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JOB 10394
BECHTEL POWER CORPORATION
San Francisco

PURGE AND VENT VALVE OPERABILITY QUALIFICATION ANALYSIS

Report No. 11-15-83

PREPARED FOR

BOSTON EDISON COMPANY
PILGRIM STATION # 600, UNIT 1

by

Steven M. Nondahl

James E. Krueger

Work performed under Bechtel Purchase Order Number -
P.O. No. 10394-M-119-1-AC, Rev. 1

Clow Job Number: 82-2739-01(N)

This report covers Valve Mark Nos:
8" HBB-BF-AO-5035A
8" HBB-BF-AO-5035B
8" HBB-BF-AO-5036A
8" HBB-BF-AO-5036B
8" HBB-BF-AO-5042A
8" HBB-BF-AO-5042B
8" HBB-BF-AO-5044B
8" HBB-BF-AO-5044A

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Sheets i - vii
and Sheets 1 thru 83

Appendix A - NR Purge Valve Operability Guide Lines

APPENDIX B - SUMMARY OF 12" CLOW TRICENTRIC CHOKE FLOW/STATIC SEISMIC OPERABILITY TEST.

CERTIFICATION

This is to certify that all valves (Tag Nos. 5035A & B, 5036A & B, 5042A & B, 5044A & B) have been evaluated for operability under the installed conditions indicated in supplied drawings and purchasing specifications. The information contained in this report is the result of complete and carefully conducted analyses and to the best of our knowledge is true and correct in all respects. The information presented in combination with the supporting documents referenced, represents a demonstrated qualification of the subject valves to the best of our knowledge for the required service application.

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1. INTRODUCTION

The Nuclear Regulatory Commission has, since 1979, been highly concerned about the operability of purge and vent valves during certain postulated occurrences. Their study in this area has shown that many valves were designed only to operate under normal flow requirements. For a postulated loss of coolant accident, such valves may fail to close in the time required to prevent discharge of radioactive gases to the outside environment. Such a failure could exceed 10 CFR 100 guidelines and present a significant hazard to the health of persons in the area. NRC Branch Technical Position CSB 6-4 gives some background on operations of purge and vent systems and basic requirements for their design. For the valves used in such systems, further guidelines are provided in "Guidelines for Demonstration of Operability of Purge and Vent Valves", which was provided to nuclear plant operators by an NRC letter in September 1979. This set of guidelines covers twenty-one points (less two) which are to be addressed by the plant operator. This paper addresses those items which may be answered by the valve manufacturer based on the conditions provided by the plant operator for the postulated loss of coolant accident.

This paper describes the design of both Clow's Tricentric butterfly valve and the Bettis pneumatic actuator used to operate the valve. In addition, descriptions of various tests performed

to determine flow and torque characteristics and application of this test data to the installed condition of the subject valves are presented. Information as to the structural integrity of the valve and operator assembly under seismic and other inplant loadings are also presented. This information, in combination with the supporting detailed technical reports (see 7.0 references), represents a demonstrated qualification of the subject valves to the best of our knowledge for the required service application.

1.1 Testing Performed

Clow became involved with design of butterfly valves specifically for purge and vent containment isolation early in 1981. A test program was initiated to determine the mass flow and aerodynamic torque characteristics of the Tricentric butterfly valve design. Tests were performed for 12", 24", 48", and 96" scale model valves (scaled to 3" pipe size) in a straight pipe run for both unchoked and choked flow regimes. Pressure ratios for choking, flow coefficients for mass flow, and aerodynamic torque coefficients were determined in these experiments. The experimental set ups met the ISA test requirements for compressible flow measurement. All measurements were automatically read, digitized, and recorded on magnetic tape. The obtained data was then evaluated by other computer programs.

Subsequently, a computer program, CVAP was developed using the measured data base to predict flow and torque values for full size valves in a straight run.

In the Spring of 1981, Clow personnel met with representatives of the NRC to review the test program to that point and to obtain recommendations for additional testing. As a result, Clow and it's fluid dynamic consultant set up two additional programs to determine how the aerodynamic torque characteristics of the Tricentric valve varied with installed piping conditions. For such conditions effects of both upstream and downstream piping elements (elbows, tees, reducers, etc.) were considered. From results of backpressure tests performed in the first set of experiments and water table studies previously done by Clow, it was determined that upstream piping elements would present a worst case condition. Further, due to the numerous types of upstream elements (upstream elbows (mitered, 90°, other angles, short radius, long radius), tees, reducers), a worst case had to be selected for evaluation. A 90° mitered elbow was selected due to the fact that this element presented the worst separated flow region at the inner corner and biased a major portion of the flow to the outer corner. A second set of tests was developed to obtain information about the effect on each other of two valves in series (the common plant installed practice). Due to the fact

that each experiment required an increasing amount of test combinations, the experiments were done in a phased approach.

The upstream elbow tests were performed first for a scale model of a 12" valve in 3 orientations relative to the elbow and at 3 spacings (2, 4, & 8 diameters) from the elbow. From the results a worst case was determined to occur at 2 diameters. Thus a scale model of the 24" and 48" were tested only at 2 diameters. Upstream elbow effects diminished significantly at 4 diameters and were barely detectable at 8 diameters.

From these results, the two valves in series tests were restricted to spacings of 2 and 4 diameters. As in the elbow experiments, the worst case occurred at 2 diameters and at 4 diameters the results approached those for the single valve experiments.

To substantiate the model tests and show the validity of scaling the model data to full size valves, Clow performed a choked flow operational test of a full size 12" valve with a pneumatic spring return actuator at Vought Corp., Dallas, Texas, in November of 1981 (see the appendix for a basic description). The test showed that the valve would operate under the choked flow test conditions, that mass flows were as predicted, and that use of the CVAP program to predict torques was a conservative method (peak measured torque was approximately 65% of that predicted). The test also incorporated a static 11.0 g load

to the actuator simulating a severe seismic/hydrodynamic induced loading. It further validated the directional effects of aerodynamic torque measured in the model tests (in the test all torques tended to close the valve).

1.2 Qualification Method

Clow provides certification of operability of valves produced for purge and vent containment isolation service by a combination of tests and analysis. The following items are considered and covered in this and supplemental reports.

A. Environmental

All portions of the Clow Tricentric is of completely metallic construction other than stem packings and the asbestos seal laminations. The valve seals by metal to metal contact between the seat and seal. The asbestos seal laminations used to separate the SST laminations do contain a SBR binder which may degrade under radiation but the asbestos is uneffected. Further, the asbestos laminations are shielded by the SST laminations and disc components. Although the asbestos may become embrittled on the periphery, the valve will still perform its sealing function (see Radiation Sensitivity Analysis Report Wyle 17629-01).

The packings will perform their function under the required environment as long as they are replaced at recommended intervals.

Actuators used on the valves are qualified for the environment by the actuator manufacturer to codes, standards, or test procedures accepted by the valve buyer.

B. Structural (For Seismic and Other Loadings)

Clow provides for each valve design, a finite element analysis of the valve structure and hand calculations of selected components. These analyses show the valve to be constructed within ASME Section III requirements and that elements not covered by the code are designed with adequate safety margin. Analyses can be found in this Qualification Report, the code required Design Report, and the Structural Analysis Report. The elements considered by these reports include:

1. Valve body
2. Valve disc
3. Valve disc shaft
4. Valve disc shaft connection
   a. Disc ear
   b. Drive keys
   c. Dowel pin (retains shaft from hydrodynamic end load only)

5. Actuator mounting structure

   a. Adaptor flange

   b. Bolting

   Actuators are qualified separately by the manufacturer by generic test results.

C. Operability Under Flow

Operability under maximum flow conditions is based on a combination of a bench test of each unit (timed test with no flow) and analysis of the torque characteristics. The bench test shows the closing cycle time when no aerodynamic torque is imposed. This data combined with conservative (see assumptions below) calculations of the aerodynamic torque is used to show the valve will close in the required time. Bench tests of actuators and valve assemblies include operation during worst case conditions (minimum voltage, air supply, or maximum backpressure for pneumatic actuators if applicable).

The following method is used to show operability:

1. Determine no flow worst case operating time from bench tests.
2. Using Clow program CVAP calculate aerodynamic torques for straight pipe conditions.

3. Determine a torque modification factor based on the installed (from buyer prints) or a worst case upstream piping condition using the mitered elbow or two valves in series test data.

4. Determine predicted torque values for all disc angles based on 2 and 3 above.

5. Provide tabulation or plot of actuator output torque for all actuator angles.

6. Show that actuator output provides sufficient margin to overcome aerodynamic and other torques (bearing, packing, disc wt.) to close the valve.

7. From the above data, actuator type, and Vought full size test valve data, project a closing rate under the conditions analyzed above.

In the above calculations, the following assumptions are employed:

a. Containment pressure is at a maximum value and full flow is developed before valve starts to close.

b. The pressure downstream of the valve is atmospheric. In the elbow experiment it was noted that downstream

elbows may choke before the valve for certain disc angles, producing a higher backpressure and lower torques.

c. Upstream piping components may produce a less severe torque condition than the experimental element (mitered elbow worse than radius elbow).

d. Torque coefficients used in the CVAP program are worse case values. In the experiments a band of coefficients was observed with some dependence on pressure ratio. The high end of the band was used in the CVAP program.

e. Scaling of torques to larger size valves by the D^3 method may be largely conservative as was shown by the Vought Test.

The net result of all such calculations and tests to date continue to show that the design and sizing of all components used in the valve or the actuator exceed the aerodynamic closure requirements based on design for suitable torques to seat and seal the valve.

2.0 DESIGN OF VALVE AND ACTUATOR ASSEMBLY

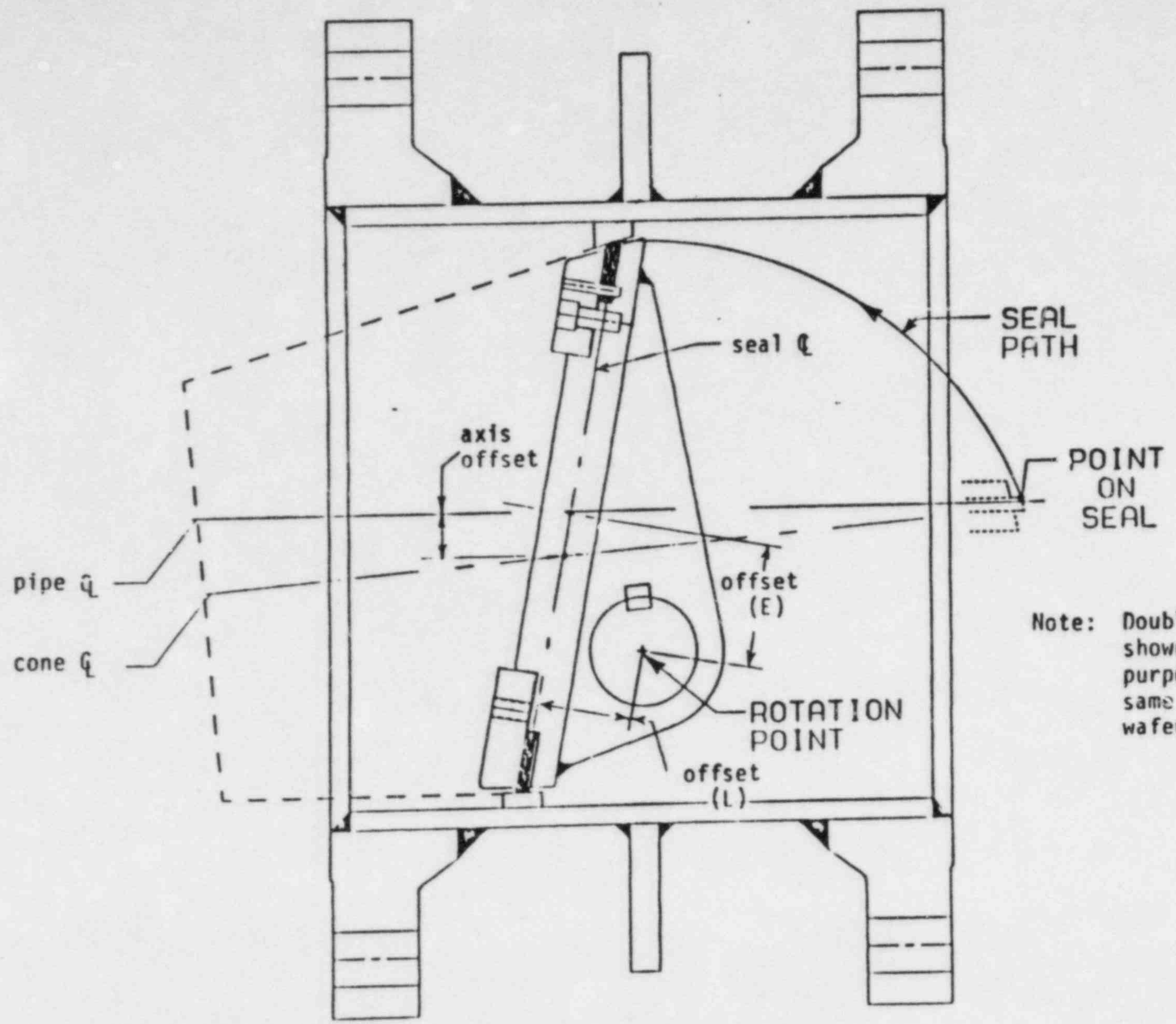
2.1 Valve Design

2.1.1 Geometry

The Tricentric valve uses a geometry that is unique not only to purge valves but to butterfly valves in general. This feature gives the Tricentric functional characteristics which are desirable in purge valve applications. Thru use of a conical sealing surface with, the cone axis offset from the pipe axis and a rotation point selected so that it is offset from both the pipe axis and the seal plane, a metal to metal seal can be obtained. (Fig. 1) The sealing is a result of normal forces acting between the sealing surfaces rather than sealing due to surface interference typical of other butterfly valves with elastomeric seals.

One of the major advantages of the conical seal design is that it provides a non-jamming action. This characteristic results from controlling the cone angle so the angle of friction of the material is exceeded. This has been proven in actual tests similar to the test described here:

> A 20 inch Tricentric wafer valve was closed by applying 20,000 in.lbs. of seating torque. Then the unseating torque was measured. This was repeated 3 times to determine an average value for the unseating



Note: Double flange style shown for illustration purposes only. The same offset apply to wafer style

FIGURE 1 - TRICENTRIC VALVE OFFSETS

torque. The test was repeated with the seating torque increased by 10,000 in.lbs. increments until a maximum seating torque of 100,000 in.lbs. had been achieved. During the entire test, the seat seal interface was dry (highest angle of friction) and no pressure was applied to the valve. The smallest value of torque that could be accurately measured was 1000 in.lbs. and at no time was more than 1000 in.lbs. required to unseat the valve regardless of the seating torque applied.

Since the shaft is offset in 2 directions, one from the pipe axis and one from the seal plane, 2 performance advantages result. The first is the sealing surface is continuous thru 360 degrees with no interruptions from the shaft penetration. This eliminates the leakage and wear associated with the shaft penetration areas. The second advantage comes from the shaft being offset (eccentric) from the pipe axis. This eccentricity produces unequal areas about the rotation point, so when the valve is closed and pressure is applied to the shaft side of the disc (normal direction), a closing moment results. This will result in increased sealing forces between the seat-seal interface as pressure increases. This force, in combination with the mechanical torque produced by the actuator, results in the tight sealing capability achieved with the Tricentric. A definite relationship between these

2 offsets is required to provide a valve that has no binding or interference problems as the seal is rotated out of the seat. This relationship is determined analytically to provide the best performance without overdesigning the valve components.

All of these features have been incorporated into the lugged wafer body that results in a very rugged and sturdy valve design capable of meeting or exceeding all the requirements set forth in most specifications.

2.1.2 Materials

A complete list of valve component materials used on Bechtel Purchase Order Number 10394-M-119-1-AC, Rev. 1 may be found on the General Arrangement Drawing (D-0741) which follows this section.

Since purge and vent valves must perform safety related functions not only during normal conditions but also during and after upset, emergency and faulted conditions, the material selections were based on a worst case event. Because the valves are required to prevent discharge of radioactive gases to the outside environment during a LOCA, the seat and seal materials are critical to the operation of the valves. During normal operation the valves are exposed to the air in the containment and outside air, but during a LOCA the media may be made up of steam, air, and boric acid, all of which may be radioactive and at elevated temperatures. The seat material selected for this application was SA240 316L SST. The 316 grade was selected due

to its corrosion resistance and ability to withstand all of the possible medias that may come in contact with the seat. The L grade of 316 SST was further specified because the seat is welded to the body (SA516 GR 70) and the L grade has a lower carbon content that will reduce the carbide precipitation in the heat affected zone of the seat. The seal is a laminate of 316 SST and asbestos. Both laminants are 1/32 inch thick. The 316 SST was chosen in the "straight" grade since no welding is done on the seal. The asbestos used is made of John Manville style 60 or equal material. The laminated type seal was selected for its ability to seal with less torque than would be required for a solid seal. The laminate allows each SST member to act independently and to conform to the contour of the machined seat as seating torque is applied. The asbestos member not only allows each SST member to act independently but also reduces the seal area in contact with the seat and therefore, results in application of higher normal stresses to the seal for any given seating torque.

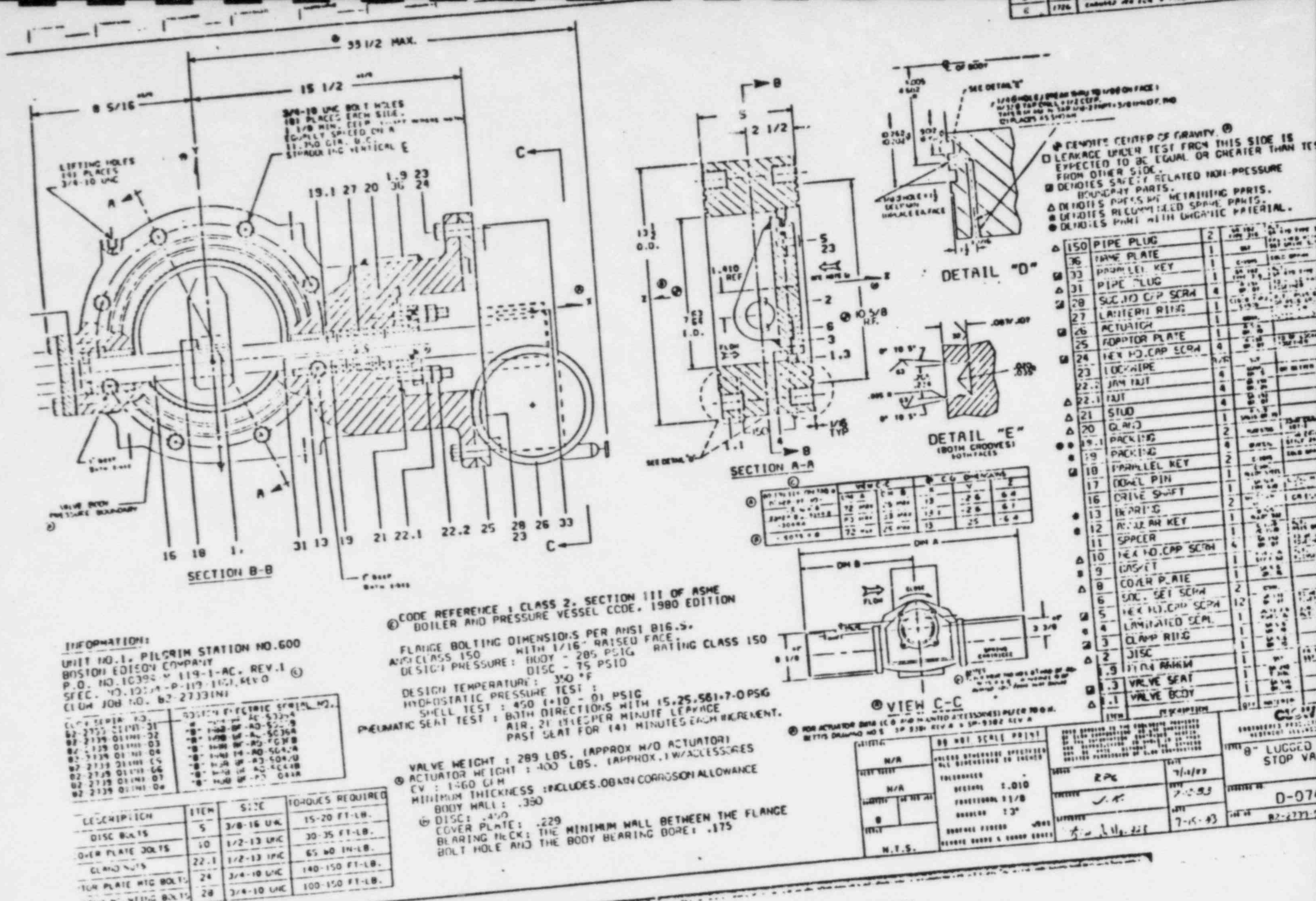


FIGURE 2 - 8" VALVE ASSEMBLY AND MATERIAL

2.1.3 Operation

The operation of the Tricentric valve is extremely simple since there are only 2 moving parts, the disc assembly and the shaft. The valve operates by changing the position of the disc relative to the seat. This is accomplished through the application or control of torque on the valve shaft through the entire operating range of 90 degress. (Zero degrees being fully closed and 90 degrees fully open). There are seven different torques of importance that the valve will encounter depending on the disc position or change in position required, if any. The valve shaft must be designed to withstand the worst case combination of these operating torques without being overstressed. These torques are described in a random sequence since they may occur in different sequences during actual valve operation.

