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AUTH.NAME	AUTHOR AFFILIATION
CURTIS,N.W.	Pennsylvania Power & Light Co.
RECIP.NAME	RECIPIENT AFFILIATION
SCHWENCER,A.	Licensing Branch 2

See Reply

SUBJECT: Forwards info in response to NUREG-0737,Item II.E.4.2 re containment vent & purge valves.Several valves do not meet required closing time.Investigation underway.Info closes SER Item 84.

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PP&L Pennsylvania Power & Light Company
Two North Ninth Street • Allentown, PA 18101 • 215 / 770-5151

Norman W. Curtis
Vice President-Engineering & Construction-Nuclear
215 / 770-5381

JUN 10 1982

Mr. A. Schwencer, Chief
Licensing Branch No. 2
U.S. Nuclear Regulatory Commission
Washington, D.C. 20555

SUSQUEHANNA STEAM ELECTRIC STATION
QUALIFICATION DOCUMENTATION FOR CONTAINMENT
VENT AND PURGE VALVES
ER 100450 FILE 841-12 Docket Nos. 50-387
PLA-1111 50-388

Dear Mr. Schwencer:

The enclosed documentation is provided in response to NUREG-0737, item II.E.4.2, part 6. We have committed to provide the enclosed documentation to substantiate that the containment vent and purge isolation valves meet the interim qualification criteria to close within the technical specification limits. This documentation completes our commitment and will allow closeout of item II.E.4.2 and SER open item number 84.

A complication has been found during testing of these valves. Several valves do not meet the required closing time. An investigation is currently underway to resolve this problem. We plan to correct the deficiency prior to fuel load or limit the valve open position to assure compliance with the technical specifications. If you have questions, please feel free to contact me.

Very truly yours,

N W Curtis

N. W. Curtis
Vice President-Engineering & Construction-Nuclear

DPM/mks

cc: R. Perch

A046

TELEPHONE 312·844·4000 · TELEX 720·424

PRATT

HENRY PRATT COMPANY

creative engineering for fluid systems

401 SOUTH HIGHLAND AVENUE · AURORA, ILLINOIS 60507

May 20, 1982

Bechtel Power Corporation
Fifty Beale Street
P. O. Box 3695
San Francisco, Calif. 94119

Attn; E. B. Poser

Subj: Susquehanna - Purge valve analysis
Your P. O. 8856-P-31-AC
Pratt Order No. D-34933

Gentlemen:

In response to your May 7, 1982 letter to Mr. R. Nelson, we offer the following:

1. Yes, the reports furnished for Susquehanna are very similar in format and content to those previously performed for other plants (and furnished to the NRC).

2 & 3. P in the bearing friction torque calculation represents ΔP, the differential pressure across the disc at any given disc angle as calculated during the LOCA flow event. Upstream pressure considers the pressure-time (drywell) curve furnished.

U, the sleeve bearing coefficient of friction equals .25 for the bronze bearings furnished. This value is identified in the appendix to AWWA C-504-80 (copy attached for your reference) and is considered to be a higher value than typically expected in service. However, for the sake of conservatism, Pratt has identified bearing friction torque in the reports but has not actually incorporated its value in our shaft stress calculations. That is, the highest stressed section of the shaft is the top stub shaft between the top bearing and the disc, and that section sees the maximum fluid dynamic torque minus the top bearing friction torque. In your reports we have used the dynamic torque value alone.

4 & 5. The basis for our fluid dynamic torque calculation and our statement that maximum dynamic torque occurs when initial sonic flow occurs coincident with 68°-72° disc angles is as follows:

$T_D = C_T D^3 \Delta P$ is the common equation used for dynamic torque calculations for flow through a butterfly valve (again reference the appendix to AWWA C-504-80). In order for Pratt to evaluate our empirical data we began plotting C_T, torque coefficient vs P_2/P_1, pressure ratio, on log-log paper. What we learned is that when C_T above is substituted by C_{T2}, where $C_{T2} = C_T \Delta P/P_2$ then the resultant curves are upward and downward straight lines peaked at the critical sonic pressure ratio. The highest peaks were obtained at 68° for asymmetric disc models and 72° for symmetric discs. Therefore, it is for this convenience in plotting that we have elected to express our torque equation on this absolute downstream pressure basis.

