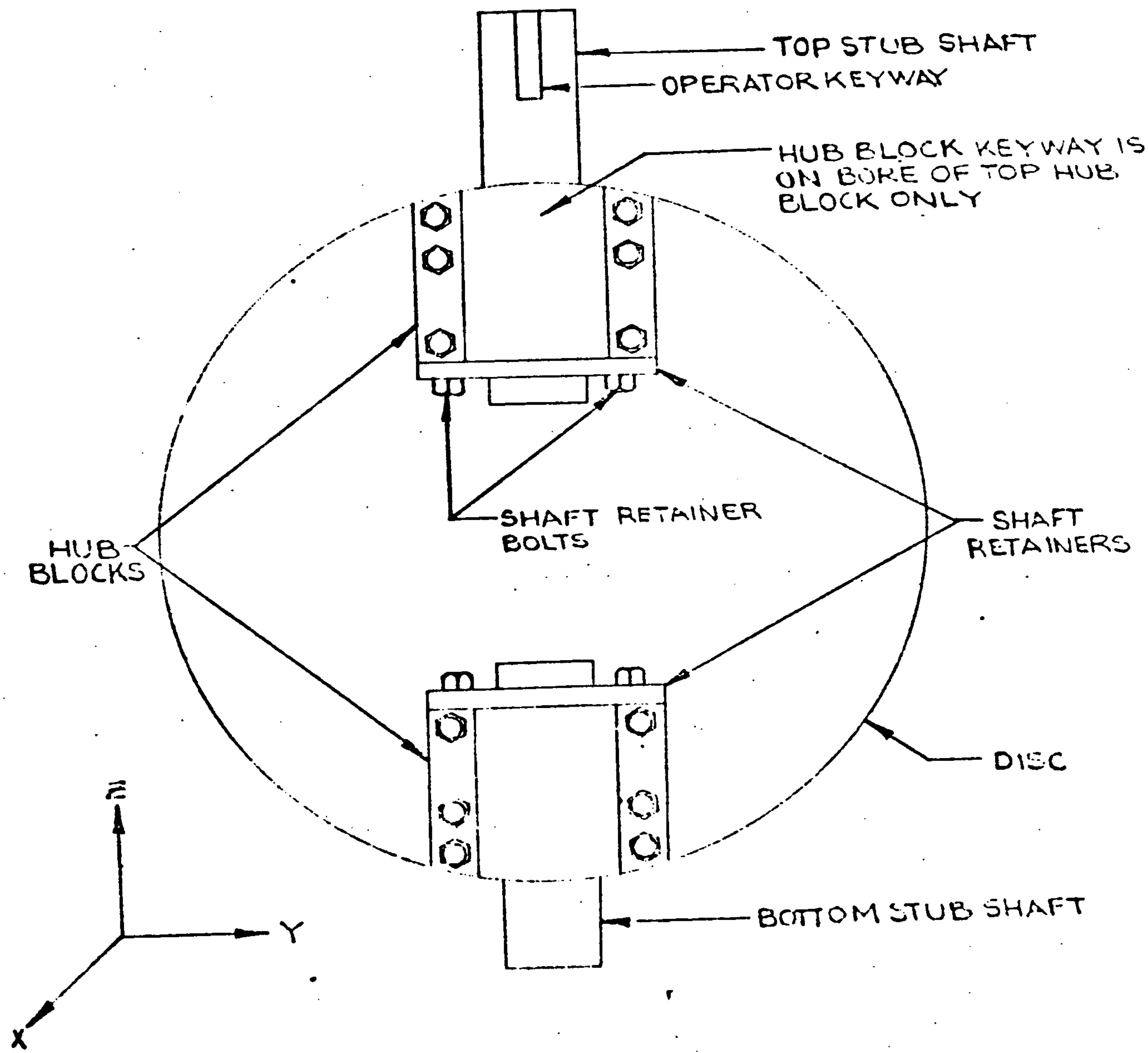

Prepared by [illegible]
Approved by [illegible]

FLANGE ANALYSIS

The flange analysis is in accordance with Appendix II, Para. VA-56 of Section VIII, Division I, of the ASME Codes for Pressure Vessels and AWWA C-207.

Approved by

BODY ANALYSIS

With one exception, the body analysis is in accordance with "Draft ASME Code for Pumps and Valves for Nuclear Power" dated Nov., 1968. This exception is in the calculation of valve body primary plus secondary stress due to internal pressure, a quantity labeled as Q_p in section 452.4a of the code. The formula which is specified in this section and considers only stresses induced by internal pressure is not used. In its place has been substituted a more complete formulation which considers stresses induced by internal pressure, flange moments, and seismic loads. All other body stress calculations are exactly per the pump and valve code.

The specific formulas used in calculating the body stresses are listed below.

1. Primary membrane stress - The following formula which satisfies the intent of section 452.36 of the code was used.

$$Pm = (Rm/h + 1/2)\ p$$

where: Rm = shell mean radius-inches
p = internal pressure-psig
h = shell thickness-inches

2. Valve body primary plus secondary stresses due to internal pressure, flange moments, and inertial loads - This is the quantity which replaces Qp as defined in section 452.4a of the code. It is calculated for two sections on the valve body, the section where the flange joins the body, and the section defined by the centerline of the valve shaft. The largest of these two valves is then taken as Qp. The formula used for calculating Qp is:

$$Qp = 1/2\ P + 1/2\ (Q_{p1} + Q_{p2}) + 1/2\ \sqrt{(Q_{p1} - Q_{p2})^2 + 4\ y^2}$$

where: Y = sum of shear stresses due to inertia torques and inertia transverse shear.-psi
Q_{p1} = axial stresses-psi
Q_{p2} = circumferential stresses-psi
P = internal pressure-psig

The quantities, Y, Q_{p1}, and Q_{p2} are calculated from the following formulas:

$$Y = \frac{2WRo}{\pi(Ro^4 - Ri^4)} \left[E_c\ g_x + L\ (g_y^2 + g_z^2)^{1/2}\right]$$

$$Q_{p1} = PRm/2h + 6M/h^2 + \frac{W}{\pi}\left[\frac{RoL\ (g_y^2 + g_z^2)^{1/2}}{Ro^4 - Ri^4} + \frac{g_x}{2Rmh}\right]$$

$$Q_{p2} = PRm/h + 6\nu M/h^2 - wE/Rm$$

where: P = internal pressure-psig
W = valve weight-pounds
Ro = outside radius of valve body-inches
Ri = inside radius of valve body-inches
L = valve length-inches
E_c = valve body eccentivity-inches
Rm = mean radius of valve body-inches
h = valve body thickness-inches
E = young's modulus-psi
ν = poisson's ratio
g_x, g_y, g_z = acceleration constants
w = deflection of valve body-inches
M = local bending moment per unit circumference-pounds
Note: W and M are calculated in a separate analysis, the details of which are not included here.