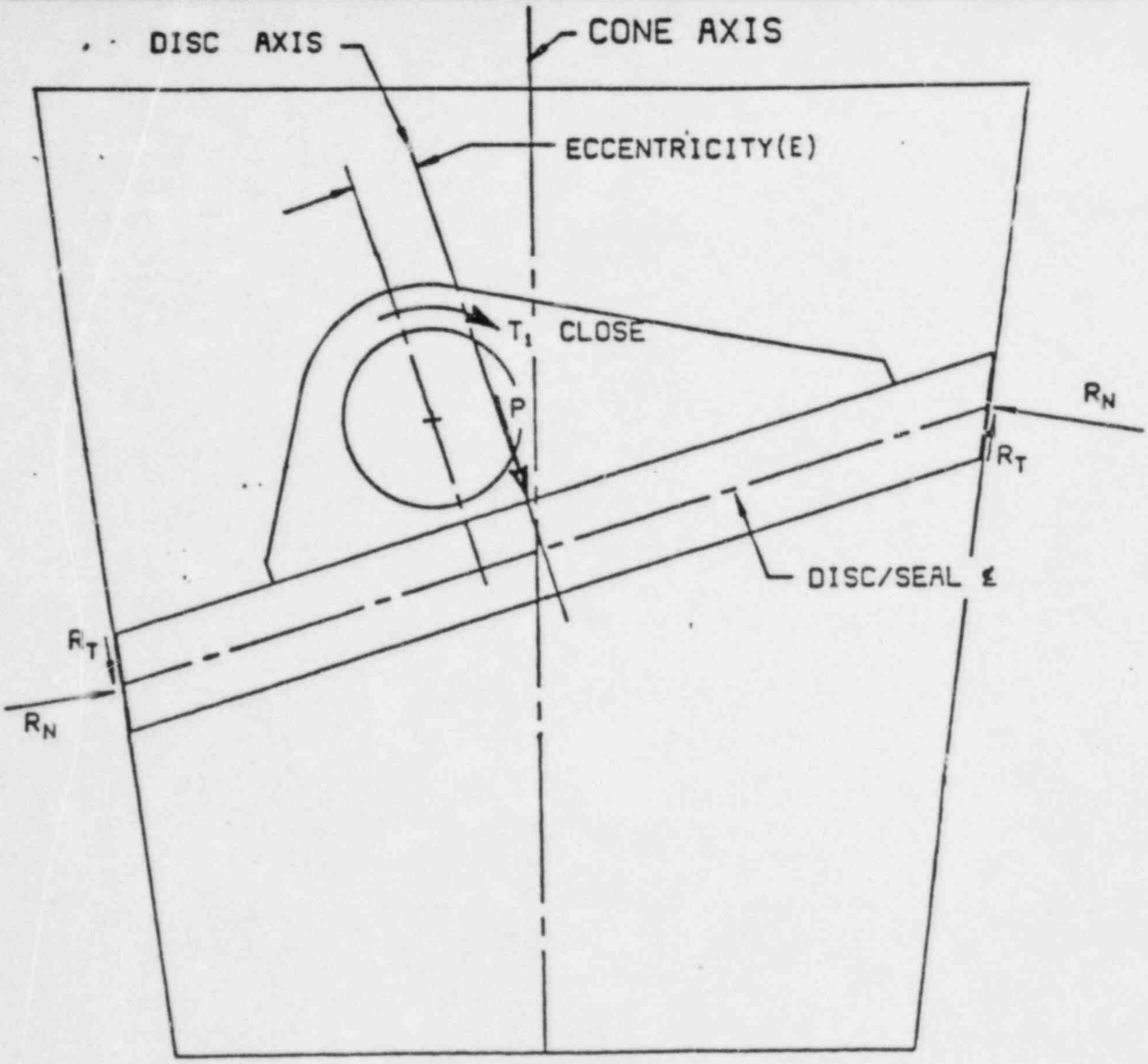
1. Bearing friction torque is the result of the flow or pressure forces acting on the disc which are transmitted to the bearing through the shaft which supports the disc. The bearing friction torque is proportional to these forces acting on the disc and the coefficient of friction between the shaft and the bearing materials. Bearing friction torque must be overcome anytime the disc is required to change position.

2. Packing or seal friction torque is the result of the normal forces the packing exerts on the shaft. These normal

forces are due to the packing gland force and the internal valve pressure. The packing gland force is required to effect a shaft seal. The packing friction torque is also dependent on the coefficient of friction between the packing and the shaft material. Packing friction torque must also be overcome when the disc is required to change positions.

3. PAM (Pressure Area Method) torque is the torque produced by the differential pressure acting on the unequal areas of either side of the eccentric shaft centerline. (Fig. 3) The PAM torque is therefore dependent on the valve size, shaft eccentricity and the differential pressure. Depending on which side of the disc the pressure is applied, the PAM torque may aid seating or unseating of the valve disc.

4. Seating torque is the amount of torque required to develop the normal forces between the seat and seal to effect a tight closure. Seating torque is dependent on the sealing materials, seal thickness, valve geometry, valve size, differential pressure and leakage requirements. As seen in Fig. 3, as the valve is seated by applying a closing moment T_1, the normal forces R_N will increase. Since the seal angle varies around the seal circumference, R_N also varies, thus the point where R_N is a minimum must be loaded sufficiently to effect a seal. Sealing characteristics will be further discussed in the section under Valve Sealing Characteristics (Section 6.0).



T_1 = Closing torque applied by actuator

P = Force equivalent to disc pressure loading

R_N = Normal seat reaction force due to torque application

R_T = Tangential seat reaction force due to disc motion (friction)

DISC WITH CLOSING FORCES APPLIED

FIGURE 3

5. Unseating torque is the torque required to move the seal out of contact with the seat. Unseating torque is also dependent on the sealing materials, seal thickness, valve geometry, valve size, differential pressure, and also the seating torque. As described in the section under Valve Design, when no pressure was applied to the valve, the unseating torque was small relative to the applied seating torque. However, when pressure is applied to the shaft side of the disc, not only does the normal force (R_N) increase but also the frictional force (R_T) which resists opening. This increase in frictional force may exceed the PAM torque. Thus an actuator is selected to provide an output torque greater than PAM torque.

6. Weight offset torque is the result of the C.G.* of the disc being displaced from the rotation point. The weight offset torque is proportional to the disc weight, shaft eccentricity, disc position, and the valve installation position. On small size valves the weight offset torque is generally an insignificant amount since the disc weight is so small.

7. Fluid aerodynamic torque is the torque due to interaction of the flowing media with the valve disc. This is covered in detail in Section 5.0.

* Center of Gravity

As seen in the Vought Corp. Test Report (Reference 7.0 B-1) the running torque was approximately 1000 in.lbs. This is seen in Fig. 8 Run 1 and Fig. 15 Run 8 with no flow through the valve. This running torque is a combination of bearing, packing, and weight offset torque values. The unseating torque may also be seen, which was approximately 1500 in. lbs. when a seating torque of approximately 18,000 in.lbs. was used to close the valve with a 80 psig air supply to the actuator.

2.2 Actuator Design

2.2.1 Geometry

The basic actuator is a device by which air pressure is converted to thrust through a linear cylinder and then converted to a rotary (90°) motion through the use of a "Scotch-Yoke". This device has a torque output at the beginning and end of its stroke, commonly referred to as breaking torque, that is approximately twice the magnitude of the torque output at the center of its stroke, referred to as running torque. The basic design of the scotch yoke can be seen in Figure 4.

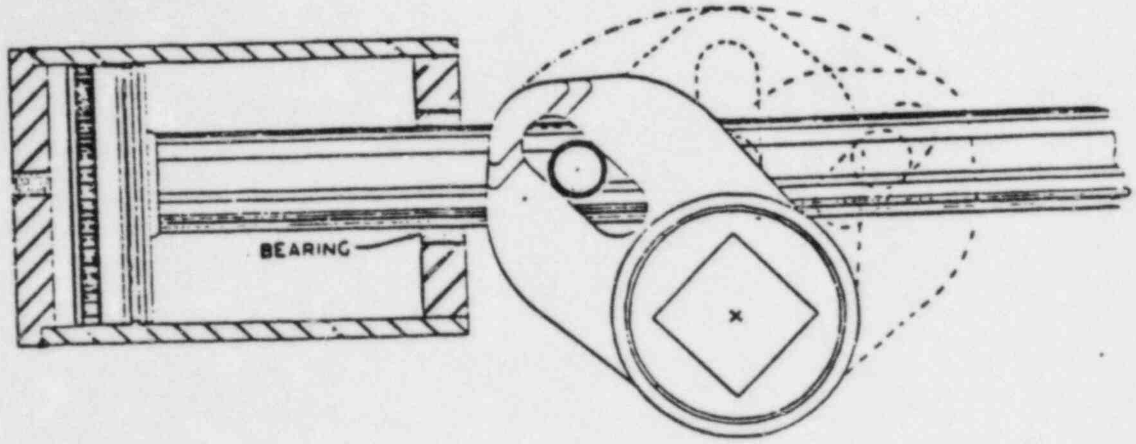


FIGURE 4 - ACTUATOR SCOTCH YOKE DESIGN

From the above it can be seen that the moment arm varies throughout the stroke. By geometric design the moment arm length at the beginning and end of the stroke can be found by dividing the moment arm length at the center by the cosine of 45^0 or .707. By performing this arithmetic it will be found that the moment arm at the beginning and ending is roughly one and one half times the moment arm at the center.

By design the "Scotch Yoke" mechanism multiplies the force imparted by the piston thru a reaction from the bearings. As pressure is applied to the piston the pin or roller is moved against the slot in the yoke causing the rod to act on the bearing. To keep the action in a static condition a force or resistance must be applied to the yoke equal to the force from the bearing. The total resultant force then becomes the piston area times the pressure applied divided by the cosine of 45^0.

The torque output from a "Scotch-Yoke" mechanism can be calculated as follows:

TORQUE AT CENTER OF STROKE

$$T = P \times A \times MA$$

Where:

T = Torque in in-lb

P = Operating pressure in p.s.i.

MA = Moment arm in inches at center

A = Area of the piston in square inches

TORQUE AT BEGINNING AND END OF STROKE

$$T = F \times \frac{MA}{\mathrm{Cos}.45^{\circ}}$$

Where:

T = Torque in in-lb

F = Resultant total force in pounds = PXA

$\frac{MA}{\mathrm{Cos}.\ 45^{\circ}}$ = Moment arm at beginning and end of stroke in inches.

A graphic representation of the torque output as a function of disc position can be seen in Figure 5.

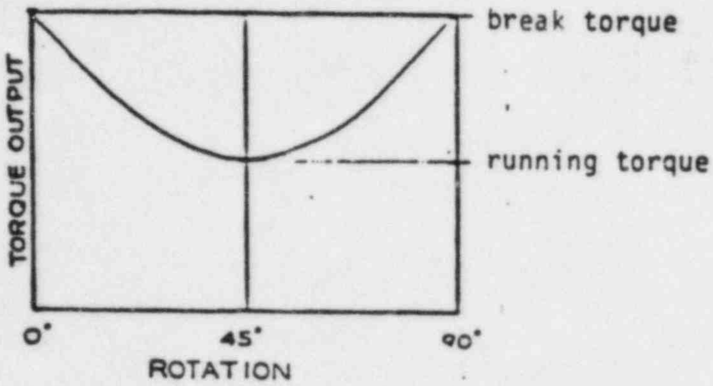


FIGURE 5 - Typical torque output for double acting scotch yoke actuator.

Since thrust is converted to rotary motion, a spring is used opposing the air cylinder to provide a "Fail Safe" actuator. The "Fail Safe" actuator is capable of performing its safety related function in the event of a loss of either the air supply or the control signal to the solenoid valve which controls the air supply to the actuator. The basic construction of the "Fail Safe" actuator is seen here.

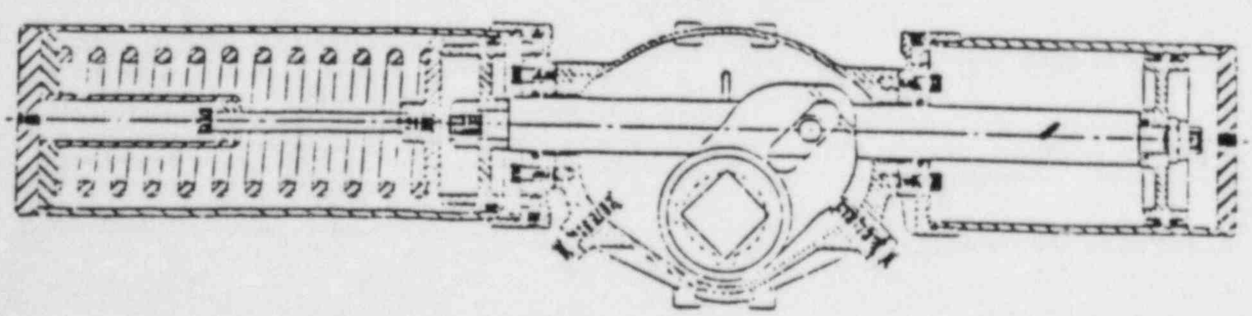


FIGURE 6 - Fail safe, spring return actuator design

2.2.2. Actuator Design Materials

The Bettis actuators used for this job are 732-SR80-S series actuators. These were further specified to be the N version for nuclear service and qualified per IEEE 323-1974, IEEE 344-1975, and IEEE 382. Also, upgraded seismic qualifications are provided based on Patel Report PEI-TR-83-29, Rev. A with Addendum I and II. These actuators incorporate use of special materials for nuclear service as listed below.

Special Material:

Grease - Mobil 28

Seals - Ethylene Propylene

Internal cylinder coating - Molybdenum disulfide

Yoke pin and rollers - Ryton coated

2.2.3 Actuator and Valve Operation

2.2.3.1 Actuator and Accessories Supplied

A complete list of all accessories specified for use on each valve can be found in Table 1 and each is further described here.

An Asco solenoid valve is used on each actuator to control the air supply to the actuator and, to "dump" the air in the cylinder which allows the valve to open or close as required. The solenoid valves are 3 way, internal piloted diaphragm valves. The solenoid valves are controlled by a coil. When the coil is de-energized by intentional or faulted conditions, the cylinder port is allowed to discharge through the exhaust port and thereby allow the spring return actuator to perform its required function. When the coil is energized, the supply pressure is directed into the cylinder and rotates the valve in a direction opposite to spring induced rotation. The solenoid valve model recommended for use is a NPL831664E. This valve is designated for use in nuclear power applications which consists of providing IEEE compliance and a waterproof solenoid closure.

It is a high flow valve which has 1/2 in. NPT ports and a 5/8 in. orifice. All elastomeric materials of construction are Ethylene Propylene material for the NP unit.

Since the output of the unit is a function of the thrust applied, a new torque output curve must be used because the air cylinder not only moves the "Scotch Yoke" but must now also compress the spring. A typical torque output graph is shown here for both the pressure stroke and the spring return stroke. A description of actual output torque values will be presented in the Operation Section.

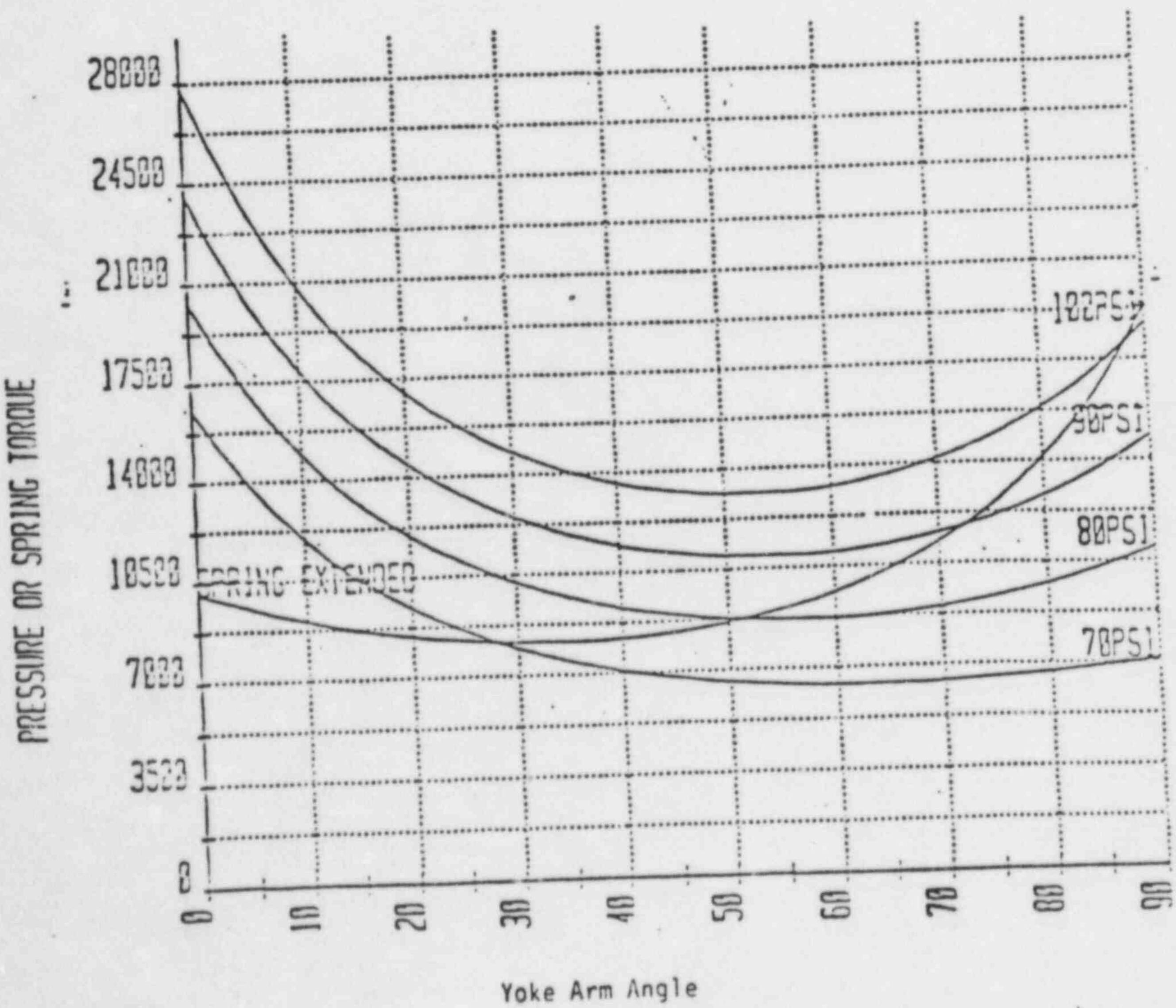


FIGURE 7 - Typical torque output curves for a spring return actuator

Limit switches are also provided. These are mounted on the actuator to indicate full open or closed position. One of each model number switch is supplied, one for the open position and the other for the closed position. The switch model numbers are Namco EA180-31302 and EA180-32302 which are DPDT switches with 2 NO and 2 NC contacts and are quick make-quick break type. The switches meet NEMA 1, 4, and 13, and IEEE 344 requirements. Both switches use the same lever arm which is a Namco model EL010-53337.

Other accessories to the actuator include a Fisher type 95H regulator, A Y6-1/2-40-CI Rosedale Filter, A1008 CHNF Hoffman Junction Box, Anaconda flexible liquid tight conduits, and various tubing, pipe, and electrical fittings and appropriate mounting hardware. All items were not supplied with full nuclear IEEE qualifications. The unit as sold will perform its intended function to fail close even if failure of unqualified components occur. Further, seismic tests performed und Clow Job 82-2053(N) did show such unqualified items performed their intended function under the required vibration level of the Specification as they were mounted for the test.

The air operators are manufactured in accordance with Bettis Engineering Design Standards.

TABLE 1

I. RECOMMENDED EQUIPMENT

Valve Size (in.)	Mark Nos.	Clow Job No.	Bettis Actuator Model No.	Fail-safe Rotation (viewed act. end of unit)	Fail-safe Valve Position	Actuator Accessories for Each Unit: Asco Solenoid Valve Model No.	Namco limit switches and lever are Model Nos. (1 closed position switch) (1 open position switch)	Other Items (each unit)
8"	5035A	82-2739(N)	N732-SR80-S	CW	Close	NPX831664E*		
	5035B	"	"	"	"	"	EA170-31302 L.S. #	Fisher type 95H
	5036A	"	"	"	"	"		regulator
	5036B	"	"	"	"	"	EA170-32302 L.S. #	Rosedale Y6-1/2-40-C
	5042A	"	"	"	"	"		filter
	5042B	"	"	"	"	"	EL010-53337 L.A.	Misc fittings and
	5044B	"	"	"	"	"		electrical accessor
	5044A	"	N732-SR80-M3-S	"	"	"		

II. SUPPLIED EQUIPMENT

Valve Size (in.)	Mark Nos.	Clow Job No.	Bettis Actuator Model No.	Fail-safe Rotation (viewed act. end of unit)	Fail-safe Valve Position	Actuator Accessories for Each Unit: Asco Solenoid Valve Model No.	Namco limit switches and lever are Model Nos. (1 closed position switch) (1 open position switch)	Other Items (each unit)
8"	5035A	82-2739(N)	N732-SR80-S	CW	Close	NPL831664E*		
	5035B	"	"	"	"	"	EA180-31302 L.S. #	Fisher type 95H
	5036A	"	"	"	"	"		regulator
	5036B	"	"	"	"	"	EA180-32302 L.S. #	Rosedale Y6-1/2-40-C
	5042A	"	"	"	"	"		filter
	5042B	"	"	"	"	"	EL010-53337 L.A.	Misc fittings and
	5044B	"	"	"	"	"		electrical accessor
	5044A	"	N732-SR80-M3-S	"	"	"		

*Number difference related to Asco numbering change between when Clow placed order and when units were received.
X indicated special which called for extra length leads
L indicates extra length leads

EA170 units are qualified for outside containment service
EA180 units are qualified for inside containment service

2.2.3.2 Actuator Output Torques

For this job, the Bettis Actuator Company ran tests of and provided certified reports confirming the ending torques of each unit. The tests were performed in both the spring and pneumatic directions, and the results are tabulated in Table 2. There is good correlation between the data and theoretical values.

TABLE 2

TORQUE DATA FOR PILGRIM AIR OPERATORS*

Unit Mark #	Spring Supplied Seating Torque (0° Rotation) in-lb
AO-5035A	9,572
AO-5035B	10,515
AO-5036A	10,432
AO-5036B	10,687
AO-5042A	9,263
AO-5042B	10,470
AO-5044B	10,266
AO-5044A	11,032

*Actuator units were not assembled to valve when torque tests were run

The torque plots provided in this section represent the calculated output torque of the actuators for the spring and various supply pressures shown. The graphs which follow show how the torque output varies for the pressure stroke as a function of supply pressure. It can also be seen that the spring output torque is not a function of supply pressure. The graphs also demonstrate that the output torque (pressure on spring stroke) is a function of yoke position. The graphs provided are based on the numerical data provided.