6. The 19-7 psia backpressure condition was an earlier assumption made by Pratt as a typical downstream pressure. In your reports, this assumption was not used as Pratt could not qualify its value. Instead, the downstream pressure was selected by considering the valve closure time and pressure-time curves furnished, such that the downstream pressure at 68° would yield the critical ratio for the air-steam mixture. This is considered to be the worst case approach.

7a. The steam/air mixture with 1.4 lbs. steam per 1 lb. of air was an assumption made by Pratt as a typical condition. Variations on this mass ratio do not have significant impact on torques.

7b. The seating factor is the coefficient of seating or unseating torque as per the appendix of AWWA C-504. It is determined by tests in our lab.

7c. The outlet pressure (P_6) refers to the new value of downstream pressure identified in response number 6 above.

7d. Flow as well as torque coefficients were determined by Pratt in our model testing. The values of mass flow rates presented are intended as reference data for approximate information only.

7e. Hub seal torque refers to the frictional torque at the disc hub to body interface on certain on-center disc designs previously used. The model 1200 valves analyzed do not have hub seals and this value does not apply.

7f. In the Susquehanna valves analyzed, Pratt has considered the effect of a 90° elbow, out of plane with respect to the valve shaft, immediately upstream of the valve as the worst case approach. This elbow effect is folded into the torque coefficients used.

8. The starting, or delay, times were defined by Bechtel as a minimum of .1 second and a maximum of 5 seconds. The .2 second figure used for the 6" valve appears to be unconservative as the maximum delay time would have resulted in a higher upstream pressure. However, it appears that the maximum dynamic torque for that size valve is so low that the variance in starting times will not adversely affect the conclusions made in that report.

The 30 second, 17 second, and 2.8 second closing times were selected from our assembly and test reports which have recorded actual closing times for those valves as measured in our shop under no-flow conditions. They are not the minimum possible times and do not relate to duct pressure. They represent the only known times that Pratt could confirm as the original order did not specify opening or closing times.

9. The appropriate closing times should be determined by the owner. We can and do hereby state that if the pneumatic controls on the actuators are revised such that a valve closing time of 5 seconds or less is achieved, that the result of the reports furnished for Susquehanna will not be adversely effected.

We trust this information will be satisfactory to your needs.

Very truly yours,

T. J. Wrona

T. J. WRONA
Manager,
Contract & Proposal Engineering

TJW:pst
Attach.

CC: A. K. Wilson
R. D. Nelson
R. N. Kaza

The upper surface of the valve disc shall be visible and shall be covered with a pool of water at 0 psi pressure. The length of test shall be at least 5 min., and there shall be no indication of leakage past the valve disc (visible in the form of bubbles in the water pool on top of the disc) during the test period. As an alternative to this test procedure, Class 150A or 150B valves may be given a 150-psi hydrostatic test. During the test, the valves shall be droptight.

Sec. 5.4 Hydrostatic Test

All valve bodies shall be subjected to an internal hydrostatic pressure equivalent to two times the specified shutoff pressure. During the hydrostatic test, there shall be no leakage through the metal, the end joints or the shaft seal, nor shall any part be permanently deformed. The time duration of the hydrostatic test shall be sufficient to allow visual examination for leakage and shall be at least 1 min. for valves 8 in. and smaller, 3 min. for valves 10 in. through 20 in. and 10 min. for valves 24 in. and larger.

Sec. 5.5 Proof-of-Design Tests

Upon request, the manufacturer shall furnish certified copies of the reports covering the tests. One prototype valve of each size and class of a manufacturer's design shall be hydrostatically tested with twice the specified shutoff pressure applied to one side of the disc and zero pressure on the other side. The test is to be made in each direction across the disc, and, in the case of flanged valves, the valve body shall be bolted to a flanged test head. Under the hydrostatic test, the manufacturer may make special provisions to prevent leakage past the seats. No part of the valve or disc shall be permanently deformed by the test. It is the purpose of this section to provide evidence of the adequacy of each basic type offered by a manufacturer to perform under design pressures within the applicable rating for a sufficient number of operations simulating a full service life. The adequacy is to be proved by tests, made on valves selected to represent each basic type of seat design of a size within each applicable group in Table 6 and in a pressure class or classes equal to or greater than the valves being purchased. The required number of cycles appears in Table 6. Every cycle shall consist of applying the differential pressure to the disc in the closed position, then opening the valve (which will relieve the pressure) to the wide-open position and then closing the disc. The valve shall be droptight under the rated pressure differential upon completion of the cycle test.