3. Secondary stresses due to pipe reaction-These are calculated using the equations of section 452.46 of the Pump and Valve code. More specifically, these are:

$$Ped = \frac{F_d S}{G_d}$$

$$Peb = \frac{C_b F_b S}{G_b}$$

$$Pet = \frac{2 F_b S}{G_t}$$

where: Ped = direct, or axial, load effect-psi
Peb = bending load effect-psi
Pet = torsional load effect-psi
F_b = bending modulus of standard connected pipe per figures 452.4b of pump and valve code-inches
F_d = 1/2 the cross sectional area of standard connected pipe-inches
C_b = stress index for body bending secondary stress per section 452.4b
S = 30,000 per section 452.4b
G_d = valve body section area-inches^2
G_t = valve body section torsional modulus-inches^3
G_b = valve body section bending modulus-inches^3

4. Thermal Secondary Stress-This stress is calculated per section 452.4c of the Pump and Valve code. More specifically, the formulas used were:

$QT = 17.5\ h^2$ for austenetic steel

$QT = 50.0\ h^2$ for ferritic steel

where: QT = thermal secondary stress
h = thickness of valve body

5. Combined Stress Intensity-This quantity, as specified in section 452.4 of the Pump and Valve code is given by the formula:

$$Sn = Qp + Pe + 2Q_T$$

where: Sn = combined stress intensity
Qp is given under number 2, above.
Q_T is given under number 4, above.
Pe is the largest of Ped, Peb, Pet as given in number 3, above.

6. Fatigue Stresses-The value taken for comparison with figures 452.5 (a) and 452.5 (b) of the Pump and Valve code is the larger of the following, as given in section 452.5:

$$S_{p1} = 2Qp/3 + Peb/2 + 1.3\ Q_T$$

$$S_{p2} = .4Q_p + Peb$$

where all terms are as previously defined

Prepared by [signature]

Approved by [signature] 7/1/70

DISC ANALYSIS

For an air purge valve, the worst load combination which occurs is combined pressure plus seismic loads. The highest magnitude stresses are present at the center of the disc and can be considered as being the result of simultaneous bending about two perpendicular axes, the y axis and the z axis. The magnitude of the stress is given by:

$$\sigma = \frac{(P + P_e)\, d}{(t)^2} \left[36\,(.125\,\pi a + .113d)^2 + \frac{d^2}{4} \right]^{1/2}$$

Where:
- Pe = equivalent seismic pressure = wtgx - psi
- w = weight density of disc - Pd/in^3
- t = thickness of disc - inches
- gx = acceleration constant
- P = applied pressure - psig
- d = diameter of disc - inches
- a = unsupported shaft length - inches

It usually occurs that disc thickness is dictated by deflection requirements and that disc stresses are well below code allowable levels.

Prepared by [illegible] 1-7-76
Approved by [illegible]

SHAFT ANALYSIS

Because of the manner in which the purge valve is used, fluid dynamic loadings can be neglected. Therefore, the worst loading condition on the shaft will be either a combination of torsional plus seismic loads or a combination of pressure plus seismic loads. Both of these conditions were checked using the formulas listed below. Columnar tensile and compressive loads on the shaft were not considered because of their obviously small effect on stress levels.

1. Shaft Stress due to torsion plus seismic loads.

$$\sigma = \frac{1}{2}\sigma_B + \frac{1}{2}\left[\sigma_B^2 + 4\sigma_T^2\right]^{1/2}$$

where: σ_B = bending stress = $\dfrac{16W(g_x^2 + g_y^2)^{1/2}a}{\pi d^3}$

σ_T = torsional stress = $\dfrac{16SD^2}{\pi d^3}$

W = weight of banjo assy. - Pds.
a = unsupported shaft length - inches
D = disc diameter - inches
S = seating factor - Pds/inch
d = shaft diameter - inches
g_x, g_y = acceleration constants

2. Shaft Stresses due to pressure plus seismic loads. - Both shear and bending stresses are calculated. However, they are not combined since their maxima occur at different locations on the cross section.

$$\sigma_S = \frac{2}{3A}\left[(\pi D^2 P/4 + Wg_x)^2 + (Wg_y)^2\right]^{1/2}$$

$$\sigma_B = \left[\sigma_x^2 + \sigma_y^2\right]^{1/2}$$

where: $\sigma_x = \dfrac{32(.125\,\pi D^2 P + .5\,Wg_x)a}{\pi d^3}$

$\sigma_y = \dfrac{16Wg_y a}{\pi d^3}$

A = cross sectional area of shaft - in^2
P = applied pressure - psig
D = disc diameter - inches
d = shaft diameter - inches
W = weight of banjo - pounds
a = unsupported shaft length - inches
g_x, g_y = acceleration constants

Approved by

SHAFT RETAINER ASSEMBLY

For purposes of convenience in description, the shaft retainer assembly is considered to consist of the shaft retainer, the shaft retainer bolts, and the grooved end of the stub shaft. The shaft retainer was checked for shear tear out and bearing stresses. The shaft retainer bolts were checked for tensile stresses assuming all four retainer bolts to be equally loaded. The grooved end of the shaft was checked for shear tear out and bearing stress. Formulas for calculating each of these stresses are listed below.

1. Shear stress in retainer

$$\sigma_{Sr} = \frac{2Wg_z}{\pi dt}$$

2. Bearing stress on retainer and groove

$$\sigma_B = \frac{8Wg_z}{\pi(d^2 - dr^2)}$$

3. Tensile stress in retainer bolts

$$\sigma_T = \frac{Wg_z}{4A}$$

4. Shear tear out of shaft groove

$$\sigma_{ss} = \frac{2Wg_z}{\pi d_r L}$$

where: W = weight of banjo - pounds
d = shaft diameter - inches
dr = diameter of retainer bore - inches
t = shaft retainer thickness - inches
A = tensile area of retainer bolts - in^2
L = length of shaft after groove - inches
g_z = acceleration constant

HUB BLOCK ASSEMBLY

The hub block assembly is considered to consist of the hub block, the hub block retaining bolts, and the hub block keyway. The two stresses of primary concern in the hub block assembly are the keyway stresses and the combined tensile plus shear stresses in the hub block bolts. The analysis of each of these is explained below.