732 SR80

DATA INPUT

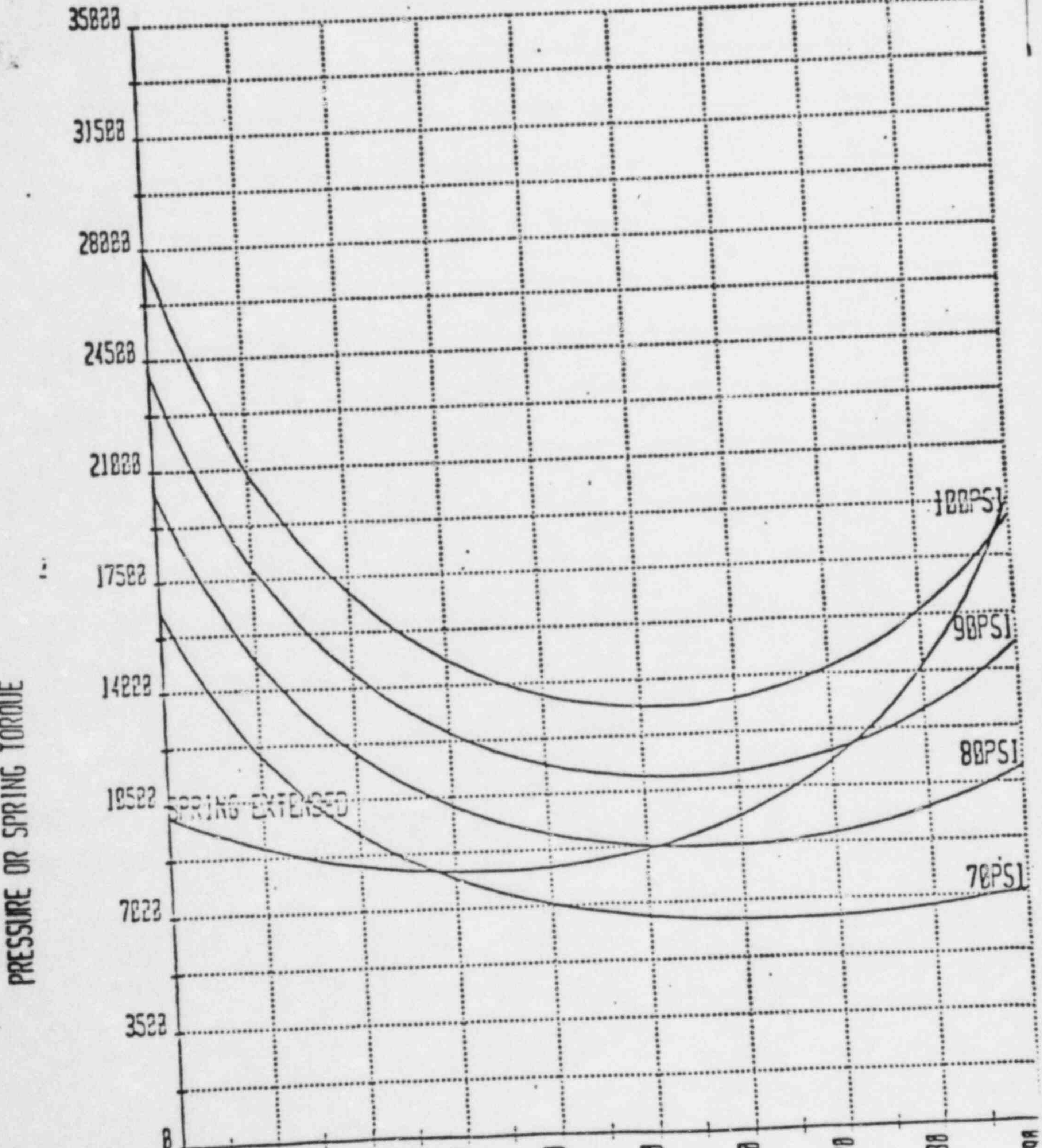
CYLINDER DIAMETER (in)=				7.03
CENTER OR TIE BAR DIAMETER (in)=				0.000
PISTON ROD DIAMETER (in)=				2.125
NUMBER OF PISTONS =				2
MOMENT ARM (in)=				3.375
SPRING LOAD A (lbs)=				1737
SPRING LOAD B (lbs)=				4247
BREAK EFFICIENCY (%)=				75
RUNNING EFFICIENCY (%) =				87
ENDING EFFICIENCY (%) =				78
PRESSURES (psi) =	70	80	90	100
ACTUATOR TYPE,CB=1,HD=2,T,TR=3, =				2

YOKE ARM ANGLE (degrees)	SPRING TORQUE (in lb)	PRESSURE TORQUE (70)psi	PRESSURE TORQUE (80)psi	PRESSURE TORQUE (90)psi	PRESSURE TORQUE (100)psi	EFFICIENCY SPR. %	EFFICIENCY PRES. %
0	10060	16541	20286	24031	27776	78	75
5	9511	13881	17179	20477	23775	80	78
10	9051	11948	14913	17878	20843	82	80
15	8687	10508	13222	15937	18651	83	82
20	8418	9416	11942	14468	16994	85	83
25	8240	8578	10965	13351	15737	86	84
30	8149	7933	10219	12504	14790	86	85
35	8146	7438	9656	11874	14092	87	86
40	8232	7061	9242	11423	13604	87	87
45	8412	6784	8955	11126	13297	87	87
50	8697	6591	8779	10967	13155	87	87
55	9101	6471	8704	10938	13171	86	87
60	9648	6417	8726	11036	13345	85	86
65	10368	6424	8844	11264	13684	84	86
70	11311	6488	9061	11633	14205	83	85
75	12544	6605	9381	12157	14933	82	83
80	14172	6765	9812	12860	15907	80	82
85	16355	6955	10363	13771	17179	78	80
90	19350	[illegible]	11033	[illegible]	[illegible]		

FIGURE 9

EFFICIENCY PLOT

EFFICIENCY vs ANGLE



2.2.3.3. Operating Time

Bench Test - The following is a summary of the operating times recorded during the operational test performed on each valve. The tests were performed using a 100 PSIG air supply. There was no flow through the valve during this test.

TABLE 3

Mark No. of Valve	Valve Size (inch)	Bettis Actuator Model No.	Opening Time Sec.	Closing Time Sec.
AO-5035A	8	N732-SR80-S	4.3	2.7
AO-5035B	8	"	2.6	2.94
AO-5036A	8	"	2.64	2.49
AO-5036B	8	"	5.3	2.1
AO-5042A	8	"	3.29	2.44
AO-5042B	8	"	2.94	2.54
AO-5044B	8	"	3.89	2.71
AO-5044A	8	N732-SR80-S-M3	2.87	2.46

3.0 VALVE OPERATING AND INSTALLATION REQUIREMENTS

3.1 Valve Operating Conditions

The valves were designed to fail close (activation of solenoid valve to allow unit to close to be provided by Boston Edison) and to allow closure and sealing against a 56 PSI differential applied to the shaft side of the disc. Sealing was also to be provided for a differential pressure of 56 PSI applied to the clamp ring side for in plant test purposes. At the request of the buyer, leakage tests were also performed at 5 and 25 PSI differential from each side.

Seismic and other loading conditions for operation are as indicated in the specification. Actuator qualifications for environmental and seismic are covered by previous actuator qualifications supplied by the manufacturer (Bettis) and added seismic tests performed in accord with NTS (National Technical System) Test Plan No. 528-0951 for Clow Job No. 82-2053(N) (see References Sect. 7.0). The Bettis units have been tested to the required level and have demonstrated their ability to function as required.

For the subject valves the following operating and design conditions are applicable.

Operating conditions:	Normal operating pressure	= 1.5 PSI
	Normal operating temperature	= 65-194^{o}F
	Normal operating flow	= 5700 SCFM

Design conditions	Max. operating pressure body only	= 285 PSIG @ 100°F
	*Max. pressure differential disc	= 56 PSID
	Max. temperature	= 350°F
	Required Torque to seat	= 7681 in-lb.
	Failure mode	= Fail close
	Allowed leakage	= .26 cc/min air @ 56 PSID

*Bidirectional sealing provided to 56 PSID only

3.2 Valve Installation Configurations

In addition to the pressure and flow conditions specified in 3.0, the valve performance is affected by the as installed orientation. Upstream and downstream, tees, elbows, reducers, and other valves can affect the aerodynamic torque characteristics of butterfly valves. These effects are discussed in Section 5.0. The installed configurations for the subject valves, as derived from Bechtel drawings SK-P-001, Rev. F, Job No. 10394-105, are summarized in Figures 10 and 11.

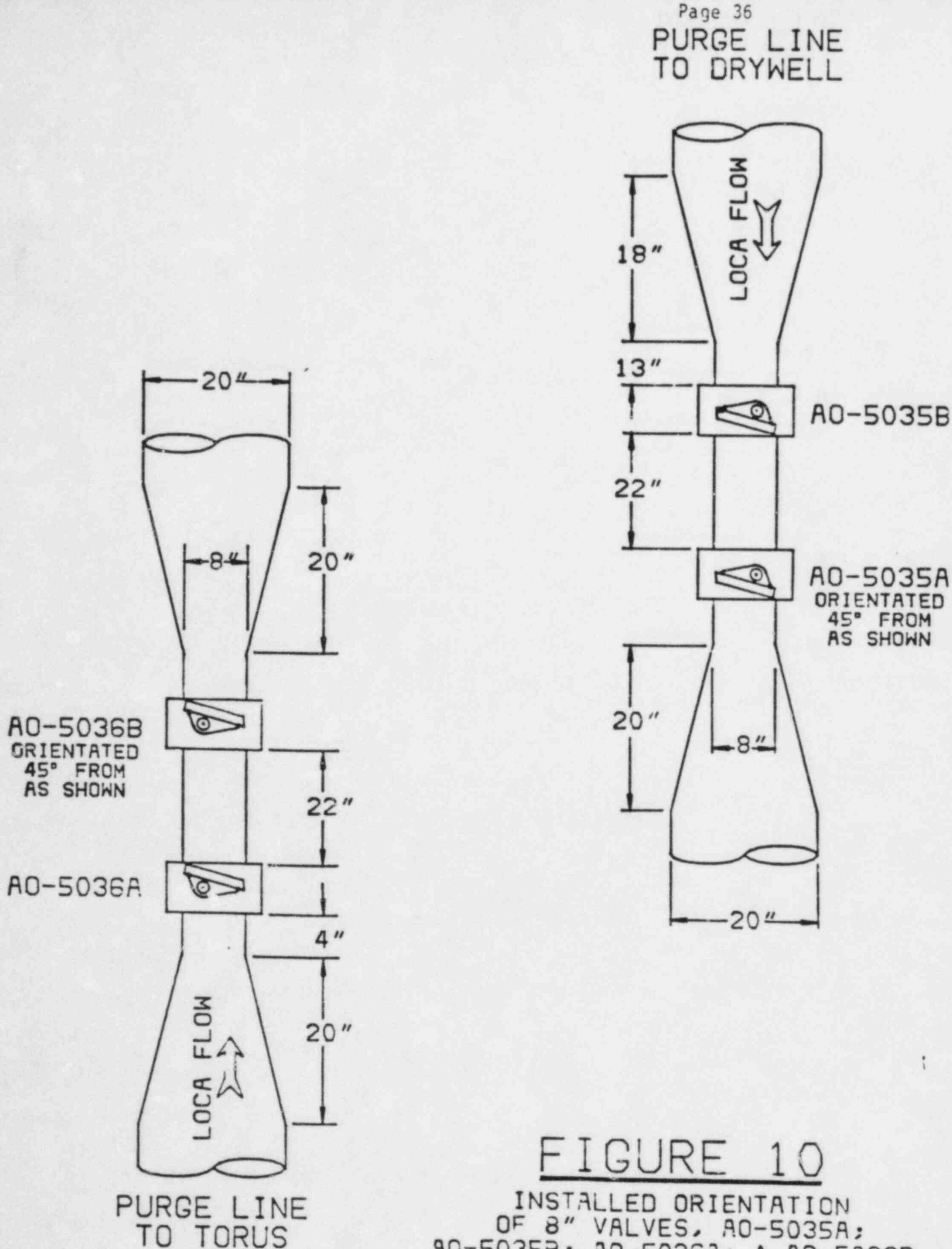
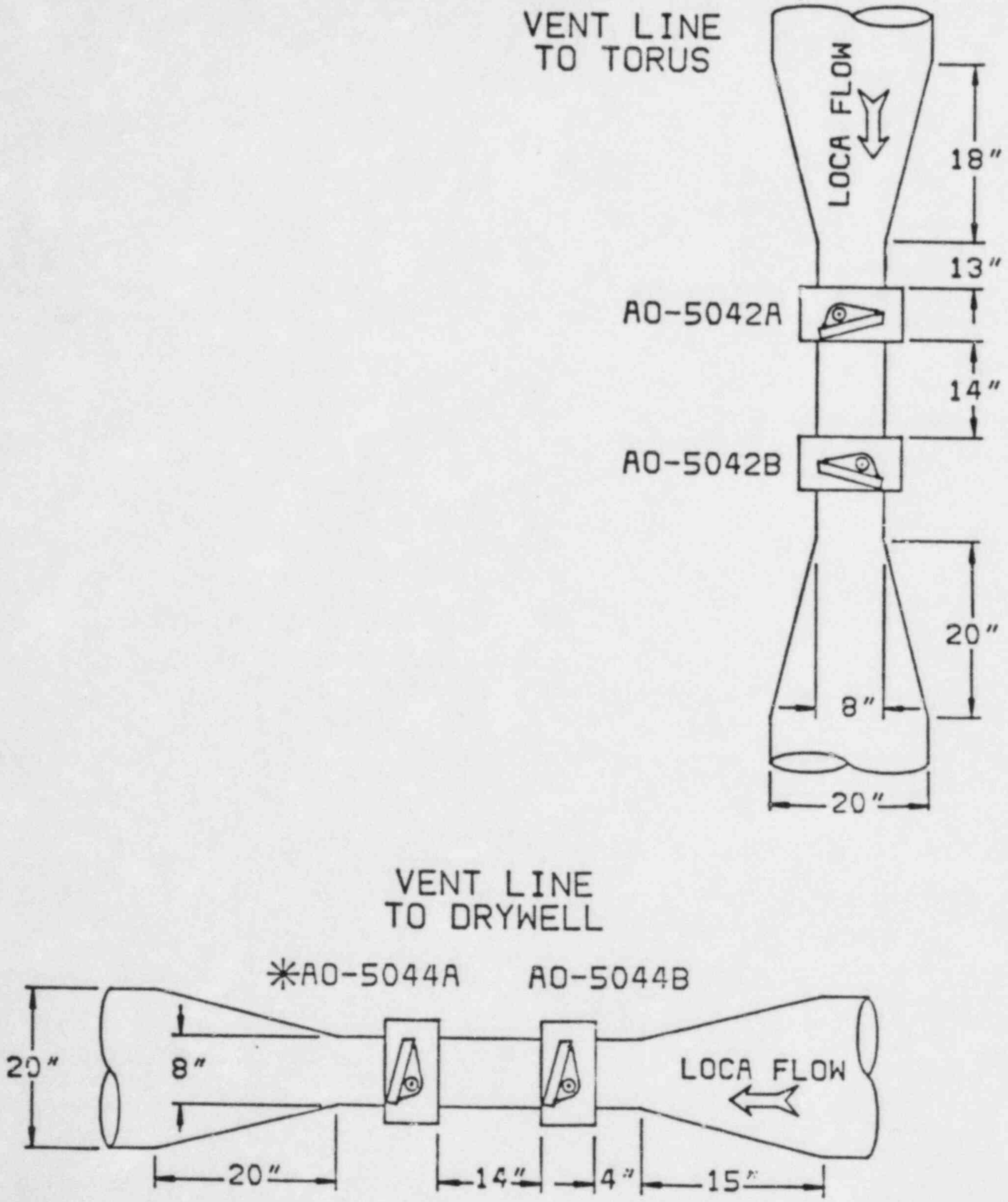


FIGURE 10

INSTALLED ORIENTATION
OF 8" VALVES, AO-5035A;
AO-5035B, AO-5036A, & AO-5036B.



* AO-5044A ORIENTATED 120° FROM AS SHOWN.

FIGURE 11

INSTALLED ORIENTATION OF 8" VALVES, AO-5042A, AO-5042B, AO-5044A, & AO-5044B.

4.0 VALVE STRUCTURAL INTEGRITY UNDER SEISMIC AND OPERATIONAL LOADINGS

Operability of the subject valves has been demonstrated by a combination of testing and analysis in accord with the design specification. Separate reports have been prepared and provided demonstrating suitability of valve components and the assembly. A listing is provided in the References (7.0) at the end of the report. This section summaries the results of such tests and analyses in meeting the conditions as presented in Section 3.0.

4.1 Valve Stress Analysis

Valve stress analysis was performed by Patel Engineers, Huntsville, Alabama for the subject valves. The analysis was made using the ANSYS finite element computer program developed by Swanson Analysis System, Inc., Houston, Pa. This public domain program has had a sufficient history or use to justify its applicability and validity. The analysis performed compares the nuclear specific requirements of the Pilgrim Job, Report PEI-TR-833700-1, to an already performed worst case generic qualification report for a Clow 8" Wafer Stop Valve, PEI-TR-83-24, Rev. A. The comparison of key elements in these reports is shown in Tables 4 and 4A.

TABLE 4

COMPARISON OF PILGRIM NUCLEAR SPECIFIC REQUIREMENTS

TO

GENERIC NUCLEAR QUALIFICATION DATA (REFERENCE 1)

LOADINGS	GENERIC	PILGRIM
Pressure		
Shell (psig)	285	285
Seat (psid)	75	56
Torque (in lb)	16643	9056
Seismic Acceleration		
NS (g)	7.0	4.5
EW (g)	7.0	4.5
Vertical (g)	7.0	4.5
OPERATOR		
Weight (lb)	550	360
Center of Gravity		
X (in)	10	10.81
Y (in)	10	1.01
Z (in)	18	3.99
FREQUENCY f_o (Hz)	59.5	$f_o \geq 33$ Hz
EVALUATION AGAINST	ASME Section III Design and Level A	ASME Section III Design and Level A

Table 4A

Summary of Allowable Stresses

LOCATION	MATERIAL	ALLOWABLE STRESS (psi) (PER ASME SECTION III, TABLES I-7.1 THROUGH I-7.3)	STRESS VALUE (psi)	REPORT IN WHICH ITEM IS ANALYZED** /SEISMIC LOAD LEVEL	STRESS RATIO
Valve Body	SA 516 GR.70	17500	5319	Generic 7.0g	0.30
Disc	SA 516 GR.70	17500	4947	Generic 7.0g	0.28
Drive Shaft	SA 564 Type 630 H-1100	34550	18794	Generic 7.0g	0.54
Operator Adapter Plate	SA 516 GR.70	17500	4936	Pilgrim 4.5g	0.28
Adapter Plate Bolts	SA 193 GR.B7	25000	14569 σ_N 4929 τ	Pilgrim 4.5g	0.08*
Operator/Adapter Bolts	SR 193 GR.B7	25000	10321 σ_N 10767 τ	Pilgrim 4.5g	0.19*
Cover Plate	SA 516 GR. 70	17500	10492	Generic 7.0g	0.60
Cover Plate Bolts	SA 193 GR.B7	25000	7000 σ_N 102 τ	Generic 7.0g	0.01*

*Per ASME, Section III, Appendix XVII, Subsubarticle 2460.

** Generic Report is Patel PEI-TR-83-24. Pilgrim Report is Patel PEI-TR-833700-1

The Pilgrim seismic analysis specifically addresses the operator adaptor plate, adaptor plate bolts, and operator/adaptor bolts since these were determined to be the weakest items. The other items are covered in the generic analysis.

The conclusion that can be drawn is that the structural integrity of the subject valve assemblies fully meets the requirements of Bechtel Design Specification 8031-P-144, Revision 2 and ASME Section III, 1980 Edition.

4.2 Actuator Tests

Two size Bettis actuators were tested for the Bechtel Limerick Project under Clow Job No. 82-2053(N) in accord with National Technical Systems (Saugus, Ca. facility) Procedure 528-0951. The units tested were as follows:

Unit 1	NT-820-SR4-S	Spring ending torque = 93,098 in lb Pressure torque+ = 178,160 in lb
Unit 2	NT-312-SR5	Spring ending torque = 5,810 in lb Pressure torque+ = 31,263 in lb

+ at 80 PSIG pressure to air cylinder

The units were both spring return fail closed units and were representative of the line of actuators which would be used on containment purge and vent valves. The units were tested for baseline performance, subjected to OBE and SSE levels in accord with the procedure and specification, and then were operationally tested. The units proved to be operable both during and after the required tests and upon inspection showed no signs of noticeable wear. Successful operation of these units in combination with previous generic qualification for enviromental conditions generically qualifies the N732-SR80 units used on this job. (Note an addenda is provided to the report justifying similarity, see References, Section 7.0)

4.3 Valve/Actuator Test

In addition to the tests performed on the preceding actuators, a 6" valve in combination with the NT-312-SR5 actuator were statically loaded and tested for operation. The valve operated as required during the tests. This demonstrated that the valve/actuator interface for the 6" valve was sufficiently rigid to allow the unit to perform its safety related function. This test further qualifies the subject 8" valve based on Table C-1 of Bechtel Spec. 8031-P-144 .

5.0 VALVE AERODYNAMIC TORQUES

Depending upon the valve design, actuator sizing, inplant installed configuration, and operating conditions, aerodynamic torque may be of major concern to valve operability. The magnitude and direction of this torque, which is produced by flow of the media over the disc, depends on several factors:

1. Disc shape
2. Pivot shaft location
3. Magnitude of differential pressure across the valve
4. As installed upstream piping elements (elbows, tees, etc.) including distance and orientation relative to these items.
5. As installed downstream piping elements (elbows, tees, length of pipe runs, etc.) including distance and orientation relative to these items.
6. Angle of the disc

Clow has done numerous tests of scale models of the Tricentric design and a test of a full size 12 inch production valve. The data obtained in these tests provide a substantial base for predicting aerodynamic torques in full size production valves under various operating conditions.

5.1 Model Tests

In 1980, Clow established a program to determine mass flow and aerodynamic torques of the Tricentric design. Exact scale models (see Table 5) were designed and built of 150 lb class Tricentric valves of standard design. Scale models of a 12, 24, 48, and 96 inch valve were constructed and tested using University of Illinois facilities under the direction of A.L. Addy, Ph. D. (Engineering Consultant in Fluid Dynamics and Engineering and Associate Head, Department of Mechanical and Industrial Engineering, U. of I. at Urbana, Champaign, Ill.). The tests were made with air in accord with ISA standards for a straight pipe run flow test. The tests were run at various pressure ratios (upstream to downstream pressure) in both the choked and non-choked pressure regimes. Very low pressure ratios were also applied to allow correlation to incompressible (liquid) flow in accord with ISA standards. Tests were made with flow in the normal direction for Tricentrics (shaft upstream) and for reverse flow (shaft downstream). Further, several pressure ratios near the choked flow point were applied to determine the point of choking. This test pointed out that the standard rule of thumb (downstream pressure/upstream pressure = .523) for determining when choking occurs is not valid at all disc angles. The tests showed choking will occur at a ratio of .75 in the full open position and .54 in the near closed

position. The test also showed, that although choking prevents the fluid velocity from increasing, aerodynamic torque will rise in a linear fashion in accord with the pressure differential across the valve in the choked flow regimes.

The models used for testing were made in accord with the Tricentric standard 150 lb class double flange design. This is a fabricated design in which the seat is at a 10 degree angle from a normal to the pipeline axis. Due to the seat position, this valve rotates only 80° from closed to full open. The valves supplied for the subject job uses a similar geometry except the seat is normal to the pipeline axis making this a 90° (¼ turn) valve design. Therefore, at small opening angles (0° to 20°) there are some differences in torque. For angles over this amount, the aerodynamics are the same. Also, at small angles the torque approaches the value of the pressure area torque (as explained in Section 2.1.3) thus, differences between the two designs are not significant. With reasonable similarity between the test models and the full size valves, the data may be used to predict torque characteristics the subject valves.