TABLE 6

Test Cycles Required

Size Group—in.	No. of Cycles
3–20	10,000
24–42	5,000
48–72	1,000

Sec. 5.6 Rejection

Any butterfly valve or part that the inspector may condemn as not conforming to the requirements of this standard shall be made satisfactory or shall be rejected and replaced.

Section 6—Marking and Shipping

Sec. 6.1 Marking

Markings for other than wafer valves shall be cast on the body or shall be on cast plates with raised letters attached to the valve body. The markings shall show the valve size, manufacturer, class, and year of manufacture. The minimum size of letters shall be ¼ in. for valves 3–12 in. in diameter and ½ in. for valves larger than 12 in. in diameter. Corrosion-resistant plates attached to the body and with ⅛-in. etched or engraved letters may be used for marking wafer valves.

Sec. 6.2 Shipping

Valves shall be complete in all respects when shipped. The manufacturer shall use care in preparing them for shipment so that no damage owing to the manufacturer's negligence will occur in handling or transit. All cavities shall be drained of water. Valves larger than 36 in. in size shall be bolted or otherwise fastened to skids in such a manner as to preclude damage in subsequent handling. All unpainted steel- and iron-machined surfaces shall be coated with a protective slushing compound. Full-face flange protectors of waterproof plywood or weather-resistant pressboard, of a least the diameter of the flange OD, shall be fastened to each flange to protect both it and the valve interior. Small valves may be fully packaged at the manufacturer's option. Components shipped unattached shall be adequately protected and identified for correct field assembly.

APPENDIX

Method for Calculating Torques Required to Operate Butterfly Valves

This appendix is for information only and is not a part of AWWA C504.

The following factors affect the torque required to operate rubber-seated butterfly valves:

1. Valve diameter
2. Shaft diameter
3. Bearing friction coefficient
4. Type of seat and seat material
5. Shutoff pressure
6. Velocity
7. Type of disc
8. System head characteristics
9. Piping arrangement

These factors combine to make up the following four classes of torque:

1. *Seating or unseating torque.* This is the torque required to overcome the rubber seat friction when the valve disc is being seated or unseated.

2. *Bearing friction torque.* This is the torque required to overcome the friction between the valve shaft and the shaft bearings.

3. *Dynamic torque.* This is the torque developed by the disc on the valve shaft because of the difference in pressures that exist across the faces of the disc as a result of flow.

4. *Hydrostatic torque.* This is the torque caused by the difference in static head of water on the valve disc above and below the valve shaft when the shaft is horizontal. It is added to or subtracted from the seating or unseating torque.

Operator torques can be calculated using the following formulas:

$$T_s = C_s D^3 \quad (1)$$

$$T_b = 4.71\, D^2 d f P \quad (2)$$

$$T_d = C_t D^3 P \quad (3)$$

$$T_h = 3.06\, D^4 \quad (4)$$

$$V = C_f \sqrt{P} = \frac{Q}{0.785 D^2} \quad (5)$$

in which:

- T_s = seating or unseating torque, in foot-pounds
- T_b = bearing torque, in foot-pounds
- T_d = dynamic torque, in foot-pounds
- T_h = hydrostatic torque, in foot-pounds
- Q = flow, in cubic feet per second
- V = velocity, in feet per second
- D = diameter of valve, in feet
- d = diameter of shaft, in inches
- P = pressure drop across valve, in pounds per square inch
- *C_s = coefficient of seating or unseating torque
- *C_t = coefficient of dynamic torque
- *C_f = coefficient of flow
- f = bearing friction coefficient (usually assumed to be 0.25 for metal bearings)

The required operator torque, T_o, is determined from the higher value given by the two formulas:†

$$T_o = (T_b + T_s + T_h) \quad (6)$$

$$T_o = (1.2\, T_b \pm T_d) \quad (7)$$

Equation 6 is solved by using the maximum pressure drop across the valve with the disc in the closed position; Eq 7 must be solved successively for several values of velocity to determine the maximum combination of dynamic and bearing torques. The head loss that will occur across the valve for each value of velocity can only be determined by considering the hydraulics of the entire system.