1. Hub Block Keyway - The hub block keyway can be safely designed by keeping the compressive bearing stress on the keyway face below the allowable stress level for the hub block material. The bearing stress is calculated using the following formula:

$$\sigma_B = \frac{4}{dKL} \times T_{MAX}$$

where:

d = shaft diameter - inches
K = key height - inches
L = key length - inches
T_{MAX} = MAX. DYNAMIC TORQUE (~435000 IN#)

2. Hub Block Bolt Stress - The hub block bolts are sized and located such that the maximum combined shear plus tensile stress does not exceed the code allowable value for the bolting material. Stresses are combined in accordance with the formula:

$$\sigma = \left[\sigma_t^2 + 4\sigma_s^2\right]^{1/2}$$

where: σ = combined stress level
σ_t = tensile stress
σ_s = shear stress

The value for σ_s is obtained by evaluating the following formula:

$$\sigma_s = \frac{Wg_z}{3A}$$

where: W = banjo weight - pounds
A = tensile area of bolt - in^2
g_z = acceleration constant

The value for σ_t is obtained by evaluating the formula given below. This formula is the result of an analysis which considers the effect or pressure plus seismic loads in the x direction, moment from these loads in the x direction resulting from unsupported shaft length, and moment and wedging effects from loads in the Y direction.

$$\sigma_t = \frac{1}{A_t}\left[\frac{77D^2P}{8} + \frac{Wg_x}{2}\right]\left[\frac{1}{6} + \frac{C\ (A+E)}{2(A^2+B^2+C^2)}\right] + \frac{Wg_y}{12A_t}\left[\frac{a}{\sqrt{5}} + \frac{d}{3H}\right]$$

where: A_t = bolt tensile area - in^2
D = disc diameter - inches
P = applied pressure - psig
W = banjo weight - pounds
g_x, g_y = acceleration constants
A = distance from hub edge to first bolt pain - inches
B = distance from hub edge to second bolt pain - inches
C = distance from hub edge to third bolt pain - inches
d = shaft diameter - inches
a - unsupported shaft length - inches
H = distance between bolt rows - inches

THRUST BEARING ASSEMBLY

The thrust bearing assembly provides restraint in the z direction for the banjo assembly, thus assuring the disc edge remains correctly positioned to maintain sealing capability. Structural adequacy of the assembly was checked using the six formulas listed below. Specific elements of the thrust bearing as referred to below are identified in figure 3.

1. Normal bearing stress on thrust washer.

$$\sigma_{bn} = \frac{W}{A_1}$$

2. Seismic bearing stress on thrust washer.

$$\sigma_{bs} = \frac{g_z W}{A_1}$$

3. Shear stress in adjusting screw head.

$$\sigma_{sn} = \frac{g_z W}{77 D t}$$

4. Tensile stress in adjusting screw.

$$\sigma_{ta} = \frac{g_z W}{A_2}$$

5. Shear stresses in cover.

$$\sigma_{sc} = \frac{g_z W}{.9 \; 77 D T}$$

6. Tensile stress in retaining screws.

$$\sigma_{tr} = \frac{g_z W}{4 A_3}$$

where: W = banjo weight - pounds
A_1 = bearing area of thrust washer - in^2
g_z = acceleration constant
D = diameter of adjusting screw - inches
t = thickness of adjusting screw head - inches
A_2 = tensile area of adjusting screw - in.^{2}
T = cover thickness - inches
A_3 = tensile area of retaining screws - in.^{2}

HENRY PRATT CO. AURORA, ILL.	Figure 3 - Essential Features Of Thrust Bearing Assembly	REFERENCE NO.
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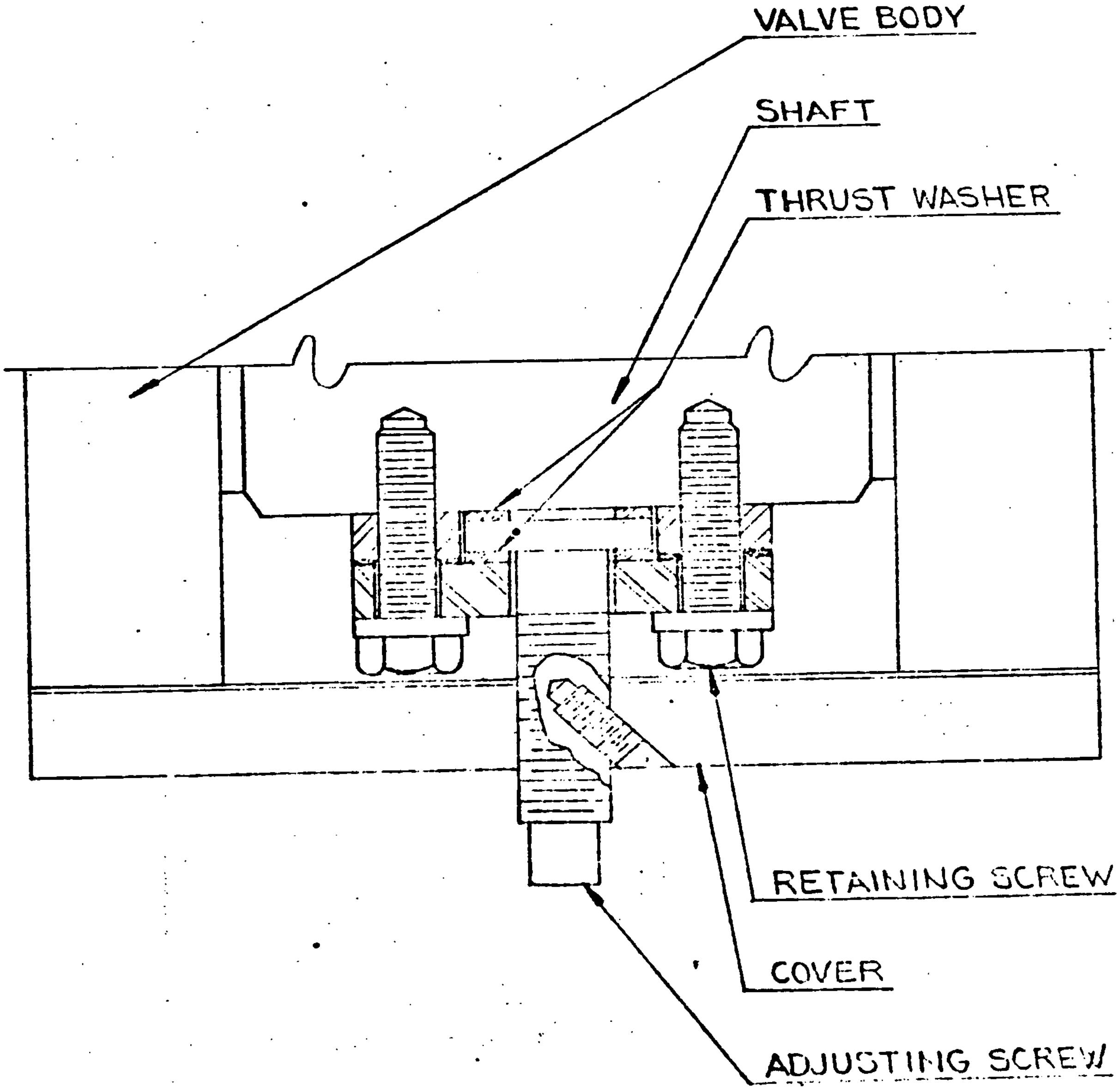
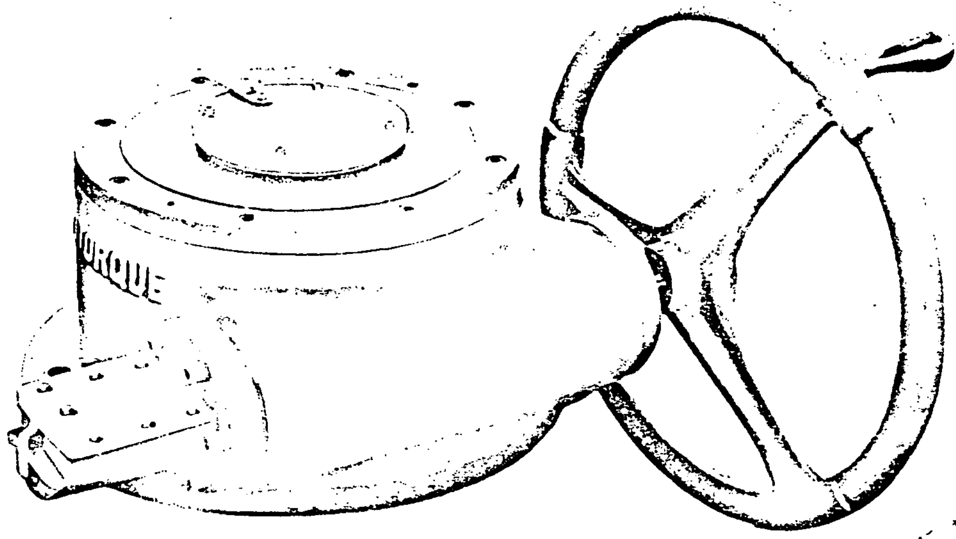