From the data base developed by the model tests a computer program CVAP (Clow Valve Analysis Program) was written for use in predicting valve operating characteristics. In this program, mass flow rates are predicted by standard equations for flow

through an ideal converging nozzle adjusted with coefficients developed in the tests. Torques are predicted on the basis of the equation

$$T = C_T \Delta P D_V^3$$

where

T = predicted aerodynamic torque (in lb)

C_T = torque coefficient developed in model tests

ΔP = pressure differential across the valve (lb/in^2)

D_V = nominal valve diameter (in.)

The test performed on a full size 12" valve showed that the mass flow obtained was within approximately 10% of that predicted by the computer model while torques were much less than predicted. Torques were on the order of 65% of that predicted which could be correlated by changing the power of 3 to 2.84 in the above equation. The power of 3 used in the equation and in the Program CVAP is a derived value obtained by use of the equations for conservation of momentum for a general control volume. Thus the program indicates torques which would be higher than those obtained in the actual situation.

Table 5 shows the dimension of critical (to torque conditions) elements of the double flange Tricentric 12, 24, 48, and 96 inch designs and their scaled down dimensions which were used for model construction. Table 6 shows a comparison between the provided size valves and the interpolated sizes.

Linear interpolation was used to predict torque characteristics in Clow Program CVAP, thus a similar interpolation of sizes is applicable for size comparison purposes. It can be seen in the table that very good (less than 10% deviation) correlation was obtained for torque critical items. Thus torque data from the program is valid for this application.

TABLE 5

Test Valve Scaled Sizes (Critical Elements)

ELEMENT	VALVE SIZE 12"		24"		48"		96"	
	Full Size	Model Size	Full Size	Model Size	Full Size	Model Size	Full Size	Model Size
I.D.	11.94	3.07	22.62	3.07	46.00	3.07	96.00	3.07
A_2	11.33	2.91	21.89	2.97	45.59	3.04	96.20	3.07
K_2	10.80	2.78	20.86	2.83	43.44	2.90	91.66	2.93
Shaft Dia.	2.25	.58	3.25	.44	6.0	.40	12.0	.38
Shaft CL to Seal CL, L	2.0	.51	2.69	.36	5.06	.34	7.51	.24
Disc Thickness	1.5	.38	1.88	.25	3.75	.25	Domed Shape 11.63	.37
Shaft Offset E	+ 1.2[illegible]	.32	.81	.11	1.31	.09	1.18	.04
Shaft Offset LC	+ 1.67	.43	1.38	.19	2.31	.15	1.66	.05
Ear Width	* 2.25	.58	3.25	.44	6.0	.40	12.0	.38
Ear Height	* 3.33	.87	4.88	.66	9.0	.60	15.25	.49

+ E is offset from disc centerline, LC is offset from body centerline

* Ear is element welded to disc which shaft is mated to.

Note: Full size dimensions are for a Clow Tricentric 150 lb class double flange design.

A_2 = Major axis of elliptical seal
K_2 = Minor axis of elliptical seal
E = Offset between shaft axis and disc center (see Figure 2)
LC = Offset between shaft axis and pipe run centerline

All dimensions in inches

TABLE 6

Comparison of Production Valve to Valve Model Sizes (Critical Elements)

VALVE ELEMENTS	8" Size	8" Ratio
*I.D.	7.981	1.05
*A_2	7.244	1.09
*K_2	7.069	1.07
Shaft Dia.	1.50	1.05
Shaft ℄ to Seal ℄ ,L	1.50	.93
*Disc Thickness	1.25	.95
*Shaft Offset E	1.375	1.05
Shaft Offset LC	1.410	NA
Ear Width	2.00	NA
Ear Height	5.937	NA

*Elements considered important to torque characteristics

NOTE: $\text{RATIO} = \frac{\text{interpolated model size}}{\text{production valve size}}$

A_2 = Major axis of elliptical seal
K_2 = Minor axis of elliptical seal
E = Offset between shaft axis and disc center (see Figure 2)
LC = Offset between shaft axis and pipe run centerline

All dimensions in inches

5.1.2 Tests With An Upstream Elbow

One element of piping system which has an effect on the aerodynamic torque of butterfly valves is a turn which may occur with an elbow or a tee. Since numerous types of elbows (short and long radius, reducing, mitered, etc.) may exist in a particular piping system, it was necessary to determine a worst case condition for testing. It was determined use of a mitered elbow would be a worst case and that this configuration had applicability to flow through tees also.

The mitered elbow produces the greatest separated flow region at the inside of the turn and biases the flow to the outside corner to a maximum. Flow around the corner produces a lower local pressure around the inside of the turn and higher local pressure to the outside. This will oppose closure for geometry 1 (see Figure 12) and aid closure for geometry 2 when the disc is in the full open position.

Based on these considerations, models of a 12", 24", and 48" valve (per Table 5) were tested for torque characteristics. All valve models were tested for geometries 1, 2, and 3 at 2 diameters downstream from the mitered elbow. In addition, the 12" model was tested at 4 and 8 diameters downstream. The test showed the greatest variation of torque from that obtained for straight-line flow occurred at 2 diameters downstream from the elbow. Differences due to valve orientation were small at 4 diameters downstream and were just detectable at 8 diameters downstream.

For the subject job, valves are installed 10 diameters or more from an elbow during LOCA conditions. Further, the pipe for the installed configuration is tapered from a 20" diameter to an 8" diameter pipeline by means of a concentric reducer installed just before the valve. Thus, any turbulance upstream that may affect the torque characteristics of the unit will be eliminated by the flow converging thru the reducer. (See Figure 13) Using a straight line pipe configuration to model the valves in the as-installed configuration, then, would be considered a good model.

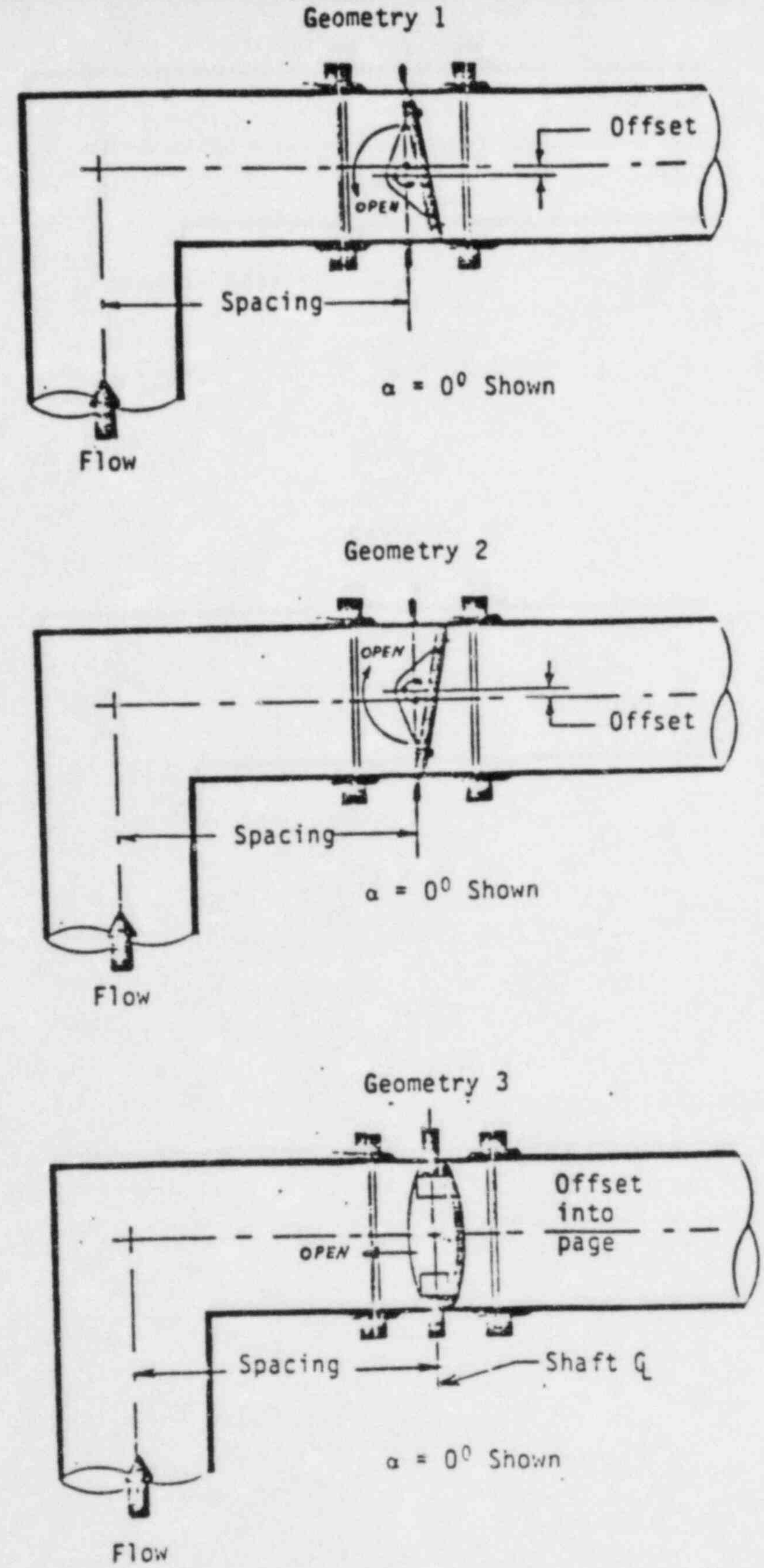
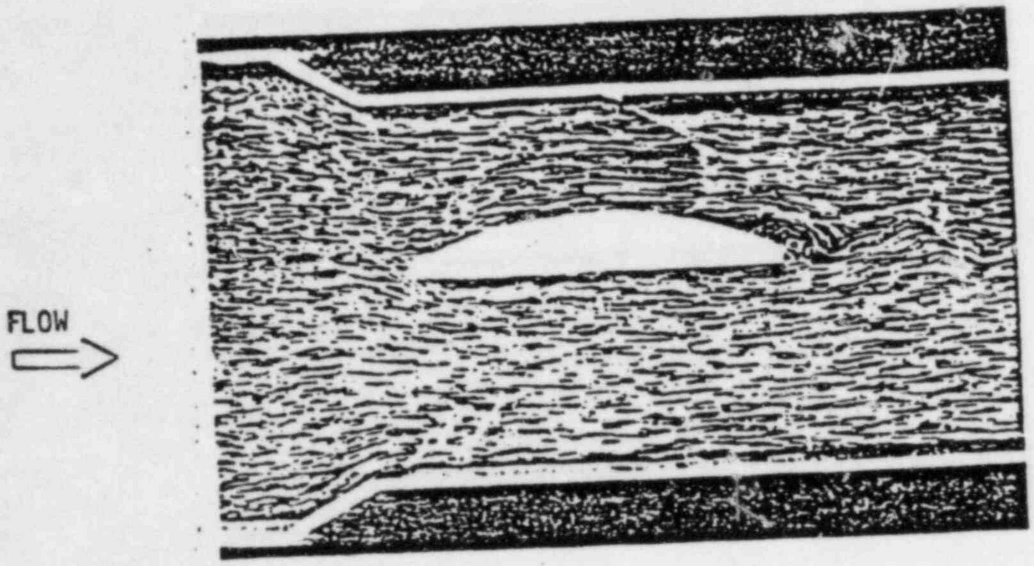
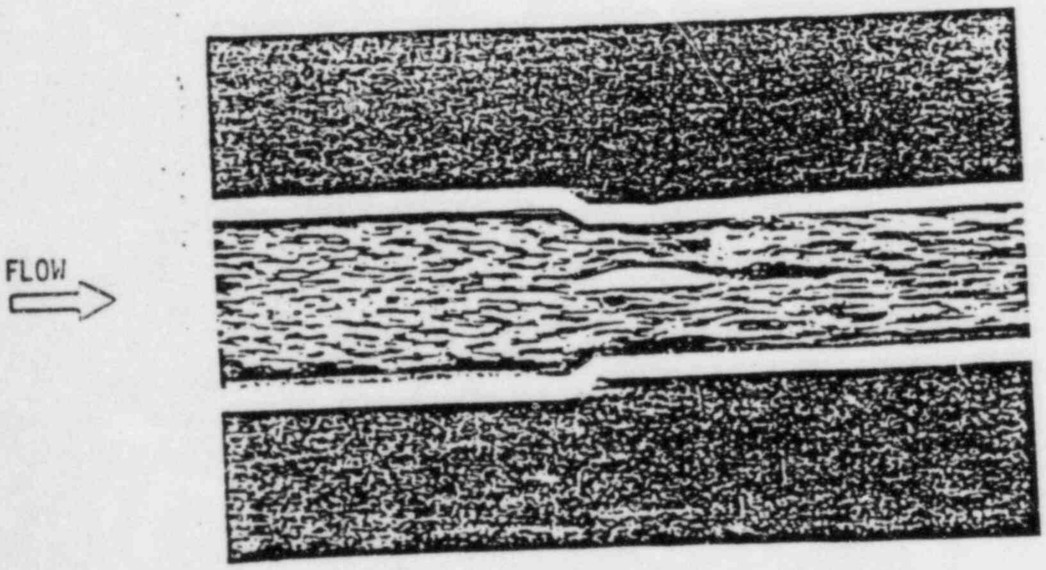


FIGURE 12- Valve Orientations Relative to Upstream Elbow



(a) 36" model



(b) 6" model

Figure 13 The high pressure valve models with a convergent entrance section and a valve-disk opening-angle of 80°.

5.1.3 Downstream Piping Effects

In various tests described in this section, it was necessary to provide downstream piping to discharge the flow. In the conduct of these tests the effects of downstream piping were noted several times. In the straight line tests, a downstream valve was installed to vary back pressure. Any increase in back pressure lowered the torque values. In the elbow tests an elbow was installed 20 or more diameters downstream. It showed that for the 24" and 48" models in the full open position, the downstream piping would choke before the valve model. This prevented any substantial increase in pressure differential across the valve model even with large increases in upstream pressure, thus the torque was limited. From the piping layouts provided downstream, piping would provide some degree of back pressure making the assumption of atmospheric pressure downstream used for calculation of torques conservative.

5.2 Model Data Verification

A test of a full size 12" valve was run at Vought's High Speed Wind Tunnel in Dallas, Texas (see reference 7.0B-1) to demonstrate operability and substantiate model test data. The tests demonstrated the valve would operate in the required 5 second period. It further showed that torque values were less than predicted from model data. The valve used for the test incorporated a one piece thru shaft design while the model had a two piece shaft. To verify the torque effect due to this change, another test was made (data not put into a formal report form) in which a 2 piece shaft was installed in place of the thru shaft. The test was made with the disc held in a stationary position by a manual worm gear type actuator. The result was that the peak torque was the same for both the one and two piece shaft design. The only difference was that the two piece shaft design showed a peak torque closer (by 5 to 10 degrees) to the full open position. A test was also run with the one piece shaft design with the disc held in a stationary position. This was done to provide direct correlation with the model tests which were done in this manner. It also allowed a comparison to the torques measured during the dynamic test with the shaft connected to the pneumatic actuator. A summary of the operability test is included in Appendix B.

5.3 Application of Model Aerodynamic Test To Full Size Valve Operability

5.3.1 Valve Operating Times Expected In Service

All valves are designed to close within 5 seconds for flow conditions produced by the maximum differential pressure of 56 PSIG when 100 PSIG is released from the actuator air cylinder. The valves will close under these installed conditions due to the fact that the operator output torque (spring torque) and the valve aerodynamic torque are tending to close the valve at all disc angles for LOCA conditions. (See Table 2 and Tables 10 & 11.) While not required for LOCA, to open the valve under the above conditions, 6120 in-lb of torque is required to crack the disc off the seat, and 5556 in-lb max is required to hold the valve disc open. (See Table 7) The air torque of the actuator (valve open direction) is rated at 9055 in-lb @ 80 PSIG, and therefore is more than adequate for the required worst case operating conditions.

In the Vought Test (Reference 7.0-B) closing times were shown to improve slightly with flow through the valve. The conduct of the test would suggest that opening times in actual service might be retarded about .3 to .5 seconds and closing times might be improved by the same amount under maximum differential pressure conditions relative to the Clow bench test data.

5.3.2 Aerodynamic Torques And Mass Flow Rates For Valves As-Installed

As described in Section 5.1, torques from straight line model tests can be used to predict full size valve torques by D^3 scaling. Tables 7 thru 9 present torque and other data for the subject valves at various operating conditions. The item of concern for valve operability is TQ (for normal operating conditions, open cycle) and TQA (for maximum operating conditions, closing cycle). All positive torque values tend to close the valve. The meanings of the other listings can be found in 7.0 Reference C-1.

To obtain torque conditions for the as installed valves a judgement must be made as to what set of test data most nearly represents the actual conditions. The eight valves for this job are installed in series pairs approximately 2 pipe diameters apart with the shaft side of the units facing upstream into the postulated LOCA flow.

Model test simulating this two valve in series configuration, as well as single valve tests, were run to determine the aerodynamic torque and mass flow rates for incompressible flow in straight pipeline. (See Reference 7.0, C-2) The applicable conclusions that can be drawn from these model tests are:

1. The aerodynamic torque coefficient (see Reference 7.0, C-2 for definition of this term) for the downstream test valve was a maximum when the upstream test valve was fully open and produced the least obstruction to flow. This maximum was less than the maximum torque coefficient of similar single test valve experiments.

2. The torque coefficient on the downstream test valve became independent of the back pressure ratio as the upstream valve-disc approached the fully closed position and was relatively constant near zero for the experiments in which the downstream valve-disc opening angle was larger than the upstream valve-disc opening angle. The pressure distribution on the downstream valve-disc approached a constant value.
3. The torque coefficients for the upstream and downstream test valves were generally independent of the spacing between the test valves and the relative orientation.
4. The torque coefficient on the upstream test valve became relatively constant and nearly zero as the downstream test valve approached the fully closed position for experiments in which the upstream valve-disc opening angle was substantially greater than the downstream valve-disc opening angle.
5. A modified torque coefficient was utilized for the upstream test valve based upon a local characteristic pressure drop and was found to be nearly the same as the single test valve torque coefficient for corresponding valve-disc opening angles and back pressure ratios.
6. The maximum value of the torque coefficient for either the upstream or the downstream test valve was always less than the maximum value determined for the single test valve case at the same back pressure ratio.

7. The mass flowrate coefficient for the single test valve experiments increased as the back pressure ratio or valve-disc opening angle increased, since each decreased the losses due to the expansion process.

8. The mass flowrate coefficient for the two test valves in series experiments was similar to the single test valve values for the cases in which the valve-disc opening angle for the single test valve was equal to the lesser of the valve-disc opening angles for the two test valves in series.

9. An upstream valve-disc opening angle greater than 40^{o} resulted in mass flowrate coefficients for the two test valves in series experiments which were consistently less than the corresponding single test valve values.

10. The mass flowrate coefficient of the single test valve experiments was improved by the addition of the second test valve in those cases where the upstream valve-disc opening angle was less than 40^{o}, probably due to a change in the effective flow area related to the expansion processes.

11. Overall, the static pressure drop across the test section has more of an influence on the mass flowrate through the test section than on the aerodynamic torque on the valve discs.

12. The test valve spacing and orientation were found to have negligible effects upon the mass flowrate coefficients for the two test valves in series experiments.

13. The performance of the two test valve system generally was strongly dependent upon the lesser of the two valve-disc opening angles; thus if the two test valves in series were operated independently, the control of the flow would transfer between the two test valves.

14. Immediate reduction of the mass flowrate, if desired, would be achieved by operating the system such that the downstream test valve approached the fully closed position in advance of the upstream test valve.

15. If a minimum torque was desired on both test valves during the closing process, then the two test valves should be closed simultaneously.

Although the experiments which investigated the torque characteristics of the upstream test valve were limited, the results were expected to be similar to the torque characteristics of the single test valve experiments which operated under a variable back pressure ratio. The presence of the downstream test valve was expected to influence the performance of the upstream test valve by altering the back pressure into which the upstream test valve exhausts. The torque coefficients for the upstream test valve as a function of the downstream valve-disc opening angle were run (see Reference 7.0. C-2). The results include various upstream valve-disc opening angles (a_1 = 40°, 60°, and 80°), a back pressure ratio of 0.33, and test valve spacings of 2D and 4D. The torque coefficients for the upstream test valve for all of the experiments were equal or less than the corresponding values for the single test valve experiments.

In two of the as-installed pair of valves, a tee is piped in between. The tee branch is a 4 inch diameter connection used for nitrogen purging and the other ends are 8 inches in diameter for mounting into the piping. From information provided by the Bechtel Corporation, this 4 inch line will not be used during LOCA conditions. While the 4 inch branch will provide a static blocked off opening in the main 8 inch pipeline, no adverse affects on either the flow or the torque should occur as a result of this opening.

In light of the above considerations, then, using the torque coefficients for a single valve in a straight pipe run will be conservative since the maximum value of the torque coefficent for either the upstream or downstream valve in series is always less than the maximum value determined for the single test valve case at the same back pressure ratio.

The mass flow rate calculated for a single valve in a straight pipe run will also be valid because the calculations will be based on the valve disc opening angle equal to the lesser of the valve disc opening angles for the two valves in series. The test valve spacing and orientation were found to have negligible effects upon the mass flow rate coefficients for the two test valve in series experiments. For the normal operating pressure of 1.5 PSID, any adverse torque or mass flow effects due to the valves being paired in series are at worst negligible because the tests were conducted at a much higher ΔP.

Tables 10 and 11 shcw model test valve angle and actual valve angle for the supplied units vs torque. There is a 10° difference between these due to the seat angle design differences explained in previous sections. It is reasonable to expect all angles over 20° to be a proper representation of the magnitude and direction of torques. At 20° or below, the magnitudes may differ but the direction is correctly indicated. Since peak torques occur in the 60 to 80° range, these low end torques are of no consequence.