Figure A1 illustrates a graphic method of determining the pressure drop across a valve for any velocity between the valve-open and closed position for two typical conditions: (1) where the elevation of the energy gradient at the source of supply is constant, as in the case of a large

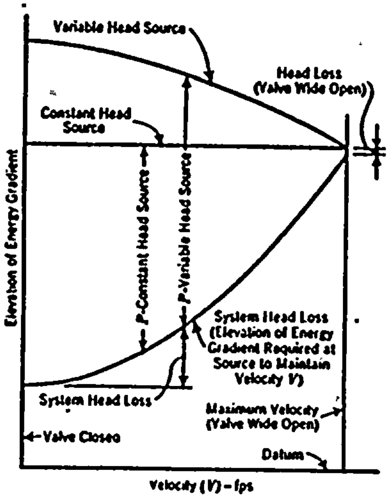

Fig. A1. Relation Between Velocity and Pressure Drop in Butterfly Valves

The figure illustrates a graphic method for determining pressure drops across a valve under the two conditions described in the text.

* The relationship of C_s, C_f, and disc angle varies depending upon the shape of the disc and how it is mounted on the shaft. Characteristic curves showing this relationship and the value of C_s may be obtained from the valve manufacturer. The values of these coefficients should not be applied indiscriminately when calculating torque requirements for operation of valves produced by another manufacturer.

† Use the plus value for T_d when calculating opening torque and the negative value when calculating closing torque.

reservoir, and (2) where the elevation of the energy gradient is variable, depending on flow, as in the case of a centrifugal pump. It should be noted that all heads are measured to energy gradients and therefore must include velocity head, if any. The ordinate difference between the head source curve (or line) and the system head curve represents the total head loss (total dynamic head) across the valve at any velocity; therefore, the pressure drop across the valve corresponding to each value of velocity can be determined and tabulated.

The coefficient, C_f, corresponding to each value of velocity can be calculated by means of Eq 5, and the corresponding coefficient, C_t, determined by using the valve characteristic curves showing the relationship of C_t, C_f, and disc angle. The dynamic torque, T_d, and the corresponding bearing torque, T_b, can then be calculated using Eq 3 and 2, respectively.

The following is an example illustrating the method for calculating the operating torque required for a valve.

Given: A 54-in. valve, a maximum discharge rate, Q, with the valve full open, of 254 cfs, a maximum shutoff pressure across the valve of 25 psi, a shaft diameter of 4⅞ in., valve shaft horizontal, a system characteristic such that head loss varies as the square of the velocity and the head source is a constant elevation, a C_f value of 22 with valve fully open, and a C_s value of 85

Problem: What is the maximum operator torque?

Solution: The velocity and pressure drop with the valve fully open will be:

$$V = \frac{254}{0.785 \times 4.5^2} = 16 \text{ fps}$$

$$P = \left(\frac{16}{C_f}\right)^2 = 0.53 \text{ psi (Eq 5)}$$

Because head loss varies as the square of the velocity and the head source is a constant elevation, the pressure drop across the valve corresponding to any velocity will be:

$$P = 25 - \frac{(25 - 0.53)V^2}{16^2} = 25 - 0.0956\, V^2$$

For any pressure drop across the valve, the bearing torque will be:

$$T_b = 4.71 \times 4.5^2 \times 4.875 \times 0.25\, P \text{ (Eq 2)}.$$

or

$$T_b = 116.2P$$

Table A1 gives the various factors

TABLE A1

Factors Indicated for Velocities at 1-ft Intervals From Maximum Velocity

V fps	Item: P psi	C_f	C_t	T_d ft-lb	$1.2T_b$ ft-lb	$T_d + 1.2T_b$ ft-lb
16	0.53	22.00				
15	3.49	8.03	15.12	4,809	487	5,296
14	6.26	5.59	9.36	5,339	873	6,212
13	8.84	4.37	7.49	6,034	1,233	7,267
12	11.23	3.58	6.19	6,334	1,566	7,900
11	13.43	3.00	5.33	6,523	1,873	8,396
10	15.44	2.54	4.18	5,881	2,153	8,034

A

Bechtel Power Corporation

Engineers—Constructors

Fifty Beale Street
San Francisco, California
Mail Address: P.O. Box 3965, San Francisco, CA 94119

IN REPLY PLEASE REF.:

BLP 20680

MAY 20 1982 0167233

Mr. T. M. Crimmins, Jr.
Pennsylvania Power and Light Company
P. O. Box 1870
Allentown, Pennsylvania 18105

Attention: W. Rhoades

Subject: Susquehanna Steam Electric Station
Units 1 and 2 Job 8856
Containment Purge Valve Analysis

Dear Tom:

The following is a complete reply to your letter No. PLB-14202 dated March 29, 1982, our corresponding Document Control No. 162703.

This is to advise you of the status of the containment purge valve operability analysis by Henry Pratt Co. (P.O. 8856-P-31).