ATTACHMENT 3

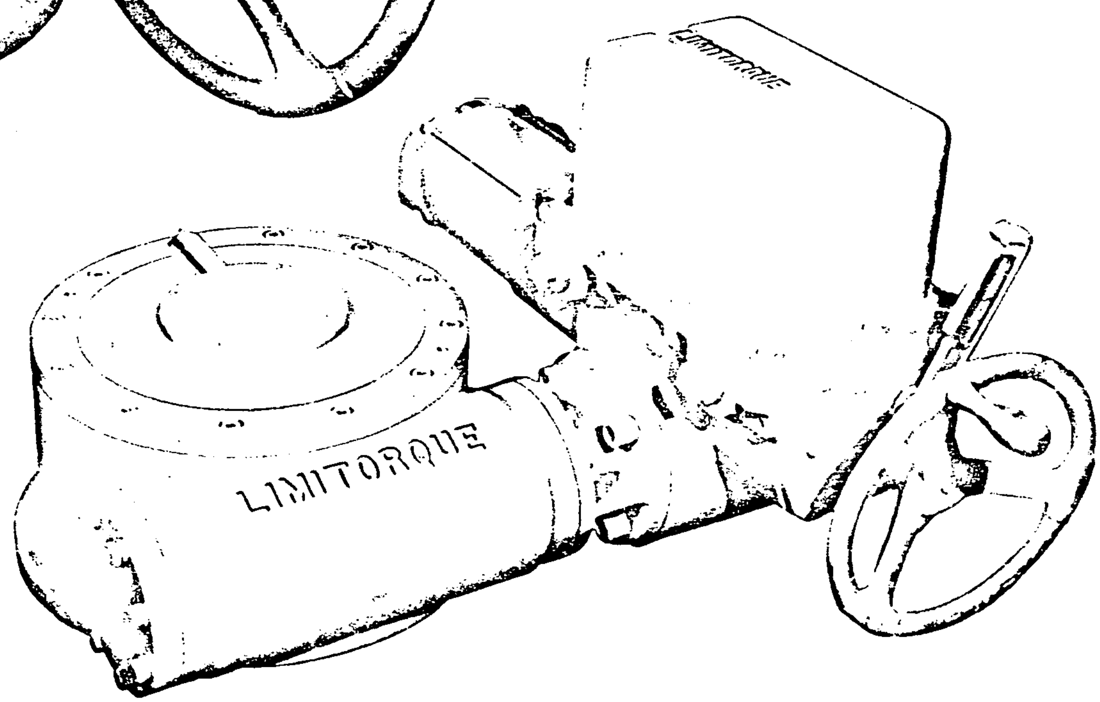
OPERATOR RATINGS

LIMITORQUE VALVE CONTROLS

MANUAL TYPE HBC



Butterfly Valve manual operator size H3BC.



Motorized Limitorque Valve Control type SMB with H4BC manual.

The hand operated type H-BC unit is a worm gear drive which may be used for any valve or device requiring a 90° movement. The H-BC manual gear operator is especially designed for operation of butterfly, plug and ball valves. Every H-BC operator has an adjustable mechanical stop limit device to prevent movement of the valve beyond 90° of travel. Instructions for setting these limit stops are described elsewhere in this bulletin.

The manual H-BC operator has an alloy steel worm shaft and a bronze worm gear. On all units, except for buried service, a valve position pointer is furnished as a standard part of the operator. On buried and submersible units, stainless steel non-corrosive input shafts are furnished.

Handwheels are optional and can be furnished in various sizes as an extra.

All units are built to meet the requirements of A.W.W.A. specifications and when spur gear or bevel gear attachments are used, the maximum input torque is less than 80 ft.-pounds to develop the maximum output torque rating of the unit with standard or optional gear ratios.