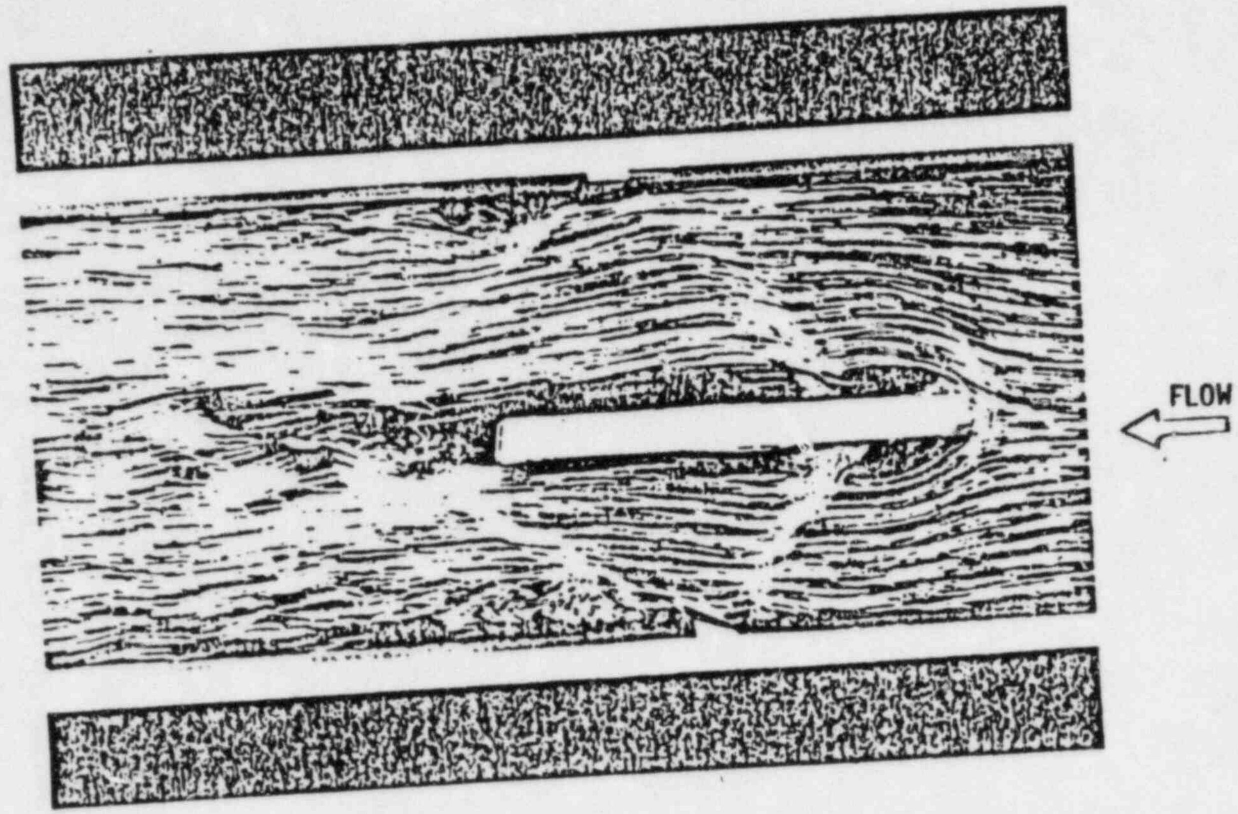


Figure 14 Single test valve water table experiments with a valve-disk opening angle of 80° and a back pressure ratio of 0.45.

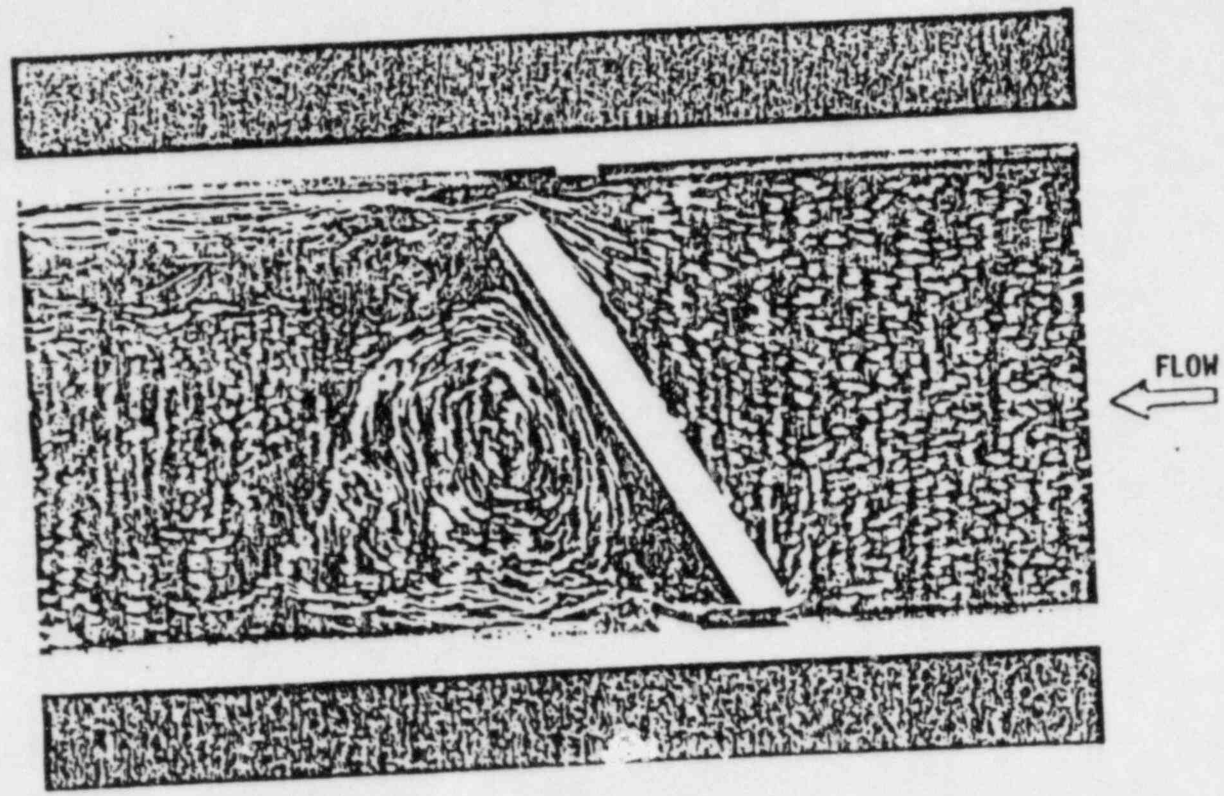


Figure 15 Single test valve water table experiments with a valve-disk opening angle of 20° and a back pressure ratio of 0.45.

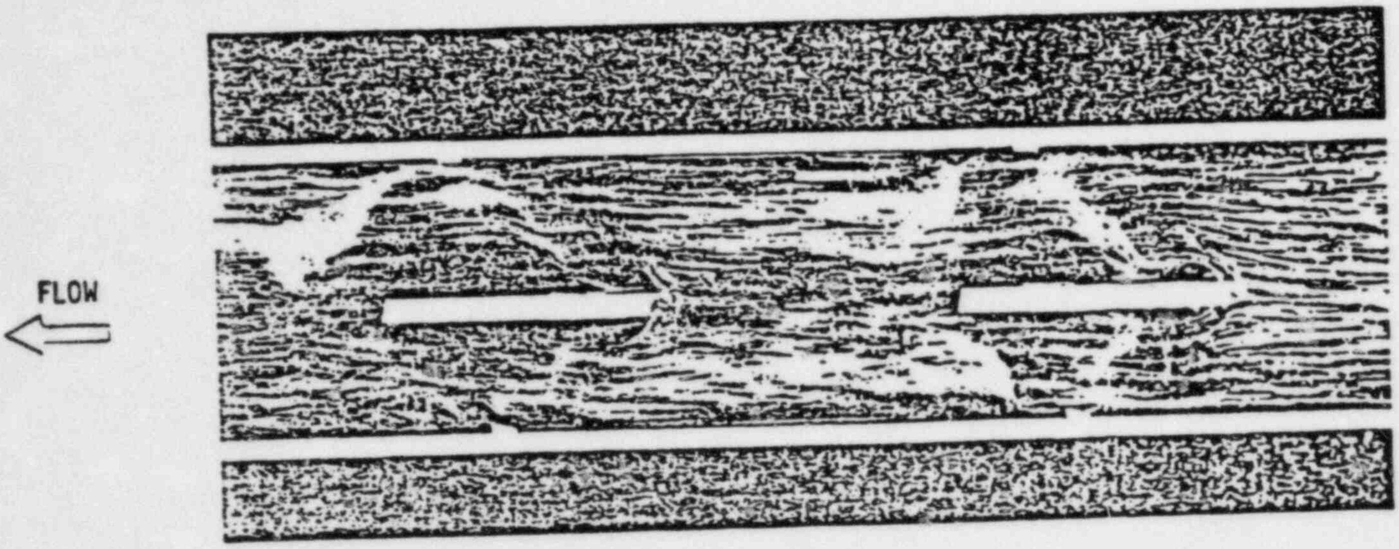


Figure 16 Two test valve water table experiments with an upstream valve-disk opening angle of 80°, a downstream valve-disk opening angle of 80°, Orientation 1, and a back pressure ratio of 0.45.

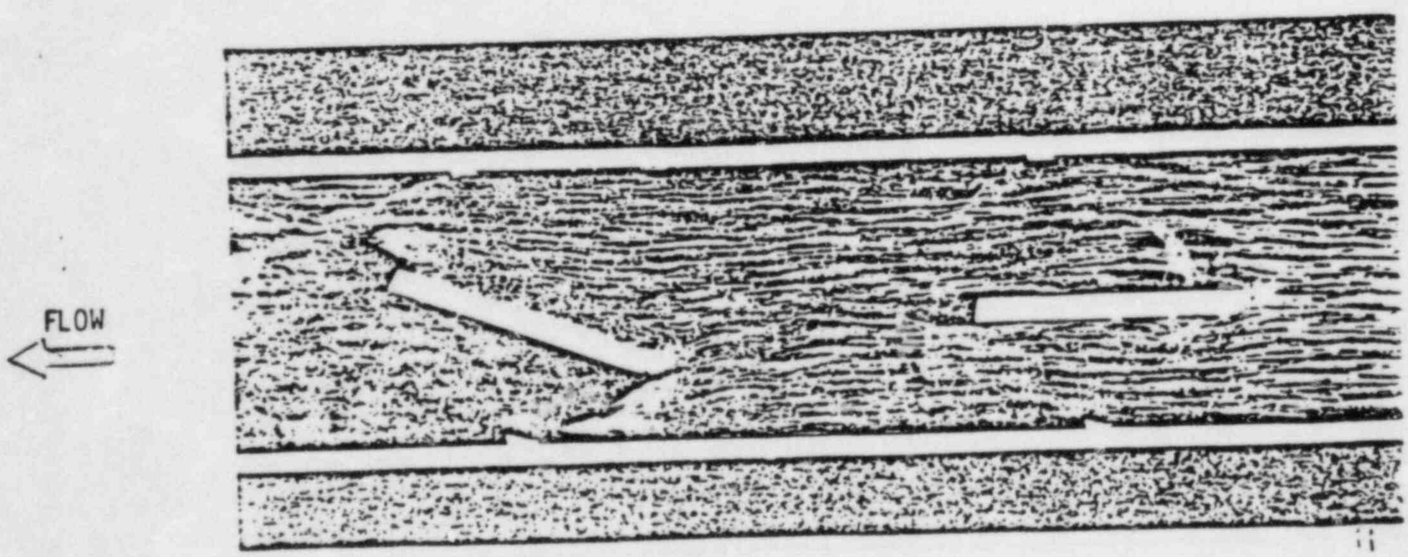


Figure 17 Two test valve water table experiments with an upstream valve-disk opening angle of 80°, a downstream valve-disk opening angle of 60°, Orientation 1, and a back pressure ratio of 0.45.

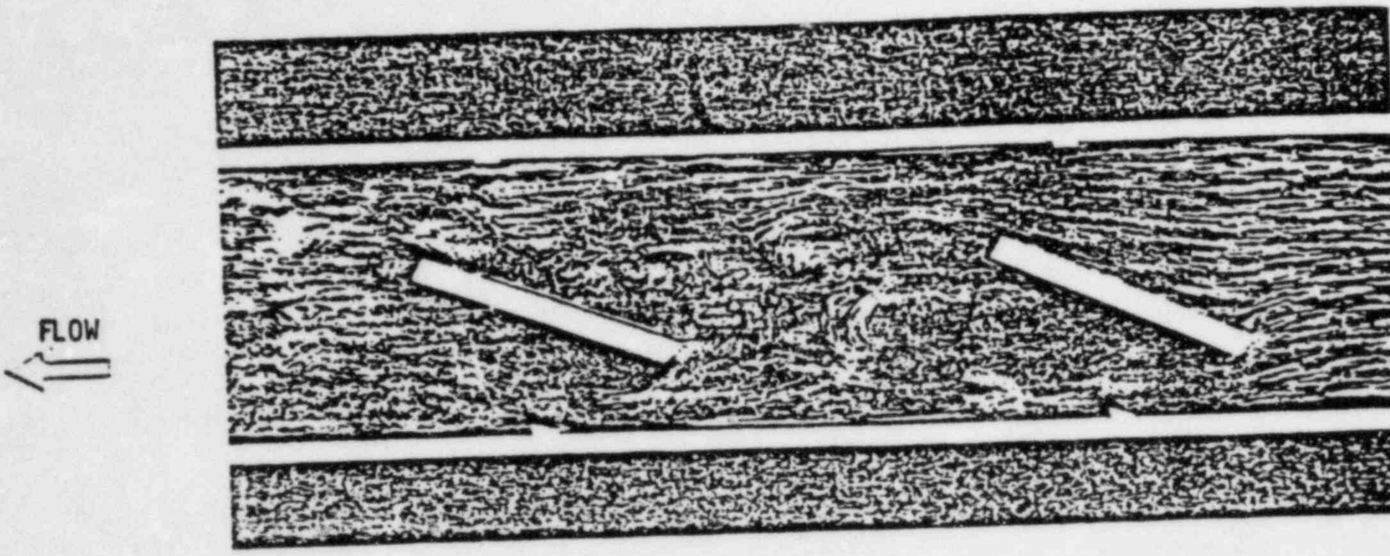


Figure 18 Two test valve water table experiments with an upstream valve-disk opening angle of 60°, a downstream valve-disk opening angle of 60°, Orientation 1, and a back pressure ratio of 0.45.

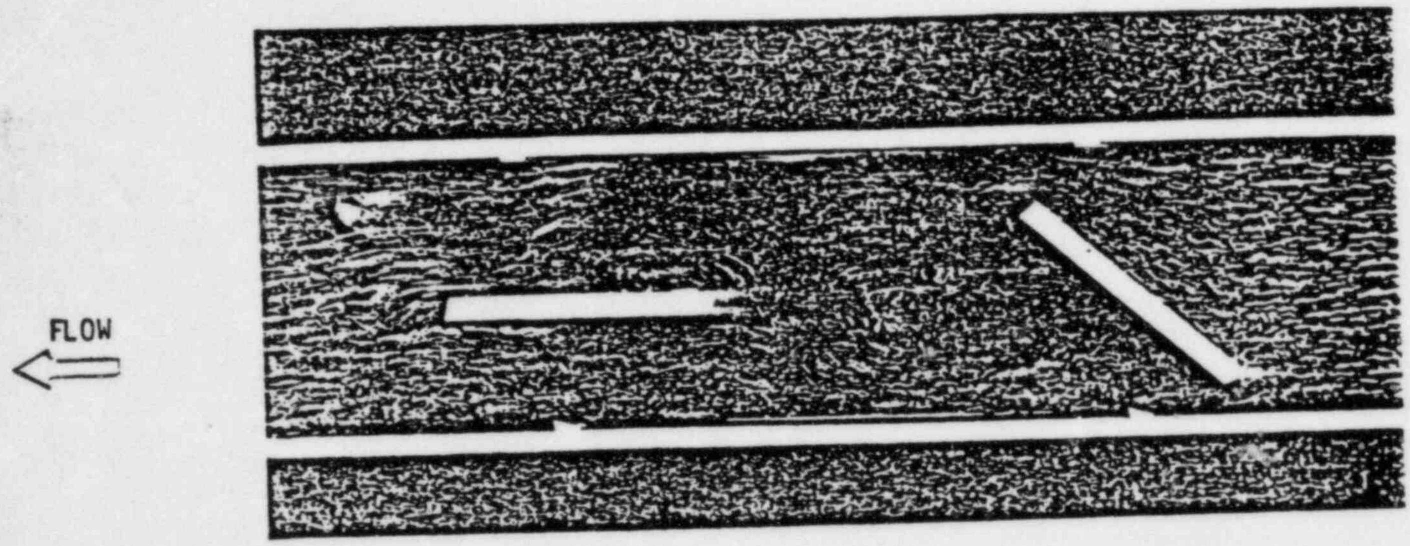


Figure 19 Two test valve water table experiments with an upstream valve-disk opening angle of 40°, a downstream valve-disk opening angle of 80°, Orientation 1, and a back pressure ratio of 0.45.

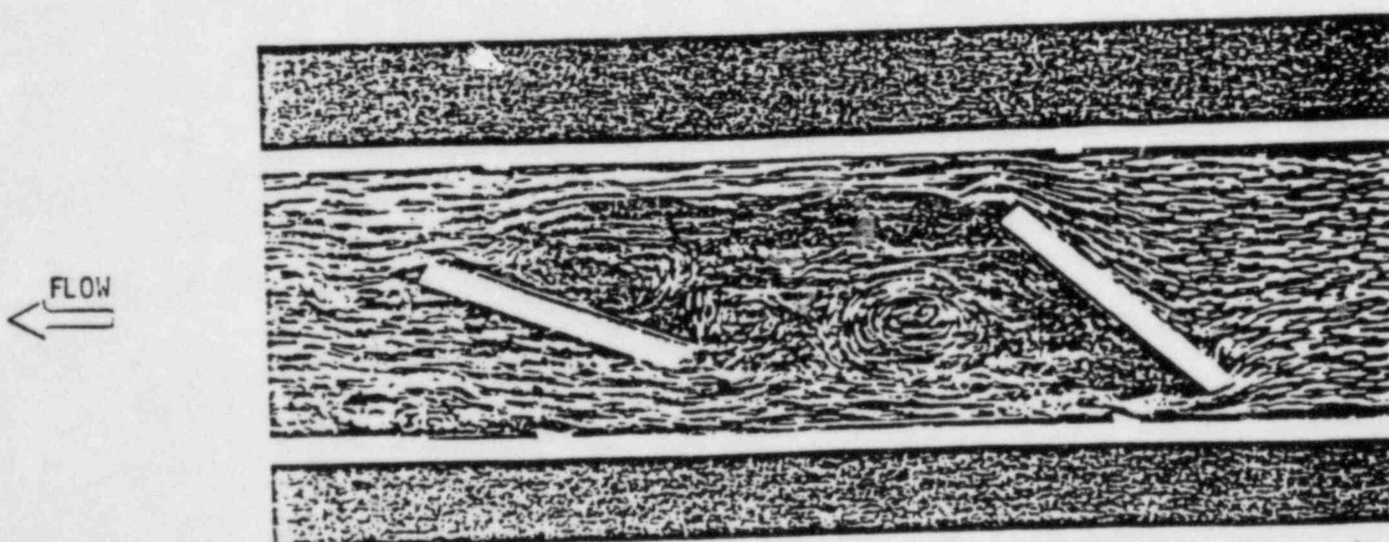


Figure 20 Two test valve water table experiments with an upstream valve-disk opening angle of 40°, a downstream valve-disk opening angle of 60°, Orientation 1, and a back pressure ratio of 0.45.

CHARACTERISTICS STRAIGHT PIPE RUN

CASE: BECHTEL PILGRIM EMERGENCY FLOW
UNITS SYSTEM: ES
SHAFT: US

DATE: 11-10-83
PATM: 14.70(PSIA)
PSU = 70.70(PSIA) TSU = 809.67(R)
MEDIUM: GAS = 4 GAMMA = 1.40 MW = 29.0
FLOW = CF OPTION = 2
DV = 8.000(IN)

OUTPUT DATA

CHOKING PRESSURE RATIOS: PSC/POU = .748 DPS/PSU = .198
SOLUTION: W80 = 36.42(LBM/S)

NOTE: TO BASED ON DIFFERENTIAL PRESSURE AT ONSET OF CHOKED FLOW
TQA BASED ON PSU UPSTREAM AND PATM DOWNSTREAM

PSD/POU = .7481

ALPHA	CF	WR	DPS/PSU	PSU/POU	PSC/POU	POD/POU	TQR1
80.0	.5162	1.0000	.1979	.9327	.7481	.8620	.0864
75.0	.5066	.9814	.2002	.9353	.7464	.8512	.1262
70.0	.4875	.9444	.2046	.9405	.7431	.8394	.1548
65.0	.4604	.8918	.2103	.9473	.7374	.8272	.1737
60.0	.4266	.8264	.2168	.9552	.7294	.8149	.1842
55.0	.3877	.7510	.2235	.9633	.7188	.8030	.1879
50.0	.3449	.6682	.2298	.9712	.7056	.7918	.1861
45.0	.2999	.5808	.2354	.9734	.6897	.7816	.1801
40.0	.2539	.4917	.2403	.9846	.6715	.7726	.1712
35.0	.2084	.4036	.2442	.9897	.6514	.7651	.1606
30.0	.1648	.3193	.2471	.9936	.6301	.7590	.1497
25.0	.1246	.2414	.2492	.9963	.6085	.7543	.1396
20.0	.0892	.1729	.2505	.9981	.5879	.7511	.1314
15.0	.0601	.1164	.2513	.9992	.5698	.7492	.1262
10.0	.0385	.0747	.2517	.9996	.5556	.7485	.1252
5.0	.0261	.0505	.2519	.9998	.5470	.7487	.1295

ALPHA (DEG)	YCV (...)	W (LBM/HR)	TO (IN-LBF)	TQA (IN-LBF)
[illegible]	[illegible]	[illegible]	612.59	2476.31
[illegible]	[illegible]	[illegible]	917.34	3629.10
[illegible]	[illegible]	[illegible]	1156.07	4476.03
[illegible]	[illegible]	[illegible]	1343.1[illegible]	[illegible]
[illegible]	[illegible]	[illegible]08348.34	1400.36	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	1611.61	[illegible]
[illegible]	[illegible]	[illegible]	1609.73	[illegible]
40.0	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	4271.51
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	[illegible]	[illegible]	[illegible]
[illegible]	[illegible]	9788.00	[illegible]	[illegible]
[illegible]	[illegible]	6624.13	1264.57	[illegible]

--------- COMPUTATIONS COMPLETE ---------

PIPE RUN (SHAFT UPSTREAM)

CASE: BECHTEL PILGRIM NORMAL FLOW
DATE: 11-10-83
PATM: 14.70(PSIA)
PSU = 16.20(PSIA)
MEDIUM: GAS = A
FLOW = UF
DV = 8.000(IN)
DPSO = 1.50(PSID)

TSU = 521.67(R)
GAMMA = 1.40
OPTION = 3

UNITS SYSTEM: ES
SHAFT: US

MW = 29.0

OUTPUT DATA

SOLUTION: WSO = 6.65(LBM/S)

PSD/POU = .9816

ALPHA	CF	WR	DPS/PSU	PSU/POU	PSC/POU	POD/POU	TQR1
80.0	.5162	1.0000	.0926	.9716	.7431	.9340	.0950
75.0	.5066	.9814	.0936	.9727	.7464	.9268	.1319
70.0	.4875	.9444	.0955	.9747	.7431	.9195	.1637
65.0	.4604	.8918	.0981	.9775	.7374	.9125	.1829
60.0	.4266	.8264	.1011	.9808	.7294	.9058	.1937
55.0	.3977	.7510	.1042	.9842	.7188	.8996	.1977
50.0	.3449	.6682	.1073	.9875	.7056	.8942	.1761
45.0	.2999	.5808	.1100	.9906	.6897	.8896	.1901
40.0	.2537	.4917	.1124	.9933	.6715	.8860	.1816
35.0	.2084	.4036	.1144	.9955	.6514	.8834	.1713
30.0	.1648	.3193	.1159	.9972	.6301	.8820	.1604
25.0	.1246	.2414	.1170	.9984	.6035	.8817	.1501
20.0	.0892	.1729	.1177	.9992	.5879	.8825	.1422
15.0	.0601	.1164	.1181	.9996	.5698	.8844	.1371
10.0	.0385	.0747	.1183	.9998	.5556	.8873	.1361
5.0	.0261	.0505	.1186	.9999	.5470	.8910	.1404