We have reviewed the reports submitted by Pratt for the three valve sizes. These reports indicate that the valves are qualified for the closure times contained in the Techical Specification to meet the NRC's interim position. Comments have been generated for Pratt's resolution. The comments are clarifying/explanatory in nature intended to provide a more complete report and do not identify any errors in the submitted material. In a discussion with Pratt, we understand that Pratt has discussed, in a general manner, the bases for their torque calculations and that reports of a similar format and level of detail have been prepared for other utilities and submitted, apparently successfully, to the NRC. On the basis of the above, we have no reason to believe that the reports are unacceptable and do not believe that the resolution of the comments is necessary prior to submittal to the NRC to close out the SER open item.

Pratt's responses to the comments are to be telecopied to us by May 21. Copies of the reports have been transmitted previously to your J. Agnew.

WP8/50-1

17

Bechtel Power Corporation

E.I 20 '520167233

We are currently investigating the modifications required in order for the valves to be fully qualified (i.e., 5 second closure time), which is required by the first refueling outage.

We will keep you closely informed of the status of this issue.

Very truly yours,

E. B. Poser
Project Engineer

Written Response Req'd: No
JS/GMD/cgs

cc: J. Agnew (PP&L)

bcc: R. Parekh
J. Saame
D. Crosby
A. Daily

MAY 7 '82 0165817

AD file

Henry Pratt Company
401 South Highland Avenue
Aurora, Illinois 60507

Attention: Mr. R. Nelson

Subject: Susquehanna Steam Electric Station
Units 1 and 2 Job 8856
Containment Isolation/Purge Valve
Analysis P.O. 8856-P-31-AC

Gentlemen:

We have reviewed the subject analysis and have the following comments which are generic to all three size valves.

1. Has the torque calculation method in the Henry Pratt's analysis been applied in the same form to other plants (in addition to SSES)?

2. What does the pressure differential, P, in the bearing friction torque calculations exactly stand for? In other words, what are the locations which are selected to determine this pressure differential? How are the pressure at these selected locations calculated from the drywell pressures time history?

3. Also in the bearing friction torque calculation, what is the physical significance of U, designated in the Henry Pratt's analysis as bearing coefficient of friction? What is its value? Has the expressioin of this quantity been verified against any measurement? If yes, please provide references.

4. For the fluid dynamic torque calculation, please provide references which delineate the analytical background of the hydrodynamic torque equations and their compatibility with measurements.

5. Please explain the arguments behind the conclusive statement on page 5 in the Henry Pratt analysis that says that "the maximum dynamic torque occurs when initial sonic flow occurs coincident with a disc angle of 72° (symmetric) or 68° (asymmetric) from the fully closed position."

6. What is the justification for stipulating that backpressure of 19.7-psia can be assumed through out valve closing cycle? Was this stipulation used in the analysis?

7. In the computer output, some of the quantities are either not clear or not well-defined. Specifically,

   7-a. Is the mass ratio of saturated steam to air which equals 1.4 to 1 an assumption?

   7-b. What is the meaning of seating factor which has the value of 15?

   7-c. What does the outlet pressure (P 6) refer to?

   7-d. How does the Henry Pratt's analysis determine the mass flow rates, CFM and SCFM?

   7-e. What does the hub seal torque mean? How is it determined?

   7-f. How are the effects of elbows close to the valve accounted for?

8. The starting times for the analysis are 5 seconds for both the 24" and 18" valves but is 0.2 second for the 6" valve. Please explain the reasons for the difference in the starting times. The closure times appear to be 30 seconds, 17 seconds and 2.8 seconds for the 24", 18" and 6" valve sizes, respectively. Are these the minimum possible closure times? Do these closure times depend on the pressure in the duct?

R. Nelson
Page 3

MAY 7 '82 0165817

9. Please include in each Conclusion Section the appropriate closing times of the valves. Also include a statement to the effect that by changing the solenoid valve on the operator so that the valves close in less than 5 seconds, that the fast closure of the valve will not effect the result of the reports.

Very truly yours,

EBPoser/FJ

E. B. Poser
Project Engineer

Written Response Required: Yes
Date Due: May 14, 1982
Fuel Load Impact: Yes

JS/AJD:sla

cc: E. W. Mead . (PL)

PE51/4-2

Grp. Resp. Code	MD
Orig.	AD
Grp. Ldr.	[illegible]
Grp. Supvr.	
APE	
Oth. Discp.	