All sizes of units can be furnished with Limitorque valve controls or can be readily converted for motor operation in the field using Limitorque valve controls. The speed of operation of butterfly, plug or ball valves, when motor operated, is usually 20 to 30 seconds, however this can be varied over a wide range limited only by motor speed and available gear ratios.

INFORMATION NEEDED FOR ORDER

To size a manual operator, we need:

1. — Torque at valve shaft.
2. — Valve shaft and keyway size.
3. — Degrees of travel.
4. — Type of enclosure, weatherproof, buried or submersible. (If submersible, describe depth and time)
5. — Position of assembly.

For motor operation, in addition to the above, we need:

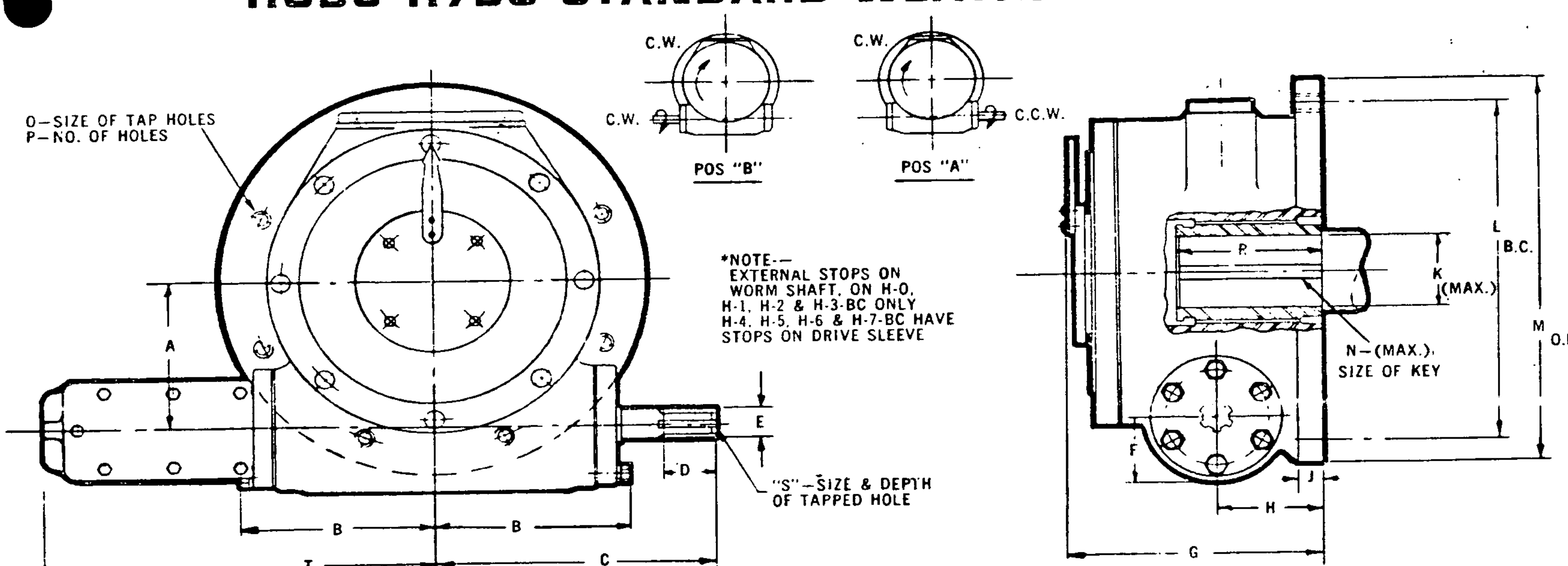
6. — Operating times.
7. — Voltage, phase and cycles (or DC volts).
8. — Type and frequency of service.
9. — Maximum ambient temperature.
10. — Class desired, weatherproof, explosion-proof, or submersible.
11. — Type of motor starter enclosure.
12. — Type of pushbutton station enclosure.

SELECTION CHART FOR MANUAL OPERATORS

UNIT SIZE	OUTPUT TORQUE RATING INCH POUNDS	FT. POUNDS	WORM GEAR RATIO *	SPUR OR BEVEL GEAR ATTACHMENT RATIO *	TOTAL H.W. TURNS FOR 90° WITH ATTACH. SPUR OR BEVEL
H0BC	5,340	445	71:1	1:1 (bevel only)	17.7
H1BC	15,600	1,300	70:1	2.86:1	50
H2BC	26,400	2,200	70:1	2.86:1	50
H3BC	67,800	5,650	70:1	2.86:1	50
H4BC	153,600	12,800	60:1	12.0:1	180
H5BC	235,000	19,583	65:1	12.0:1	195
H6BC	552,000	46,000	66:1	38.9:1	641.8
H7BC	760,000	63,333	69:1	38.9:1	671

*ALTERNATE OPTIONAL RATIOS AVAILABLE ON REQUEST.

H0BC-H7BC STANDARD WEATHERPROOF UNIT



02-440-0167-2

FOR INSTALLATION PURPOSES USE CERTIFIED DIMENSIONS ONLY.

UNIT SIZE	A	B	C	D	E	F	G	H	J	K	L	M	N	O	P	R	INPUT SHAFT SPLINES	S	T
H-0-BC	2½	4⅝	7 1/16	1⅝	1	2 1/16	6 13/16	3	¾	1½	8¼	9½	½x¼x3⅛	½-13	8	3⅛	15 T. INV. SPL. 16/32 D.P.	⅜-16x¾	9 15/16
H-1-BC	3½	5 7/16	8 9/16	2	1¼	2 21/32	8 5/16	3½	¾	1½	10	11¾	⅝x⅝x4⅝	⅝-11	8	4¾	6 SPLINES 1.050 R.D. - .3125 WIDE 1.250 - 1.249 O.D.	⅝-16x⅝	11⅞
H-2-BC	4¼	5 15/16	9¼	2	1¼	2 11/16	8 7/16	3½	1"	2¼	11¾	13½	¾x½x5¼	¾-10	8	5½		⅝-16x½	12⅜
H-3-BC	6	7 5/16	10⅞	2	1¼	2 11/16	9 5/16	4½	1"	3¾	14	16	¾x½x6	¾-10	8	6		⅝-16x⅝	13¾
H-4-BC	7¾	9 5/16	13¼	3	1¾	3⅞	10 13/16	4¼	1"	4¼	16	18½	1"x¾x7¾	¾-10	8	7¾	6 SPLINES 1.525 R.D. - .435 WIDE 1.749 - 1.748 O.D.	⅜-16x¾	
H-5-BC	9¼	10 3/16	14¼	3	1¾	3½	11 9/16	5	1"	6½	18½	21	1½x1x8¼	¾-10	8	8¼		⅜-16x¾	
H-6-BC	13	13⅝	18¾	4	2.415	4½	13 13/16	6½	1¼	7½	23	26¼	1¾x1¼x10½	1¼-7	8	10½	28 INVOLUTE SPLINES 12/24 D.P. 2.333 P.D.	½-13x¾	
H-7-BC	16	15⅛	19¼	4	2¼	5½	14½	6¼	2½	7½	29	31⅛	1¾x1¼x8⅛	1¼-7	8	8⅛	K.W. ⅜x5/16x4 LG.	½-13x1	

NOTE: FOR SIZE H-7-BC WITHOUT SPLINED ADAPTER MAXIMUM BORE IS 8¼" WITH 2"x1½" KEY.