ALPHA (DEG)	YCV (...)	W (LBM/HR)	TQ (IN-LBF)
80.0	1074.17	23923.47	72.96
75.0	1047.22	23478.27	104.07
70.0	995.49	22593.77	130.11
65.0	924.72	21334.04	149.72
60.0	841.37	19771.25	163.77
55.0	750.50	17965.5[illegible]	172.23
50.0	[illegible]	[illegible]	177.27
45.0	[illegible]	10373.[illegible]	177.0[illegible]
40.0	[illegible]	11764.[illegible]	[illegible]
35.0	[illegible]	[illegible]	[illegible]
30.0	[illegible]	[illegible]	[illegible]
25.0	[illegible]	[illegible]	149.37
20.0	[illegible]	[illegible]	142.71
15.0	97.30	[illegible]	139.11
10.0	63.76	1706.10	137.37
5.0	[illegible]	[illegible]	[illegible]

------ COMPUTATIONS COMPLETE---------

TABLE 9 NORMAL FLOW CHARACTERISTICS, STRAIGHT PIPE RUN
(SHAFT DOWNSTREAM)

CASE: BECHTEL PILGRIM NORMAL FLOW
DATE: 11-10-83 UNITS SYSTEM: FS
PATM: 14.70(PSIA) SHAFT: DS
PSU = 16.20(PSIA) TSU = 524.67(R)
MEDIUM: GAS = 0 GAMMA = 1.40 MW = 29.0
FLOW = NF OPTION = 3
DV = 8.000(IN)
DPBO = 1.50(PSID)

OUTPUT DATA

SOLUTION: WBO = 7.26(LBM/S)

PSD/POU = .5767

ALPHA	CF	WR	DPS/PSU	PSU/POU	PSD/POU	POD/POU	TQR1
80.0	.5519	1.0000	.0926	.9662	.7532	.9212	-.0444
75.0	.5467	.9942	.0930	.9666	.7528	.9087	-.0638
70.0	.5329	.9655	.0949	.9685	.7507	.8961	-.0776
65.0	.5063	.9174	.0978	.9717	.7464	.8835	-.0869
60.0	.4709	.8531	.1014	.9756	.7396	.8710	-.0923
55.0	.4265	.7763	.1053	.9799	.7299	.8587	-.0955
50.0	.3910	.6903	.1092	.9842	.7169	.8468	-.0964
45.0	.3304	.5987	.1123	.9882	.7007	.8353	-.0960
40.0	.2786	.5047	.1159	.9916	.6815	.8243	-.0945
35.0	.2273	.4117	.1184	.9944	.6600	.8139	-.0926
30.0	.1776	.3225	.1203	.9966	.6370	.8039	-.0903
25.0	.1341	.2430	.1216	.9981	.6133	.7945	-.0878
20.0	.0957	.1734	.1224	.9990	.5916	.7855	-.0853
15.0	.0650	.1178	.1229	.9995	.5730	.7770	-.0824
10.0	.0439	.0792	.1231	.9993	.5591	.7690	-.0791
5.0	.0334	.0605	.1232	.9999	.5523	.7612	-.0751

ALPHA (DEG)	YCV (...)	W (LBM/HR)	TQ (IN-LBF)
80.0	1174.97	26140.10	-34.03
75.0	1185.32	25997.09	-49.20
70.0	1118.05	25247.06	-61.15
65.0	1040.04	23937.34	-70.52
60.0	943.64	22307.93	-73.57
55.0	842.76	20099.12	-84.62
50.0	733.12	18051.11	-88.99
45.0	[illegible]	15603.9[illegible]	-91.00
40.0	[illegible]	[illegible]	-93.24
35.0	[illegible]	[illegible]	-93.[illegible]
30.0	[illegible]	8479.[illegible]	-92.07
25.0	[illegible]	[illegible]	-91.47
20.0	[illegible]	[illegible]	-89.[illegible]
15.0	116.19	3001.45	[illegible]
10.0	73.12	2071.31	[illegible]
5.0	75.50	1000.27	-77.37

--------COMPUTATIONS COMPLETE--------

TABLE 10

Torque for as installed conditions for valves;
AO-5035A, AO-5035B, AO-5044A, AO-5044B.

All torques in in-lb: (Positive torque tends to close valve)

MODEL TEST VALVE ANGLE	ACTUAL VALVE ANGLE	TORQUE FOR INSTALLED CONDITION NORMAL*	MAXIMUM**
80	90	73	2477
70	80	130	4476
60	70	164	5410
50	60	177	5556
40	50	173	5180
30	40	158	4572
20	30	142	4030
10	20	137	3848

*At 1.5 PSID
**At 56 PSID

TABLE 11

Torque for as installed conditions for valves;
AO-5036A, AO-5036B, AO-5042A, AO-5042B.

All torques in in-lb: (Positive torque tends to close valve, negative torque tends to open valve.)

MODEL TEST VALVE ANGLE	ACTUAL VALVE ANGLE	TORQUE FOR INSTALLED CONDITION NORMAL*	MAXIMUM**
80	90	-34	2477
70	80	-61	4476
60	70	-79	5410
50	60	-89	5556
40	50	-93	5180
30	40	-93	4572
20	30	-89	4030
10	20	-84	3848

*At 1.5 PSID
**At 56 PSID

5.3.3 Conclusions Concerning Valve Operability

For a LOCA condition it can be seen in Tables 10&11 that torques for the subject valves are positive (closing) torques for all disc positions. For these valves, any flow condition from none to maximum, in combination with the timed bench tests show the valve will close within 5 seconds or less. As shown in Section 5.3.1, the valves will operate in both the open and closed directions under fully developed LOCA parameters.

For the presented data and supplemental test reports, it has been shown that the valves will operate as designed under the prescribed conditions. This has been shown using the conservative assumption of no credit taken for pressure ramp in containment and no credit taken for back pressure due to downstream piping.

The critical elements of the valve, except for the dowel pin and keys, were shown to be well within code limits for a design seating torque of 9056 in lb. per the torque data submitted and summarized in Table 2. Torque values to the keys could reach 11,032 in.lbs. The keys are sized based on a minimum tensile strength of 85,000 PSI at 350°F (actual tests for typical material show 100,000 PSI yield at room temperature), a shear value of 60% of tensile and a 3.0 or better safety factor for internal keys (actuator key has 2.5 safety factor). For the subject valves with a shear area of 1.69 sq. in. for the disc keys and 1.125 for the actuator key, the following stress values result.

Ear key stress = 8703 PSI   Actuator key = 13,074 PSI

The dowel pin is designed only to resist shaft end pressure load. The resultant dowel pin stress for 56 PSIG internal valve pressure is:

Dowel pin stress = 1135 PSI

The dowel pin is made of alloy steel with a minimum tensile strength of 110,000 PSI. For the shaft the maximum stress for the given torque is within code limits (See References 7.0, A1,2)

5.3.4 NRC 21 Questions

Clow has pursued an extensive program to demonstrate operability of purge and vent valves in accord with NRC Guidelines. Since every installation is unique, Clow's basic approach is to use a combination of test and analysis data. The following pages give an item by item response to the 21 point (less 2) list of considerations issued by the NRC to utilities. These responses include descriptions of such tests. A copy of the NRC questions responded to in this paper is attached (Appendix A).

1. The ΔP across the valve is determined from the customer's spec and/or data sheet. Clow assumes downstream pressure is atmospheric although it may, in fact, be higher.
2. Dynamic torque coefficients were developed based on scale models of a 12", 24", 48", and 96" valve. These were shown to be conservative by a test of a full scale 12" valve. Further, model tests were performed for an upstream mitered elbow for 12", 24", and 48" models and for 2 valves in series using the 24" models. For actual production valve disc shapes are identical or only

slightly different. All differences, although small, are fully documented. (Section 5.1, 5.2, 5.3)

3. Installation effects were accounted for in all cases, but downstream piping back pressure was not, since this produces a more conservative calculation. (Section 5.1.3)

4. Clow does not consider containment pressure response profile. Clow assumes signal may be delayed until full containment pressure is reached, then the valve will be called upon to close. Time lag for equipment response is not considered by Clow since this is the responsibility of the buyer. Clow does however, record test lag time as part of unit bench testing. (Section 1.2.C, 5.3.1, 5.3.3)

5. Valve angle and predicted ΔP for choking across the valve is presented, however due to piping considerations, some degree of question as to their validity to actual operation is present. The maximum ΔP for all angles. (Section 5.3.2)

6. Codes used, allowed stresses, and predicted stresses are presented in the Code Design Report and/or Seismic Analysis Report(s). Load combinations are described in these reports. The valve is analyzed by finite element techniques. (Section 4.0)

9. Unless indicated to Clow in the customer spec, no back pressure is considered. When back pressure requirements are specified, Clow considers it and incorporates it as part of the unit bench test.
10. Clow to date has not used accumulators for valves used in containment isolation system service.
11. NA to Clow design.
12. Units are not modified to limit the travel angle, except by actuator stops or limit switch settings. Such settings are made to limit travel of the disc to being parallel with the pipe centerline. Clow's report shows, from the test data base and bench tests of each unit, that sufficient torque is available to close and seat the valve against all flow induced loads. Since Clow's seat/seal design is conical, no special considerations for low heat temperature is required. (See Section 5.3.1, 5.3.2)
13. Clow selects operators for each unit with maximum operating torques much larger than that produced by flow interaction with the disc. (See Section 5.3.1)
14. Not applicable to air operators.
15. Not applicable to air operators.
16. Not applicable to air operators. Where a manual jack screw is provided, the unit is tagged indicating the full disengagement length of the screw. No <u>automatic</u> features are provided to insure disengagement. Proper operation is the responsibility of the user.

17. The valve, being of all metal construction except for packings, seal laminations, and gaskets, will not degrade under the required environmental conditions. Metal components are generally accepted in the industry as suitable for the required environmental conditions. Tests at both high and low temperatures have been performed by Gebruder Adams of Bokum, West Germany for the subject seal/seat design. Seismic considerations are covered by both analysis and previous static load tests. (See Section 1.2, Section 6.0, Section 7.0 A,B).
18. All operators and solenoid valves installed by Clow are qualified to appropriate IEEE requirements by testing. (See Section 2.2.2, 2.2.3).
19. All tests are summarized in the supplied qualification report and are documented by separate test reports. (See Section 7.0)
20. Assumptions and the basis for use of analysis combined with test data are presented in the report. (All Sections)
21. Clow provides operation and maintenance manuals describing required maintenance intervals (typically replacement at least every 5 years on all elastomers).

6.0 VALVE SEALING CHARACTERISTICS

6.1 Normal Sealing

The following chart shows the sealing ability of the valves as they were shop tested for record. The tests were performed with pressure on the indicated side of the disc and the opposite side open to atmosphere. The normal recommended flow direction for these valves is with pressure on the shaft side, so when pressure is applied to the clamp ring side, it is considered to be the reverse flow direction. During this test, the air under water method was used to indicate leakage.

Table 12

VALVE SEALING CHARACTERISTICS

VALVE MARK NO.	VALVE SIZE (IN.)	TEST PRESSURE PSIG	PRESSURIZED SIDE SHAFT SIDE	PRESSURIZED SIDE CLAMP RING SIDE	LEAKAGE (BUBBLES/MIN)
8-HBB-BF-AO-5035A	8	56	X		0
		25	X		0
		5	X	X	0
		56		X	0
		25		X	0
		5			0
8-HBB-BF-AO-5035B	8	56	X		0
		25	X		0
		5	X		0
		56		X	0
		25		X	0
		5		X	0
8-HBB-BF-AO-5036A	8	56	X		.25
		25	X		0
		5	X		0
		56		X	0
		25		X	0
		5		X	0
8-HBB-BF-AO-5036B	8	56	X		0
		25	X		0
		5	X		0
		56		X	0
		25		X	0
		5		X	0
8-HBB-BF-AO-5042A	8	56	X		0
		25	X		0
		5	X		0
		56		X	0
		25		X	0
		5		X	0
8-HBB-BF-AO-5042B	8	56	X		0
		25	X		0
		5	X		0
		56		X	0
		25		X	1.75
		5		X	0
8-HBB-BF-AO-5044B	8	56	X		0
		25	X		0
		5	X		0
		56		X	0
		25		X	0
		5		X	.25
8-HBB-BF-AO-5044A	8	56	X		0
		25	X		0
		5	X		0
		56		X	0
		25		X	0
		5		X	0

6.2 Long Term Sealing

The conical seal/seat design of the Tricentric valve in combination with the laminated metal/asbestos seal offers good long term sealing characteristics. When the seal and seat are machined a certain surface finish is obtained. With this finish certain leak rates are obtained during a bench test (see 6.1). On a microscopic scale these surfaces contain peaks and valleys. When the disc is seated, these surfaces mate and high local (above yield) stresses are induced at the peaks. The peaks will yield and deform and form a match between the seat and seal. As the valve is cycled throughout its life, this match tends to improve and a visual seating pattern appears. This results in improved sealing as the valve ages.

This has been verified by experience and is documented in the Shell International Cycling Test (Reference 7.0 D-3). This test was performed by Gebruder Adams of Bochum, West Germany. Clow's Engineered Products Division produces the Tricentric design under license of Gebruder Adams. The test showed sealing improved continuously up to 41,000 cycles, the limit of the test.

6.3 Debris Effects On Sealing

A test was performed to determine the effect on sealing capability of a Tricentric valve if a foreign object became trapped between the seat and seal. As with any valve, if the object is large enough and hard enough and happens to be caught between the sealing surfaces, the valve will fail to close completely and the valve will leak. Leakage will be dependent on the size and shape of the object and open gap size which remains when the valve does not fully close. Since no standards as to debris size exist, the test made determined leakage due to object damage after the object was removed. For in plant operation this would represent leakage after recycling of the valve if the object was blown out of the way during recycling.

The object selected was a cooling tray liner used in the petrochemical industry. It's dimensions were approximately 1/8" x 1" x 6" and was a filled polyvinyl chloride plastic of 80 shore D hardness. The valve was closed upon this material, opened to remove the material, then closed again to measure leakage. Depending on the applied seating torque, a leakage of .015 SCFM to .333 SCFM was measured. This test showed the valve could tolerate some large debris and still maintain a relatively low leakage even with a damaged seal (See reference 7.0 D-2.)

6.4 Sealing Under Temperature Variations

The Tricentric design has been used successfully for sealing applications from cryogenic to 900°F. The Shell International Cycling Test describes sealing characteristics for a media operating temperature of 842°F when the body reached a temperature of 716°F.

The Tricentric conical seal/seat design lends itself well to accommodating temperature changes in the body and resultant size variation of the sealing components. Due to the torque seating design and some seal flexibility, the valve will self adjust to the small dimensional variations which could be anticipated for the subject valves. Of course, if large thermal gradients (very unlikely from information provided to Clow) existed around the body circumference higher levels of leakage could be expected. Again no standards exist to the knowledge of Clow personnel which could become a basis for prediction or a test of such leakage.

7.0 REFERENCES (con't)

C. Air Flow Tests

prepared by: A.L. Addy, Ph.D.
Urbana, Illinois
(Engineering Consultant in Fluid Dynamics)

1. Final report on the Clow Valve Analysis Program CVAP (Oct. 1981). Report covers methods of analysis, development of data base from model tests, and set-up of computer program to predict characteristics of full size valves.

2. "Aerodynamic Torque And Mass Flow Rate For Compressible Flow Through Geometrically Similar Scale-Model Clow Valves In Series." (October, 1982)

D. Other Reports and Information

1. Operating Instructions for Clow Tricentric Wafer Stop Valve covers installation, maintenance, and operating instructions for 82-2739(N) valves.

2. Clow Test Report Project No. 82-003 "Effects of Foreign Bodies on Tricentric Sealing" by Robert Sansone.

3. Shell International Cycling Test (2/6/72) by M. Nijenhuis (Note: Clow produces Tricentric valves under license of Gebruder Adams of Bochum, West Germany.)

E. Other References

1. Bechtel Power Corp. Design Specification 10394-P-119-1(Q), Rev. 0.

2. "A Water Table Investigation of Two-Dimensional Models of The Clow Corporation Tricentric Valve" by Dr. Robert F. Hurt, Engineering Consultant, Professor of Mechanical Engineering, Bradley University, Peoria, Illinois, Sept. 14, 1979.

3. "A Parametric Study of A Butterfly Valve Utilizing The Hydraulic Anology" by Bruce A. Coers, Bradley University, 1983.

4. "Radiation Sensitivity Analysis of Luminated Valve Seals For Clow Corporation." Wyle No. 17629-01 (Jan. 31, 1983)

7.0 REFERENCES

A. Seismic Analysis Reports

prepared by: Patel Engineers
Huntsville, Alabama

The following include stress and frequency analysis for the subject valves:

1. Technical Report PEI-TR-83-24, Rev. A. Seismic Qualification Analysis of Clow 8-Inch Wafer Stop Valve.

2. Technical Report PEI-TR-833700-1, Addendum to PEI Technical Report PEI-TR-83-24 covering 8"-HBB-BF-AO-5035A, 5035B, 5036A, 5036B, 5042A, 5042B, 5044A, and 5044B.

B. Seismic Qualification Test Reports

prepared by: Vought Corp.
High Speed Wind Tunnel Facility
Dallas, Texas

1. Report No. 2-59700/1R-52972 "Simultaneous Static Seismic Load of Flow Interruption Capability Tests of a 12 Inch Valve for the Clow Corporation" (Dec. 15, 1981). Application of 11.0 g biaxial static load to valve actuator during operation with choked air flow thru the valve.

2. Patel Report PEI-TR-83-29, Revision A (Aug. 10, 1983) "Seismic Qualification of Clow Wafer Stop Valve Assemblies" including Addendum I and II.

APPENDIX A

NUCLEAR REGULATORY

PURGE VALVE

OPERABILITY

GUIDE LINES

BRANCH TECHNICAL POSITION CSB 6-4 *

CONTAINMENT PURGING DURING NORMAL PLANT OPERATIONS

A. BACKGROUND

This branch technical position pertains to system lines which can provide an open path from the containment to the environs during normal plant operation; e.g., the purge and vent lines of the containment purge system. It supplements the position taken in SRP section 6.2.4.

While the containment purge system provides plant operational flexibility, its design must consider the importance of minimizing the release of containment atmosphere to the environs following a postulated loss-of-coolant accident. Therefore, plant designs must not rely on its use on a routine basis.

The need for purging has not always been anticipated in the design of plants, and therefore, design criteria for the containment purge system have not been fully developed. The purging experience at operating plants varies considerably from plant to plant. Some plants do not purge during reactor operation, some purge intermittently for short periods and some purge continuously.

The containment purge system has been used in a variety of ways, for example, to alleviate certain operational problems, such as excess air leakage into the containment from pneumatic controllers, for reducing the airborne activity within the containment to facilitate personnel access during reactor power operation,

*Note: This paper is retyped for legibility from paper supplied by NRC.

and for controlling the containment pressure, temperature and relative humidity. However, the purge and vent lines provide an open path from the containment to the environs. Should a LOCA occur during containment purging when the reactor is at power, the calculated accident doses should be within 10 CFR 100 guideline values.

The sizing of the purge and vent lines in most plants has been based on the need to control the containment atmosphere during refueling operations. This need has resulted in very large lines penetrating the containment (about 42 inches in diameter). Since these lines are normally the only ones provided that will permit some degree of control over the containment atmosphere to facilitate personnel access, some plants have used them for containment purging during normal plant operation. Under such conditions, calculated accident doses could be significant. Therefore, the use of these large containment purge and vent lines should be restricted to cold shutdown conditions and refueling operations.

The design and use of the purge and vent lines should be based on the premise of achieving acceptable calculated offsite radiological consequences and assuring that emergency care cooling (ECCS) effectiveness is not degraded by a reduction in the containment pressure.

Purge system designs that are acceptable for use on nonroutine basis during normal plant operation can be achieved by

providing additional purge and vent lines. The size of these lines should be limited such that in the event of a loss-of-coolant accident, assuming the purge and vent valves are open and subsequently close, the radiological consequences calculated in accordance with Regulatory Guides 1.3 and 1.4 would not exceed the 10 CFR 100 guideline values. Also, the maximum time for valve closure should not exceed five seconds to assure that the purge and vent valves would be closed before the onset of fuel failures following a LOCA.

The size of the purge and vent lines should be about eight inches in diameter for PWR plants. This line size may be overly conservative from a radiological viewpoint for the Mark III BWR plants and the HTGR plants because of containment and/or core design features. Therefore, larger line sizes may be justified. However, for any proposed line size, the applicant must demonstrate that the radiological consequences following a loss-of-coolant accident would be within 10 CFR 100 guideline values. In summary, the acceptability of a specific line size is a function of the site meteorology, containment design, and radiological source term for the reactor type; e.g., BWR, PWR or HTGR.

B. BRANCH TECHNICAL POSITION

The system used to purge the containment for the reactor operational modes of power operation, startup, hot standby and hot shutdown; i.e., the on-line purge system, should be independent of the purge system used for the reactor operation modes of cold shutdown and refueling.

1. The on-line purge system should be designed in accordance with the following criteria:

   a. The performance and reliability of the purge system isolation valves should be consistent with the operability assurance program outlined in MEB Branch Technical Position MEB-2, Pump and Valve Operability Assurance Program. (Also see SRP Section 3.9.3) The design basis for the valves and actuators should include the buildup of containment pressure for the LOCA break spectrum, and the purge line and vent line flows as a function of time up to and during valve closure.

   b. The number of purge and vent lines that may be used should be limited to one purge line and one vent line.

   c. The size of the purge and vent lines should not exceed about eight inches in diameter unless detailed justification for larger line sizes is provided.

   d. The containment isolation provisions for the purge system lines should meet the standards appropriate to engineered safety features; e.e., quality, redundancy, reliability and other appropriate criteria.

   e. The instrumentation and control systems provided to isolate the purge system lines should be independent and actuated by diverse parameters; e.g., containment pressure, safety injection actuation, and containment radiation level. If energy is required to close the valves, at least two diverse sources of energy shall be provided, either of which can affect the isolation function.

f. Purge system isolation valve closure times, including instrumentation delays, should not exceed <u>five seconds</u>.

g. Provisions should be made to ensure that isolation valve closure will not be prevented by debris which could potentially become entrained in the escaping air and steam.

2. The purge system should not be relied on for temperature and humidity control within the containment.

3. Provisions should be made to minimize the need for purging of the containment by installing containment atmosphere cleanup systems within the containment.