A Galveston-Houston Company

7031 Grand Blvd.
P.O. Box 14689
Houston, Texas 77021
(713) 748-1143 Telex 76-2713

ATTACHMENT 3

January 15, 1981

Henry Pratt Company
401 South Highland Ave.
Aurora, Illinois 60507

Attention: Mr. Ted Wrona

Subject: T-5 Actuator Yoke Assembly Torque Absorbing Capabilities

Dear Mr. Wrona:

This is in response to our telephone conversation of January 12, 1981 concerning the torque absorbing capabilities of T-5 actuators; specifically a Model T516B-SR3.

Attached is a typical set of data for a Model T-520B double acting actuator. Please note that the yoke assembly mechanism for both double acting and spring return actuators is identical. Consequently, the torque absorbing capabiltiy of a spring return actuator is the same as a double acting unit (i.e., 225,000 lb-in at either the full open or full closed (0-90°) positions). From the graph or tabulated data the percentage of torque outputs at 15° and 75° positions with respect to 0° and 90° torques is 74.5 and 72.6 percent each, respectively. Based on this, the yoke assembly (rated at 225,000 lb-in) should be capable of absorbing at least 163,350 lb-in at the 75° position.

T520B

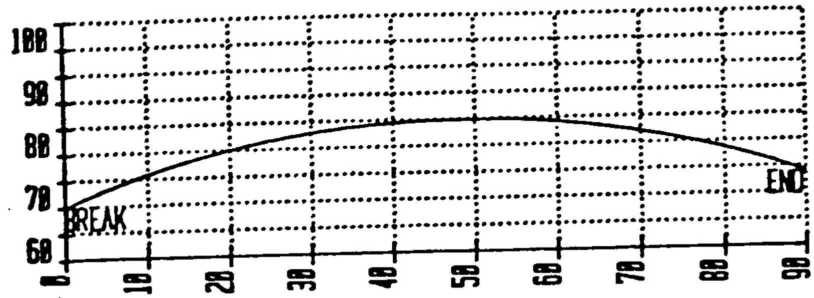
DATA INPUT

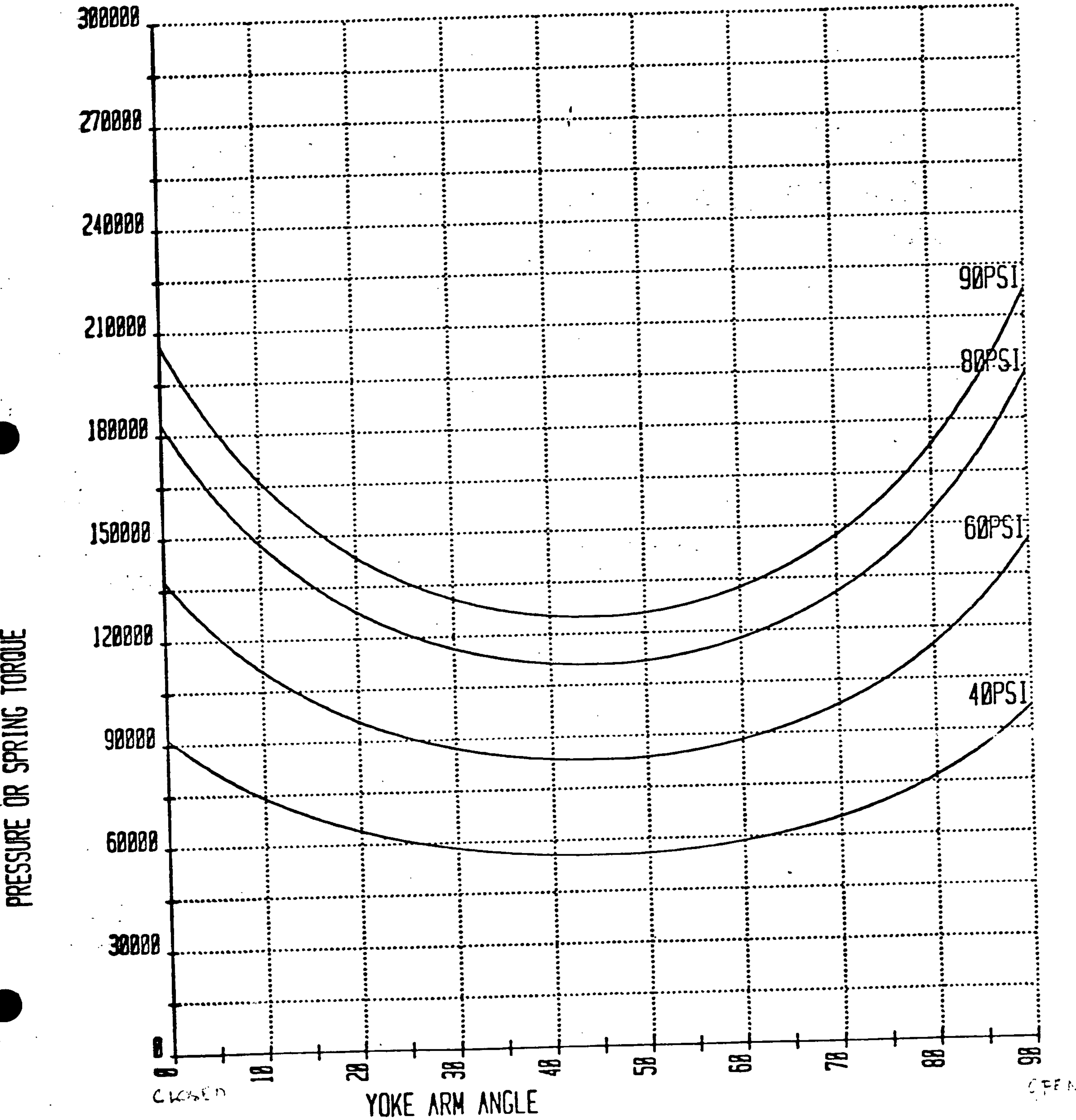
CYLINDER DIAMETER (in)=				19.58
CENTER OR TIE BAR DIAMETER (in)=				1.000
PISTON ROD DIAMETER (in)=				1.750
NUMBER OF PISTONS =				1
MOMENT ARM (in)=				5.500
SPRING LOAD A (lbs)=				0
SPRING LOAD B (lbs)=				0
BREAK EFFICIENCY (%)=				70
RUNNING EFFICIENCY (%) =				85
ENDING EFFICIENCY (%) =				74
PRESSURES (psi) =	40	60	80	90
ACTUATOR TYPE,CB=1,HD=2,T,TR=3, =				3

YOKE ARM ANGLE (degrees)	SPRING TORQUE (in lb)	PRESSURE TORQUE (40)psi	PRESSURE TORQUE (60)psi	PRESSURE TORQUE (80)psi	PRESSURE TORQUE (90)psi	EFFICIENCY SPR. %	EFFICIENCY PRES. %
0	0	91515	137273	183030	205909	74	70
5	0	81533	122299	163066	183449	77	73
10	0	73965	110947	147929	166420	79	76
15	0	68195	102292	136389	153438	81	78
20	0	63811	95716	127621	143574	82	80
25	0	60533	90799	121065	136199	83	82
30	0	58170	87255	116340	130883	84	83
35	0	56595	84892	113190	127339	85	84
40	0	55727	83590	111454	125385	85	85
45	0	55523	83285	111047	124928	85	85
50	0	55975	83963	111951	125945	85	85
55	0	57106	85660	114213	128489	84	85
60	0	58975	88462	117949	132693	83	84
65	0	61681	92521	123361	138781	82	83
70	0	65379	98069	130759	147103	80	82
75	0	70301	105451	140602	158177	78	81
80	0	76785	115177	153570	172766	76	79
85	0	85340	128010	170680	192015	73	77
90	0	96745	145117	193489	217675	70	74

EFFICIENCY PLOT

EFFICIENCY vs ANGLE





ATTACHMENT 4

SUPPLEMENTAL TORQUE CALCULATIONS

ATTACHMENT 4

The following pages illustrate the combined effects of disc blockage and delay time on dynamic torque. In each case, the delay time is fixed at that which produced the worst case torque for the full open, unblocked condition. The initial disc angle is reduced by blocking to illustrate the resultants of several different initial angles of opening.

D-29254-1

JOB:FLOR.PWR;CRYST.RIV P2-VARIABLE SIZE ADJUSTED(REYNLDS NO.FNCTN!)
SAT.STEAM/AIR MIXTURE WITH 1.4 LBS STEAM PER 1-LBS AIR
SPEC.GR.= .738255   MOL.WT.= 21.3872   KAPA(ISENT.EXP.)= 1.19775   R= 72.1972
GAS CONSTANT-CALC.
SONIC SPEED(MOVING MIXTR.)= 1316.65   FEET/SEC AT 225   DEG.

CRIT.CASE INLET VELOCITY IS 1.46694   TIMES HIGHER AS AIR CRIT.CASE INLET V1-OF
5 INCH MODEL

MAX. TORQUE IS AT THE CRITICAL PRESS.RATIO(.585-(5 IN)MODEL OR APPX .695051
(47.375   IN)WITH STMIX.)FIRST SONIC@ 72 DEG.V.A.)
ABSOL.MAX.TORQUE(FIRST SONIC)AT 72-68 DG.VLV.ANG.= 171488 IN-LBS @ 35 DEG.
MAX.TORQUE INCLUDES SIZE EFFECT(REYNOLDS NO.ETC)APPX. X 1.16482   FOR 47.375
INCH BASIC LINE I.D.
ALL PRESSURES USED:STATIC(TAP)PRESS.-ABSOLUTE;P2 INCL.RECOVERY PRESS.
(TORQUE)CALC'S VALIDITY:P1/P2>1.07;

VALVE TYPE: 48"-R1A;1/6   CLASS 75
DISC SIZE: 46.718 INCHES   OFFSET ASYMMETRIC DISC
SHAFT DIA.: 4.75 INCHES
BRG. COEF. OF FRCTN.: 5.00000E-03
SEATING FACTOR: 15
INLET PRESS.VAR.MAX.: 36.8224   PSIA
OUTLET PRESSURE(P6): 23.5   PSIA (72 DEG. ACTUAL PRESS.ONLY(VAR.))
MAX.ANG.FLOW RATE: 98904.2   CFM; 121031.   SCFM; 6653.43   LB/MIN
CRIT.SONIC FLOW-90DG: 36162.3   LB/MIN AT 19.1166   INLET PSIA
VALVE INLET DENSITY: 6.72714E-02   LB/FT^3-MIN. .104519   LB/FT^3-MAX.
FULL OPEN DELTA P: 5.75262   PSI
SYSTEM CONDITIONS:
PIPE IN-PIPE-OUT -AND- AIR/STEAM MIXTURE SERVICE @ 225 DEG.F
MINIMUM 0.75 DIAM. PIPE DOWNSTREAM FROM CENT.LINE SHAFT.

P1 ABS. PRESSURE(ADJ.)FOLLOWS TIME/PRESS.TRANSIENT CURVE.
ABSOLUTE MAX.TORQUE IS DEPENDENT ON DELAY TIME AND 3.43 TO 2.15-TH POWER
OF (P1/P2)IN WORST RANGE X LINEAR CONSTANT X DWNSTR.PRESS.   P6-ABS.(75-60DEG.)
IN SUBSONIC RANGE LIMITS-ONLY;SEE FORMULATIONS.-PER TESTS H.PRATT
THIS TQ. AT 72 DEG.SYMM. DISC (68=OFFSET SHAFT)CT=T/D^3/P2(ABS)

--5 IN.MODEL EQUIV.VALUES------					ACTUAL SIZE VALUES-----				
ANGLE APPRX.	P1 PSIA	P2 PSIA	DELP PSI	PRESS. RATIO	FLOW (SCFM)	FLOW (LB/MIN)	TD ----INCHLBS----	TB+TH	TD-TB-TH	TIME(LOCA) SEC.
35	23.70	15.84	7.86	.669	121031	6653	26296	115	26180	1.00
30	26.92	14.80	12.12	.550	106855	5874	22622	187	22434	1.42
25	29.02	14.76	14.26	.509	86530	4756	18309	235	18073	1.76
20	30.12	14.73	15.39	.489	52561	2889	14270	273	13997	1.95
15	30.46	14.71	15.76	.483	28632	1574	7030	311	6718	2.00
10	31.98	14.70	17.28	.460	14868	817	4783	384	4398	2.18
5	34.48	14.70	19.78	.426	4620	254	3856	445	3410	2.52
0	36.82	14.70	22.12	.399	0	0	34116	450	33665	2.94

SEATING + BEARING + HUB SEAL TORQUE (M/M)=   34116 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) =   26296 IN-LBS @ 35 DEG.