4. Provisions should be made for testing the availability of the isolation function and leakage rate of the isolation valves, individually, during reactor operation.

5. The following analyses should be performed to justify the containment purge system.

   a. An analysis of the radiological consequences of a loss-of-coolant accident. An analysis should be done for a spectrum of break sizes, and the instrumentation and setpoints that will actuate the vent and purge valves closed should be specified. The source term used in the radiological calculations should be based on a calculation under the terms of Appendix K to determine the extent of a failure and the concomitant release of fission products, and the fission product activity in the primary coolant. A pre-existing iodine spike should

be considered in determining primary coolant activity. The volume of containment in which fission products are mixed should be justified, and the fission products from the above sources should be assumed to be released through the open purge valves during the maximum interval required for valve closure. The radiological consequences should be within 10 CFR 100 guideline values.

b. An analysis which demonstrates the acceptability of the provisions made to protect structures and safety-related equipment; e.g., fans, filters and ducting located beyond the purge system isolation valves against loss of function to control the environment created by the escaping air and steam.

c. An analysis of the reduction in the containment pressure resulting from the partial loss of containment atmosphere during the accident for ECCS backpressure determination.

d. The allowable leak rates of the purge and vent isolation valves should be specified for the spectrum of design basis pressures and flows against which the valves must close.

GUIDELINES FOR DEMONSTRATION OF OPERABILITY OF PURGE AND VENT VALVES

OPERABILITY

In order to establish operability it must be shown that the valve actuator's torque capability has sufficient margin to overcome or resist the torques and/or forces (i.e., fluid dynamic, bearing, seating, friction) that resist closure when stroking from the initial open position to full seated (bubble tight) in the time limit specified. This should be predicted on the pressure(s) established in the containment following a design basis LOCA. Considerations which should be addressed in assuring valve design adequacy include:

1. Valve closure rate versus time - i.e., constant rate or other.
2. Flow direction through valve; ΔP across valve.
3. Single valve closure (inside containment or outside containment valve) or simultaneous closure. Establish worst case.
4. Containment back pressure effect on closing torque margins of air operated valve which vent pilot air inside containment.
5. Adequacy of accumulator (when used) sizing and initial charge for valve closure requirements.
6. For valve operators using torque limiting devices - are the settings of the devices compatible with the torques required to operate the valve during the design basis condition.

7. The effect of the piping system (turns, branches) upstream and downstream of all valve installations.
8. The effect of butterfly valve disc and shaft orientation to the fluid mixture egressing from containment.

DEMONSTRATION

Demonstration of the various aspects of operability of purge and vent valves may be by analysis, bench testing, insitu testing or a combination of these means.

Purge and vent valve structural elements (valve/actuator assembly) must be evaluated to have sufficient stress margins to withstand loads imposed while valve closes during a design basis accident. Torsional shear, shear, bending, tension and compression loads/stresses should be considered. Seismic loadings should be addressed.

Once valve closure and structural integrity are assured by analysis, testing or a suitable combination, a determination of the sealing integrity after closure and long term exposure to the containment environment should be evaluated. Emphasis should be directed at the effect of radiation and of the containment spray chemical solutions on seal material. Other aspects such as the effect on sealing from outside ambient temperatures and debris should be considered.

The following considerations apply when testing is chosen as a means for demonstrating valve operability:

Bench Testing

A. Bench testing can be used to demonstrate suitability of the in-service valve by reason of its tracibility in design to a test valve. The following factors should be considered when qualifying valves through bench testing.

1. Whether a valve was qualified by testing of an identical valve assembly or by extrapolation of data from a similarly designed valve.

2. Whether measures were taken to assure that piping up-stream and downstream and valve orientation are simulated.

3. Whether the following load and environmental factors were considered

   a. Simulation of LOCA
   b. Seismic loading
   c. Temperature soak
   d. Radiation exposure
   e. Chemical exposure
   f. Debris

B. Bench testing of installed valves to demonstrate the suitability of the specific valve to perform its required function during the postulated design basis accident is acceptable.

1. The factors listed in items A.2 and A.3 should be considered when taking this approach.

In-Situ Testing

In-situ testing of purge and vent valves may be performed to confirm the suitability of the valve under actual conditions.

When performing such test, the conditions (loading, environment) to which the valve(s) will be subjected during the test should simulate the design basis accident.

NOTE: Post test valve examination should be performed to establish structural integrity of the key valve/actuator components.

End CSB 6-4

CLARIFICATION OF SEPT. 27 LETTER TO LICENSEES REGARDING *
DEMONSTRATION OF OPERABILITY OF PURGE AND VENT VALVES

1. The ΔP across the valve is in part predicated on the containment pressure and gas density conditions. What were the containment conditions used to determine the ΔP's across the valve at the incremental angle positions during the closure cycle?
2. Were the dynamic torque coefficients used for the determination of torques developed, based on data resulting from actual flow tests conducted on the particular disc shape/design/size? What was the basis used to predict torques developed in valve sizes different (especially larger valves) than the sizes known to have undergone flow tests?
3. Were installation effects accounted for in the determination of dynamic torques developed? Dynamic torques are known to be affected for example, by flow direction through valves with off-set discs, by downstream piping backpressure, by shaft orientation relative to elbows, etc. What was the basis (test data or other) used to predict dynamic torques for the particular valve installation?
4. When comparing the containment pressure response profile against the valve position at a given instant of time, was the valve closure rate vs. time (i.e. contstant or other) taken into account? For air operated valves equipped with spring return operators, has the lag time from the time the

* Note: This paper is retyped for legibility from paper supplied by NRC.

valve receives a signal to the time the valve starts to stroke been accounted for?

NOTE: Where a butterfly valve assembly is equipped with spring to close air operators (cylinder, diaphragm, etc.), there typically is a lag time from the time the isolation signal is received (solenoid valve usually deenergized) to the time the operator starts to move the valve. In the case of an air cylinder, the pilot air on the opening side of the cylinder is approximately 90 psig when the valve is open, and the spring force available may not start to move the piston until the air on this opening side is vented (solenoid valve de-energizes) below about 65 psig, thus the lag time.

5. Provide the necessary information for the table shown below for valve positions from the initial open position to the seated position (10° increments if practical).

Valve Position (in degrees - 90° = full open)	Predicted ΔP (across valve)	Maximum ΔP (capability)

6. What Code, standards or other criteria, was the valve designed to? What are the stress allowables (tension, shear, torsion, etc.) used for critical elements such as disc, pins, shaft yoke, etc. in the valve assembly? What load combinations were used?

9. For those valve assemblies (with air operators) inside containment, has the containment pressure rise (backpressure) been considered as to its effect on torque margins available (to close and seat the valve) from the actuator? During the closure period, air must be vented from the actuators opening

side through the solenoid valve into this backpressure. Discuss the installed actuator bleed configuration and provide basis for not considering this backpressure effect a problem on torque margin. Valve assembly using 4 way solenoid valve should especially be reviewed.

10. Where air operated valve assemblies use accumulators as the fail-safe feature, describe the accumulator air system configuration and its operation. Provide necessary information to show the adequacy ofthe accumulator to stroke the valve i.e. sizing and operation starting from lower limits of initial air pressure charge. Discuss active electrical components in the accumulator system, and the basis used to determine their qualification for the environmental conditions experienced. Is the accumulator system seismically designed?

11. For valve assemblies requiring a seal pressurization system (inflatable main seal) describe the air pressurization system configuration and operation including means used to determine that valve closure and seal pressurization have taken place. Discuss active electrical components in this system, and the basis used to determine their qualification for the environmental condition experienced. Is this system seismically designed.
For this type valve, has it been determined that the "valve travel stops" (closed position) are capable of withstanding the loads imposed at closure during the DBA-LOCA conditions.

12. Describe the modification made to the valve assembly to limit the opening angle. With this modification, is there sufficient torque margin available from the operator to overcome any dynamic torques developed that tend to oppose valve closure, starting from the valve's initial open position? Is there sufficient torque margin available from the operator to fully seat the valve? Consider seating torques required with seats that have been at low ambient temperatures.

13. Does the maximum torque developed by the valve during closure exceed the maximum torque rating of the operators? Could this affect operability?

14. Has the maximum torque value determined in #12 been found to be compatible with torque limiting settings where applicable?

15. Where electric motor operators are used, has the minimum available voltage to the electric operator under both normal or emergency modes been determined and specified to the operator manufacturer, to assure the adequacy of the operator to stroke the valve at DBA conditions with these lower limit voltages available. Does this reduced voltage operation result in any significant change in stroke timing? Describe the emergency mode power source used.

16. Where electric operator units are equipped with handwheels, does their design provide for automatic re-engagement of the motor operator following the handwheel mode of operation? If not, what steps are taken to preclude the possibility of

the valve being left in the handwheel mode following some maintenance, test etc. type operation.

17. Describe the tests and/or analysis performed to establish the qualification of the valve to perform its intended function under the environmental conditions exposed to during and after the DEA following its long term exposure to the normal plant environment.

18. What basis is used to establish the qualification of the valve, operators, solenoids, valves? How was the valve assembly (valve/operators) seismically qualified (test, analysis, etc.)?

19. Where testing was accomplished, describe the type tests performed conditions used etc. Tests (where applicable) such as flow tests, aging simulation (thermal, radiation, wear, vibration endurance, seismic) LOCA-DBA environment (radiation, steam, chemcials) should be pointed out.

20. Where analysis was used, provide the rationals used to reach the decision that analysis could be used in lieu of testing. Discuss conditions, assumptions, other test data, handbook data, and classical problems as they may apply.

21. Have the preventive maintenance instructions (part replacement, lubrication, periodic cycling, etc.) established by the manufacturer been reviewed, and are they being followed? Consideration should especially be given to elastomeric components in valve body, operators, solenoids, etc. where this hardware is installed inside containers.

DESCRIPTION OF OPERATIONAL TESTS

OF A 12 INCH CLOW TRICENTRIC VALVE

FOR

NUCLEAR PURGE SYSTEM SERVICE

BY

J. E. KRUEGER
NUCLEAR VALVE DESIGN ENGINEER

NOVEMBER 30, 1981

APPENDIX B

SUMMARY OF 12" CLOW TRICENTRIC

CHOKED FLOW/STATIC SEISMIC

OPERABILITY TEST

(Refer to Vought Corp. Report No. 2-59700/1R-52972)

INTRODUCTION -

A test was performed at Vought Corp., Dallas, Texas, on November 16, 1981, to demonstrate operability of a 12 inch Tricentric valve for flow and load conditions possible in case of a LOCA (Loss of Coolant Accident) in a nuclear plant. The test was run with a valve to be used in Jersey Central Power and Light's Oyster Creek Plant. The test was performed by Vought personnel under the direction of a Clow Engineer. Witnesses to the tests included representatives of GPU Nuclear of New Jersey and Bechtel of San Francisco.

OBJECTIVE -

The test was performed to demonstrate that the valve would operate under pressure, flow, and loadings simulating operating and seismic conditions possible during a LOCA. It was also desired that the open to close cycle be demonstrated to occur in less than 5 seconds. A secondary objective was to show aerodynamic torques produced by air flow over the disc were equal or less than those predicted and used in designing the valve and selecting the actuator. (Predicted torques used in design derived from previous air flow test performed with 3 inch scale models.)

TEST SET-UP -

The valve was installed in a straight pipe run with a stagnation chamber upstream approximately 6 feet. Downstream 3 feet was a diverging nozzle to prevent downstream pressure

from exceeding one atmosphere. Upstream of the stagnation chamber there were several servo-controlled valves used to maintain a constant pressure in the chamber. Air to this system was supplied from Vought's 28,000 cubic feet air storage tanks. The tanks were pressurized to 600 psig with the servo-valves used to maintain a pressure of 65 psig at the stagnation chamber upstream of the valve. Hydraulic load cylinders were provided to produce an 11.0 g load in two perpendicular directions through the valve actuator center of gravity.

INSTRUMENTATION -

Numerous measurements were made during the test with those relating directly to valve operation being printed on an oscillographic chart. These measurements were used to verify test parameters were met during the test and to monitor valve performance. All data was fed through a digitizer and recorded directly on magnetic tape for later study. Measurements were made at a rate of 10 per second. The measurements taken during the demonstration runs were as follows:

1. Total pressure in the stagnation chamber.
2. Total temperature in the stagnation chamber.
3. Total and static pressure upstream of the Clow valve.
4. Total and static pressure downstream of the Clow valve.
5. Static pressure in the pneumatic actuator cylinder.
6. Hydraulic pressure to the static load cylinders.
7. Angle of the disc in the Clow valve.
8. Torque on the valve drive shaft.

VALVE AND ACTUATOR DESIGN PARAMETERS -

The valve tested was designed for a differential operating pressure of 65 psi and combined operating and seismic loads of 11.0 g's. The seal was of laminated 316 SST and asbestos. The body design was 150 lb. class per ANSI B16.34. The shaft used for transmitting torque to close and seal the valve was of a 17-4 PH age hardenable stainless steel, heat treated to condition H-1100. The actuator used was a Bettis NT-316B-SR2 pneumatic spring return actuator. The actuator was of a fail closed design with the spring supplying the closing and seating torque (Note: Tricentric valves are designed for torque seating). The actuator was qualified for nuclear service.

CONDUCT OF TEST -

The test consisted of applying the static loads to the actuator and establishing a 65 psig upstream pressure with the Clow valve closed. A signal was then initiated to open the valve. The valve then cycled full open against flow and remained open until a signal to close the valve was provided. The valve then cycled to the closed position and seated. During this period data was taken automatically at 10 measurements per second at all sensors. This test was repeated 4 additional times at 65 psig and once at 35 psig. Note: These upstream pressures produced choked (flow at sonic velocity) flow through the valve during the valve open period.

RESULTS OF TESTS -

The tests demonstrated the following:

1. The Clow disc and shaft geometry provides for a positive aerodynamic closing torque for all angles from full open to full closed.
2. The aerodynamic torque values used for design of the Clow valve are conservative relative to measured torques. (Design torques were based on previous 3" scale model tests.)
3. The construction of the valve is rigid in its design such that no binding resulted under an 11.0 g load applied in two directions simultaneously.
4. The valve will cycle from full open to full closed in less than 5 seconds with any amount of flow from none to the maximum tested (108 lb/sec of air). Any value of flow above zero tended to close the valve faster (the valve closed in 3.6 sec. for a no flow condition).
5. Operator sizing was sufficient to cycle the valve from full closed to full open in less than 5 seconds for any tested flow rate.

CONCLUSION -

Clow has demonstrated that their nuclear purge valve design can meet and exceed typical specifications for this type of service. It was further shown that the valve will function as

required regardless of the LOCA pressure ramp curve (assumes lower pressures upstream at start of valve closure) often used by other valve manufacturers to show operability. In conjunction with other tests (now in progress) to show operability under many installed piping configurations, Clow valves can allow full open purge function during shutdown for refueling as opposed to the partially open position now allowed by the NRC. Further, it has been shown that the Tricentric can meet tight leak rate requirements with a metal to metal sealing which is more reliable and less costly in maintenance than sealing with elastomers.

Attachment No. 2

CLOW CORPORATION ADDENDUM II

TO

PATEL TECHNICAL REPORT PEI-TR-83-29

COVERING

EXTENSION OF SEISMIC QUALIFICATION

TO

CLOW WAFER STOP VALVE ASSEMBLIES
8" HBB-BF-AO-5035A&B
8" HBB-BF-AO-5036A&B
8" HBB-BF-AO-5042A&B
8" HBBOBF-AO-5044A&B and to
N732-SR80 Bettis Actuators
used on the subject valves

by

STEVEN NONDAHL
Design Engineer-Nuclear

Prepared for Bechtel Power Corporation

for

Boston Edison Pilgrim Station #600 Unit 1
In Accordance with Bechtel Specification 10394-P-119-1(Q) Rev. 0

Work Performed Under
Bechtel Purchase Order No. 10394-P-119-1-AC, Rev. 1

10394-M117-1-27-1

By

CLOW CORPORATION
ENGINEERED PRODUCTS DIVISION
WESTMONT, ILLINOIS

November 11, 1983

11-23-83

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REVIEWED	A	C	E	J	M	N	P	Q
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BY H. F. Phipps DATE 12-15-83

REPORT NO.: Clow Corp. Addendum II to Patel Technical Report PEI-TR-83-29

PREPARED BY: Steven M. Nondahl
Steven Nondahl
Design Engineer-Nuclear

APPROVED BY: James Krueger 11/11/83
James Krueger
Manager-Nuclear Engineering

Q.A. REVIEW: V. Pauperas
Victor Pauperas
Quality Assurance Manager

ORIGINAL ISSUE: November 11, 1983

REV. NO.	REV. DATE	REV. BY	CHECKED BY	Q.A. BY	DESCRIPTION OF CHANGES AND PAGES REVISED

Pages i - xv
and 1 thru 5
Appendix A page 1A & 2A

The subject valve assemblies, manufactured by Clow Corporation, are shown by this addendum and the referenced reports to meet the requirements of Bechtel Power Corp. Design Specification 10394-P-119-1(Q), Rev. 0 "Design Specification For Flanged Butterfly Valves For Nuclear Service for the Pilgrim Station No. 600, Unit No. 1, Boston Edison Co."

My review of the above mentioned design specification and this addendum allows me to certify to the best of my knowledge that the requirements of the specification have been met.

Theodore E. Thygesen

Theodore E. Thygesen
Registered Professional Engineer
State of Illinois
Registration No. 62-34780

ABSTRACT

This addendum describes the basis for extension of previous qualifications of Clow Stop Valves and Bettis Actuators tested for Job 82-2053(N) (Philadelphia Electric) to Job 82-2739(N) (Boston Edison). It includes identification of candidate assemblies, actuators, and components, consideration of the parameters necessary for qualification, and a comparison of the supplied equipment to these parameters. The result is that all units for the subject job are qualified in accord with the presented basis.

TABLE OF CONTENTS

1.0 SCOPE

This addendum presents information detailing similarities between the equipment supplied under Bechtel P.O. No. 10394-P-119-1-AC, Rev. 1 (Clow Job No. 82-2739(N)) and equipment tested under Bechtel P.O. 8031-P-144, Rev. 1 (Clow Job No. 82-2053(N)). It further provides the basis for extending the previous qualification tests to the subject valves.

2.0 EQUIPMENT TO BE SHOWN QUALIFIED

The following Valve Assemblies and Actuators constitute the candidates for qualification:

Bechtel Mark No.	Clow Serial No.	Bettis Act. Model No.
8" HBB-BF-AO-5035A	82-2739-01(N)-01	N-732-SR80
8" HBB-BF-AO-5035B	82-2739-01(N)-02	N-732-SR80
8" HBB-BF-AO-5036A	82-2739-01(N)-03	N-732-SR80
8" HBB-BF-AO-5036B	82-2739-01(N)-04	N-732-SR80
8" HBB-BF-AO-5042A	82-2739-01(N)-05	N-732-SR80
8" HBB-BF-AO-5042B	82-2739-01(N)-06	N-732-SR80
8" HBB-BF-AO-5044B	82-2739-01(N)-07	N-732-SR80
8" HBB-BF-AO-5044A	82-2739-01(N)-08	N-732-SR80-M3

3.0 QUALIFICATION BASIS

3.1 Valve Assemblies

Valve assemblies may be qualified by a combination of analysis and testing. Analysis is required to show that the stress levels and frequency response characteristics are within required code and spec allowed ranges. Testing is required to show operability in accord with Spec 8031-P-144, Rev. 2 Appendix 17, Paragraph 2B thru 2D.

3.1.1 Analysis Performed

For valve assemblies the following is provided to meet these requirements:

Analysis of the subject valves is covered and stresses and frequency response is shown suitable by

a) Generic report, PEI-TR-83-24, Rev. A, "Seismic Qualification Analysis of Clow 8 Inch Wafer Stop Valve" and

b) Site Specific Report PEI-TR-833700-1.

These have been supplied by separate submittal.

Note: The generic report meets requiements of ASME Section III, 1980 Edition through Winter 1982. The site specific report meets ASME Section III, 1980 Edition through Winter 1981. The required design code is ASME Section III, 1980 Edition with no addenda. Although the codes specified in the generic and site specific report differ from the required design code a review of the required addenda show no impact on the analysis method or conclusions.

In accord with Table C-1 of Appendix 17 of 8031-P-144, Rev. 2, the subject valve size 8" may be qualified by operability testing of a similar valve design in the size range of 6" to 12". Successful testing performed under Bechtel P.O. 8031-P-144-AC of a 6" valve assembly Serial No. 82-2053-02(N)-01 (Tag No. 6-HBB-BF-AO-57-121) and separate tests of a Bettis NT312-SR5 actuator qualifies the subject valve assemblies. Further, although not submitted for Bechtel approval, (tests were performed for another Clow Job) a test performed at Vought, Dallas, Texas, which was witnessed by John Strohm of Bechtel further backs up this qualification for an 11.0 g static load and full choked flow condition. Results of this test are covered in Vought Report 2-59700/1R-52972 which is submitted separately for reference.

3.2 Actuators and Accessories

3.2.1 Qualification Basis

Although Bechtel and qualification codes give no specific basis for generic qualification of actuators, the following requirements are considered reasonable and prudent

1. The candidate actuator shall be of similar construction to the qualified actuator.

2. The candidate shall supply an output torque less than or equal to a qualified actuator.

3. Major components of the candidate actuator (spring can, air cylinder) shall have a lower mass, lesser overhang, or more rigid construction than the qualified actuator.

4. Accessory mounting allows components to supply their safety-related function even though mounting details may be different than that previously qualified.

3.2.2 Actuator Candidate Qualification

The candidate actuators are Bettis Model N732-SR80 and N732-SR80-M3. These units incorporate a dual air cylinder and a return spring which provides the safety related function.

The subject units are similar in design to those tested previously, being of a pneumatic spring return design. One difference between the supplied units and tested units exists. The present updated Bettis design incorporates a bolt on spring can support in lieu of the weld on type supplied on the larger units previously tested. The bolting is prevented from loosening by using suitable lock washers. Also construction and function are suitable to prevent spring can motion when tied in to the pipe run as required. On this basis design similarity is maintained. (See Picture #1 for actual configuration of support).