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SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 40 DEG.

MAX.ANG.FLOW RATE: 133998. CFM; 163977. SCFM; 9014.24 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M)= 34147 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 42734 IN-LBS @ 40 DEG.

AT 1 SEC.DELAY TIME TO 3.22222 CLOSED VLV.(LOCA)TIME(23.7 TO 38.357 P
SIA UPSTR.PRESS.)

REYNLDS NO.FACTOR(MULTIPL.)= 1.31498
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)*(P6/P2)*J9= 1.44557

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SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 45 DEG.

MAX.ANG.FLOW RATE: 166950. CFM; 204300. SCFM; 11231. LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M)= 34172 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 68292 IN-LBS @ 40 DEG.

AT 1 SEC.DELAY TIME TO 3.5 CLOSED VLV.(LOCA)TIME(23.7 TO 39.6 PSIA UP
STR.PRESS.)

REYNLDS NO.FACTOR(MULTIPL.)= 1.34852
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)*(P6/P2)*J9= 1.48244

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SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 50 DEG.

MAX.ANG.FLOW RATE: 205708. CFM; 251730 SCFM; 13838.3 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M)= 34201 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 89725 IN-LBS @ 45 DEG.

AT 1 SEC.DELAY TIME TO 3.77778 CLOSED VLV.(LOCA)TIME(23.7 TO 41.0175
PSIA UPSTR.PRESS.)

REYNLDS NO.FACTOR(MULTIPL.)= 1.31623
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)*(P6/P2)*J9= 1.44695

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SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 55 DEG.

MAX.ANG.FLOW RATE: 253658. CFM; 310407. SCFM; 17063.9 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M)= 34229 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 113972 IN-LBS @ 45 DEG.

AT 1 SEC.DELAY TIME TO 4.05556 CLOSED VLV.(LOCA)TIME(23.7 TO 42.4069
PSIA UPSTR.PRESS.)

REYNLDS NO.FACTOR(MULTIPL.)= 1.30717
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)*(P6/P2)*J9= 1.43698

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SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 60 DEG.

MAX.ANG.FLOW RATE: 303125. CFM; 370942. SCFM; 20391.7 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M)= 34257 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 150084 IN-LBS @ 55 DEG.

AT 1 SEC.DELAY TIME TO 4.33333 CLOSED VLV.(LOCA)TIME(23.7 TO 43.7648
PSIA UPSTR.PRESS.)

REYNLDS NO.FACTOR(MULTIPL.)= 1.29943
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)*(P6/P2)*J9= 1.42848

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SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 65 DEG.

MAX.ANG.FLOW RATE: 362164. CFM; 443188. SCFM; 24363.3 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M)= 34284 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 188248 IN-LBS @ 55 DEG.

AT 1 SEC.DELAY TIME TO 4.61111 CLOSED VLV.(LOCA)TIME(23.7 TO 45.0868
PSIA UPSTR.PRESS.)

REYNLDS NO.FACTOR(MULTIPL.)= 1.28947
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)*(P6/P2)*J9= 1.41753

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SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 70 DEG.

MAX.ANG.FLOW RATE: 405528. CFM; 496253. SCFM; 27280.4 LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M)= 34310 IN-LBS @ 0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) = 264620 IN-LBS @ 65 DEG.

AT 1 SEC.DELAY TIME TO 4.88889 CLOSED VLV.(LOCA)TIME(23.7 TO 46.367 P
SIA UPSTR.PRESS.)

REYNLDS NO.FACTOR(MULTIPL.)= 1.26626
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)*(P6/P2)*J9= 1.39201

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SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 75   DEG.

MAX.ANG.FLOW RATE:    476489.    CFM; 583090.    SCFM; 32054.1    LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M)=      34335 IN-LBS @  0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) =  331197 IN-LBS @ 65 DEG.

AT 1    SEC.DELAY TIME TO 5.16667    CLOSED VLV.(LOCA)TIME( 23.7    TO 47.5964
PSIA UPSTR.PRESS.)

REYNLDS NO.FACTOR(MULTIPL.)= 1.25698
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)*(P6/P2)*J9= 1.38181

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SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 80   DEG.

MAX.ANG.FLOW RATE:    496742.    CFM; 607875.    SCFM; 33416.5    LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M)=      34359
CPU STEP LIMIT OF     20 EXCEEDED IN STEP VARTOBL1 ENTER NEW LIMIT --35
 IN-LBS @  0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) =  387114 IN-LBS @ 65 DEG.

AT 1    SEC.DELAY TIME TO 5.44444    CLOSED VLV.(LOCA)TIME( 23.7    TO 48.7609
PSIA UPSTR.PRESS.)

REYNLDS NO.FACTOR(MULTIPL.)= 1.25024
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)*(P6/P2)*J9= 1.3744

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SUMMARY TORQUE TABLE-VALVE BLOCKED TO: 85   DEG.

MAX.ANG.FLOW RATE:    526939.    CFM; 644828.    SCFM; 35447.9    LB/MIN

SEATING + BEARING + HUB SEAL TORQUE (M/M)=      34381 IN-LBS @  0 DEG.
MAX.DYN. - BEARING - HUB SEAL TORQUE (M/M) =  416576 IN-LBS @ 65 DEG.

AT 1    SEC.DELAY TIME TO 5.72222    CLOSED VLV.(LOCA)TIME( 23.7    TO 49.8322
PSIA UPSTR.PRESS.)

REYNLDS NO.FACTOR(MULTIPL.)= 1.24543
TOTAL TORQ.INCREASE-FACTOR(TO MODEL BASIS)-F(RE)*(P6/P2)*J9= 1.36912
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ATTACHMENT 5

GENERAL ARRANGEMENT
AND CROSS SECTION DRAWINGS