The torque outputs, weights, and overhangs of the candidate and qualified actuators are presented below

	Actuator Model	Spring Torque (approx.) Valve Open (in. lb)	Valve Closed (in. lb)	Overhang (in)* Spring Can	Air Cylinder	Weight (lb)
candidate	N732-SR80	19,000	9,000	36.88	23.38	397
qualified	NT-312-SR5	13,400	5,800	35.38	17.25	533
qualified	NT-820-SR4-S	144,500	93,000	73.25	27.95	1491

*Length from mounting surface on rigid center housing

From the above and actual actuator dimensions it can be seen that tests of the NT-312-SR5 and NT-820-SR4-S will qualify the N732-SR80 units. Although not presented here, dimensions of the N732-SR80 (spring and air cylinder diameter) would suggest greater stiffness than the qualified units. Thus primary size and design criteria are met and the units are considered qualified. Note that Bettis Report 37274 includes a test of a N732-SR80-M3. Thus the actuator provided for Mark No. 8" HBB-BF-AO-5044A is also qualified

3.2.3 Actuator Accessory Qualification

Accessories which require separate qualification are the Asco solenoid valve and Namco switches. Both are qualified separately by the manufacturer. Model numbers and qualification report numbers are shown below:

Item	Model No.	Qual. Report No & Date	
Asco Solenoid Valve	NPL 831664E	AQR-67368 Rev. 0	Nov. 2, 198
Namco Limit Switch	EA180-31302	QTR-105	Aug. 28, 19
Namco Limit Switch	EA180-32302	QTR-105	Aug. 28, 19

The mounting configuration of various components differs to some extent from that previously supplied. The changes that have been made were necessitated by actuator size and requirements imposed by Clow on Bettis to provide a design for accessory mounting suitable for the required seismic environment. The junction box provided is mounted in the same fashion as on units previously tested. The filter and regulator are also mounted in a similar fashion as the tested units. The solenoid valve mounting has been changed from mounting on a hex nipple at the end of the air cylinder to mounting on a bracket connected to the rigid actuator center housing. Also the Namco switches were fastened on their side rather than their back to the mounting bracket. This was done to allow a short straight run of flex conduit to the junction box. A corresponding change was made to the switch trip method from a cam to a trip plate. The trip plate weight is approximately 6 oz. compared to approximately 80 oz. for the previous cam design. The trip plate is securely fastened by two #10 SHCS which provides for a rigid connection. The solenoid valve mounting will provide for a lower acceleration at this point as compared to mounting at the end of the air cylinder. The air cylinder mounting provides for a greater cantilevered length and higher accelerations. This can be seen in the accelerometer response curves in Bettis Qualification Report 37274. Since the safety related function of the unit is to close, any failure to tubing connections between the solenoid valve and air cylinder will result in performance of this function. On this basis accessory mounting is considered qualified in accord with test described in Patel Report PEI-TR-83-29

4.0 REFERENCES

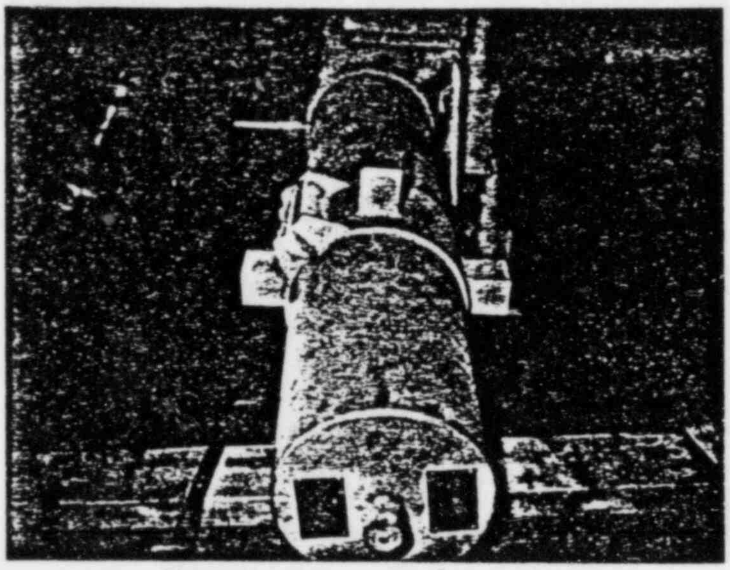
Report No./Date	Title
Patel-PEI-TR-83-29, Rev. A Aug. 10, 1983	Seismic Qualification of Clow Wafer Stop Valve Assemblies Job Number 82-2053(N)
Clow-Addendum I to PEI-TR-83-29 Aug. 16, 1983	Static Load Test and Seismic Qualification of Clow Wafer Stop Valve Assemblies
Patel-PEI-TR-83-24,Rev.A June 24, 1982	Seismic Qualification Analysis of Clow 8 Inch Wafer Stop Valve

REFERENCES (con't)

Report No./Date	Title
Patel-PEI-TR-833700-1/ Oct. 4, 1983	Seismic Qualification Analysis of Clow 8 Inch Wafer Stop Valve Job Number 82-2739-(N)-all
Vought-2-59700/1R-52972 Dec. 15, 1981	Simultaneous Static Load And Flow Interruption Capability Tests of a 12 Inch Valve For The Clow Corporation
Bettis-37274 Aug. 12, 1980	Nuclear Qualification Test Report
Namco-QTR-105 Aug. 28, 1980	Qualification of EA180 Series Limit Switches
Asco-AQR67368, Rev.0 March 2, 1982	Report on Qualification of Automatic Switch Co. (ASCO) Catalog NP-1 Solenoid Valves for Safety-Related Applications In Nuclear Power Generating Stations.

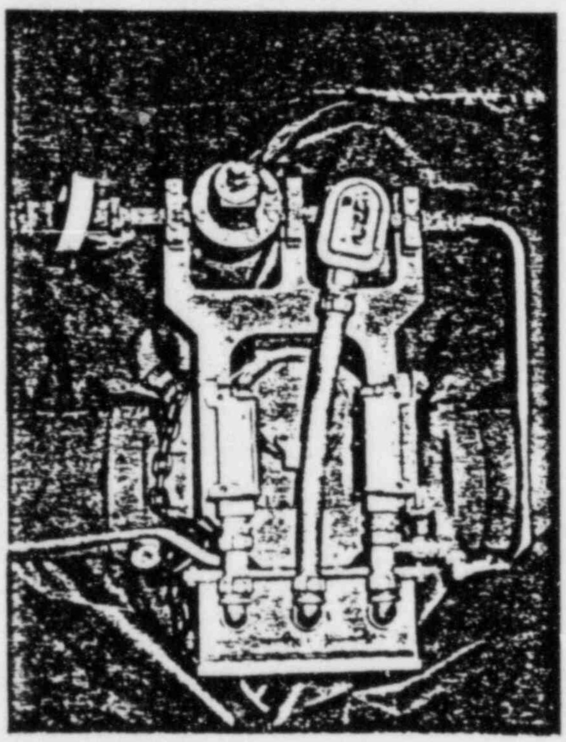
Picture No.	Description
1	Picture of spring can end with bolt on support bracket N-732-SR80 actuator
2	Picture of accessory mounting on N732-SR80 actuator showing filter, regulator, solenoid valve, junction box, and switch mounting.
3	Picture of actuator with accessories mounted on valve

Figure No.	Description
1	Excerpt from Bettis Drawing SPC-9153A showing accessory mounting on NT312-SR5 actuator used on 82-2053(N) Job
2	Excerpt from Bettis Drawing SPC-9382B showing accessory mounting on N732-SR80 actuator used on 82-2739(N) Job.



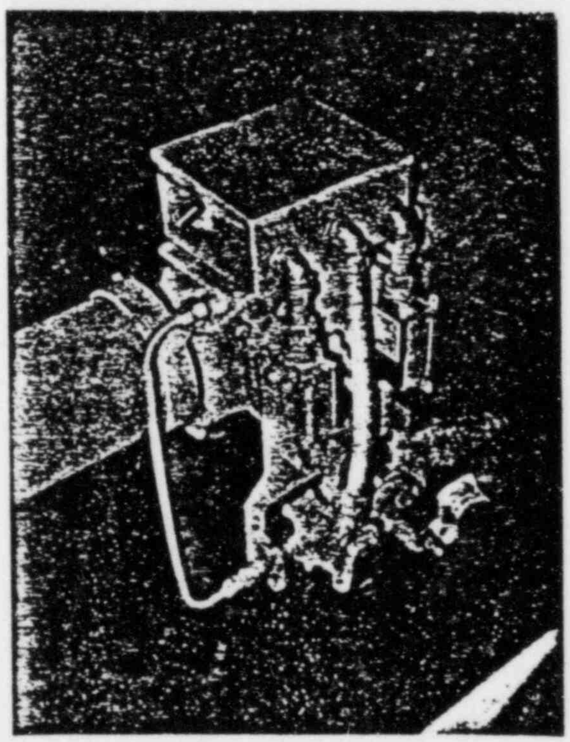
PICTURE #1

Picture #1



PICTURE # 2

Picture #2



PICTURE NO 3

Picture #3

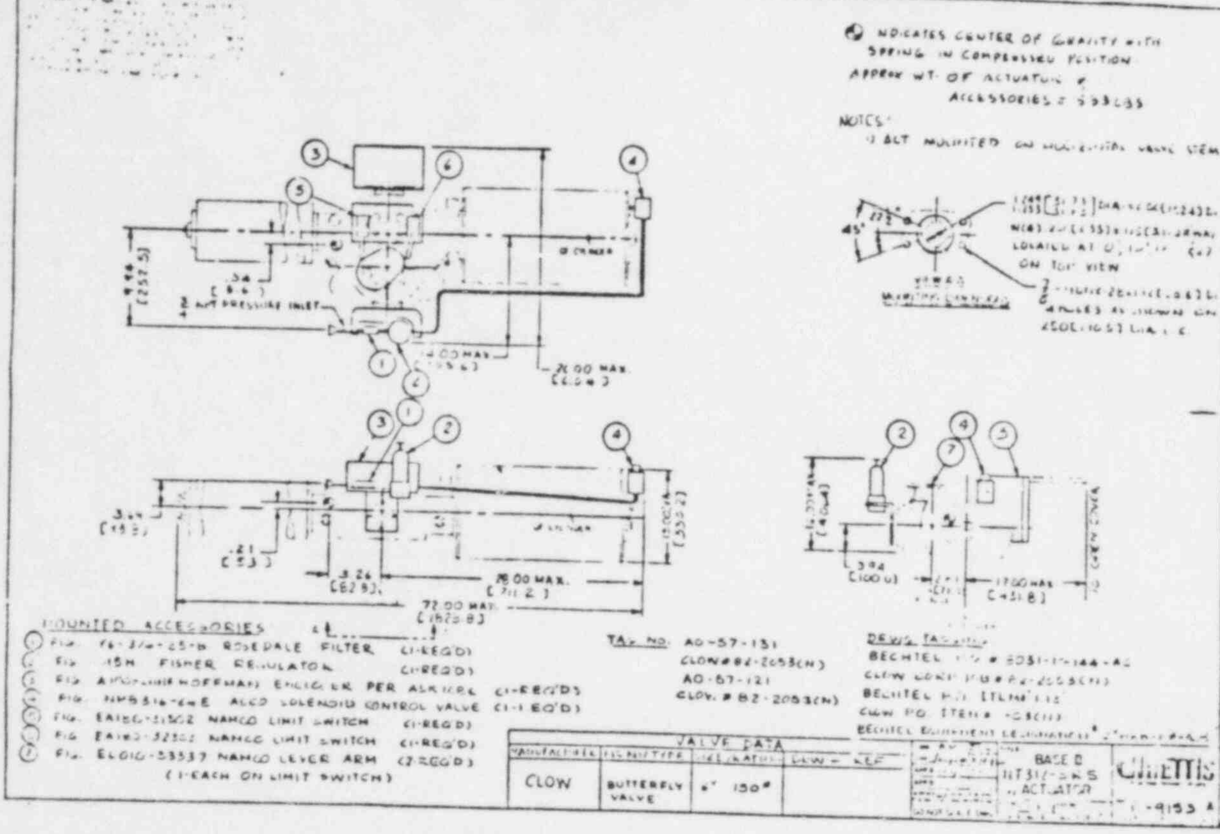


Figure 1

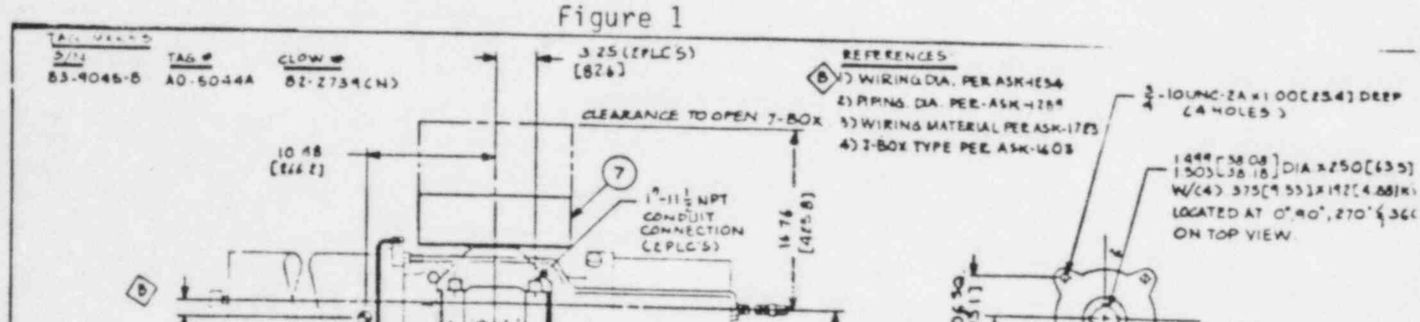


Figure 2

DESIGN REPORT
DR-82-2739(N)
Rev. A

REPORT COVERS DESIGN OF VALVE TAG NOS.

8" HBB-BF-AO-5035A
8" HBB-BF-AO-5035B
8" HBB-BF-AO-5036A
8" HBB-BF-AO-5036B
8" HBB-BF-AO-5042A
8" HBB-BF-AO-5042B
8" HBB-BF-AO-5044B
8" HBB-BF-AO-5044A

by

STEVEN M. NONDAHL

DATE: October 7, 1983

PREPARED FOR

BOSTON EDISON COMPANY
PILGRIM STATION #600, UNIT 1

P.O. No. 10394-M-119-1-AC, Rev. 1

REPORT NUMBER: DR-82-2739(N), Rev. 0

PREPARED BY: Steven M. Nordoll 10/7/83

APPROVED BY: J E Krueger 10/7/83

Q.A. REVIEW: D. Pampanas 10/31/83

ORIGINAL ISSUE: (REV. 0) OCTOBER 7, 1983

REV. NO.	REV. DATE	REV. BY	CHECKED BY	Q.A. BY	DESCRIPTION OF CHANGES AND PAGES REVISED
A	1/3/84	S. Nordoll	J E Krueger 1/4/84	D. Pampanas 1/4/84	P.2-change III to II P.8-delete last sentence of V-A-6

I. DESIGN CODE AND SPECIFICATIONS

The following codes, standards, and customer specifications were utilized for the design, analysis, and construction of the subject valves as identified in Section II of this document.

A. Codes and Standards

ASME Section III - 1980 Edition

ANSI B16.34 - 1980 - Steel Valves
IEEE-323-1974 - Standard for Qualifying Class 1E Equipment
IEEE-344-1975 - Recommended Practices for Seismic Qualification of Class 1E Equipment
IEEE-382-1980 - Standard for Qualification of Safety Related Valve Actuators
ANSI B16.5 - 1981 - Steel Pipe Flanges and Flanged Fittings

B. Customer Specifications

10394-P-119-1(Q), Rev. 0 Design Specification, 9/2/83
8031-P-353, Rev. 4, Adden. 1, Butterfly Valves, (6/28/73)
8031-G-5, Rev. 5, Document Requirements, (2/8/80)
8031-G-13, Rev. 8, Supplier Q.A., (6/21/77)
8031-G-18, Rev. 5, Environmental Requirements (6/16/82)
8031-P-358, Rev. 0, Dynamic Qualification Act. (12/1/81)
8031-P-359, Rev. 1, Air Operators (5/13/82)
P.O. Item 1.47 "General Requirements" replaces 8031-G-1
The above as modified by the clarifications and exemptions stated in Clow to Bechtel letter (Jacobi to Solyan) dated 6/2/83.

(A) II. ITEMS COVERED BY DESIGN REPORT

This report covers the design of Clow Tricentric Wafer Style Butterfly Valves and accessories produced under the following numbers:

CLOW SERIAL NO. (S)	BOSTON EDISON SERIAL NO.
82-2739-01(N)-01	8" HBB-BF-AO-5035A
82-2739-01(N)-02	8" HBB-BF-AO-5035B
82-2739-01(N)-03	8" HBB-BF-AO-5036A
82-2739-01(N)-04	8" HBB-BF-AO-5036B
82-2739-01(N)-05	8" HBB-BF-AO-5042A
82-2739-01(N)-06	8" HBB-BF-AO-5042B
82-2739-01(N)-07	8" HBB-BF-AO-5044B
82-2739-01(N)-08	8" HBB-BF-AO-5044A

III. DESIGN DOCUMENTS WHICH CONSTITUTE THE VALVE AND ACCESSORY DESIGNS ARE AS FOLLOWS:

(Note: Numerous purchased items do not have drawings, but are adequately described in the manufacturing bill of materials)

A. Manufacturing Bill of Material for 82-2739-01(N), Rev. C
8" Wafer Valve, R.H.
Prints per above Bill of Material are as follows:

B/M Item No.	Print No.	Item Description
0	D-0741A	Assembly - Valve & Actuator
1	B-4245	Valve Body Machining
1.1	B-4170	Valve Body Rough
1.3	A-5431	Valve Seat
1.10	A-3450C	Mounting Plate
2	B-4206	Disc & Seal Assembly
2.1	B-4172	Disc & Ear Assembly
2.2	C-0299	Disc Ear (single)
2.4	A-5409	Disc Plate
3	B-4171	Clamp Ring
4	A-5464A	Seal, Laminated
8	B-4173	Cover Plate
11	A-5410	Spacer
12	A-5411	Annular Key
13	A-5497	Bearing
16	B-4247	Thru Shaft
18	A-5413	Parallel Key (ear)
20.1	B-4273	Gland Tube
20.2	B-4176	Gland Flange
25	A-5477	Adaptor Plate
27	A-5496	Lantern Ring
33	A-5415	Parallel Key (Actuator)

B/M Item No.	Description	Size
5	Hex Head Cap Screw (disc)	3/8-16 UNC x 1"
6	Socket Set Screw (disc)	3/8-16 UNC x 1/2"
10	Hex Hd. Cap Screw (cov. plate)	1/2-13 UNC x 1 1/2"
21	Stud (gland assembly)	1/2-13 UNC x 3"
22.1	Hex Nut (gland assembly)	1/2-13 UNC
22.2	Jam Nut (gland assembly)	1/2-13 UNC
24	Socket Head Cap Screw (act. mounting)	3/4-10 UNC x 2 3/4"
28	Socket Head Cap Screw (act. mounting)	3/4-10 UNC x 3"

B. Bettis Air Operators:

Item Description:

1. Bettis N732-SR80-5
2. Bettis N732-SR80-M3-S (for use on 8"-HBB-BF-AO-5044A only)

IV. APPLICABLE ITEMS COVERED IN ANALYSIS:

The code states in NCA-1130(a) that the rules..."are applicable only to those components that are designed to provide a pressure retaining or containing barrier." NCA-1130(b) further states "The rules are not intended to be applicable to valve operators, controllers, position indicators, pump impellers, pump drivers, or other accessories and devices, unless they are pressure retaining parts or act as core support structures or component supports." Thus this report and supplemental reports cover only those items relating to pressure retention or as specifically requested by purchase order or spec.

A. For pressure retention the pressure containing parts are:
(1) Valve Body
(2) Valve Disc
(3) Cover Plate
(4) Cover Plate Bolts
(5) Gland Flange
(6) Gland Studs and First Nut
(7) Pipe Plug (for packing leakoff)
(8) Pipe Plug (for "O" Ring leakoff)

B. Non-pressure retaining accessories requiring analysis for operability are as follows:
(1) Valve Disc Clamp Ring and Bolts
(2) Valve Disc Parallel Key
(3) Valve Adaptor Plate Bolts attaching the Adaptor Plate to the Valve Body
(4) Valve Actuator Attachment Bolts
(5) Valve Shaft to Actuator Parallel Key
(6) Valve Shaft to Ear Dowel Pin
(7) Valve Drive Shaft

V. CALCULATION OF STRESSES FOR DESIGN CONDITIONS

A. For pressure retaining parts, wall thickness is demonstrated as follows:

(1) a. For valve body, the required wall thickness is derived from ANSI B16.34 (Per NC 3512.3). The wall thickness is taken from Table 3 of the ANSI B16.34 - 1981 for a valve body I.D. of 7.981.

required wall = B_w = .31" for Class 150
actual wall = 1.44"

b. For the thickness between the shaft penetration bore and flange bolt holes NC 3512.3(d) applies and required wall = S_w = .25 x body wall thickness.

$S_w = .25\ B_w = (.25)(.31) = .0775"$
actual thickness is .24"

(2) The code does not cover thickness requirements for valve discs of the Tricentric design. The minimum design thickness is established from the following equation:

$$td = .707\ D\left(\frac{Pw}{Sa}\right)^{\frac{1}{2}}$$

D = valve I.D. (larger than disc diameter to be conservative) = 7.981

Pw = design working pressure (ΔP) = 75 PSI

S_a = code allowed stress (from Section III Appendix, Table I-7.1 for SA516 Gr. 70 plate) = 17500 PSI

Thus $td = (.707)(7.981)\ (75/17{,}500)^{\frac{1}{2}}$
$= .370$ = min required thickness

The actual thickness specified on design drawing is .520". The equation used corresponds to NC 3325.2 (b) equation (5) where C = .5 (a conservative assumption for the type of disc boundary condition employed).

(3) Cover plate thickness is analysed by NC 3325.2 (b), equation (5)

$$t = d\left(\frac{CP}{S}\right)^{\frac{1}{2}}$$

where:

d = bolt circle diameter = 3.25
(per Figure NC 3325.1(i))

C = dimensionless factor = .17

P = hydro pressure of body = 450 PSI

S = allowable stress per Table I-7.1 = 17,500 PSI

thus $t = 3.25\ (.17 \times 450/17{,}500)^{1/2} = .215''$

A nominal cover plate thickness of 5/8" was selected for the detail.

(4) Stress levels for cover plate bolting are as follows, using methods of Section III, Appendix XI-3220:

Wm1 = min bolt load to keep gaskets seated during operation

$= .785G^2P + (2b\pi GmP)$

Wm2 = min required load for gaskets seating
$= b\pi Gy$

G = 2.50
P = 450 (hydro)
b = .25
m = 2.75
y = 3700

Thus:

Wm1 = 7067 LBF
Wm2 = 7264 LBF

Wm2 > Wm1, so use
Wm2 = 7264 LBF

The root area for 1/2-13 UNC is .126 in^2
Thus, .126 in^2 x 4 bolts = .504 in^2
Bolt stress is then 7264 LBF/.504 in^2 = 14,413 PSI
The allowable stress for SA193 Gr. B7 bolts is 25,000 PSI

(5) & (6) The gland flange and studs are not addressed by the code but the following simplified calculations are supplied:

Shaft O.D. = 1.5"
Packing O.D. = 2.275
Stud size = 1/2-13 UNC: (root area = .126")

L = load on studs and gland flange

$L = \pi/4\ (2.275^2 - 1.5^2)(450 \text{ hydro pressure})$

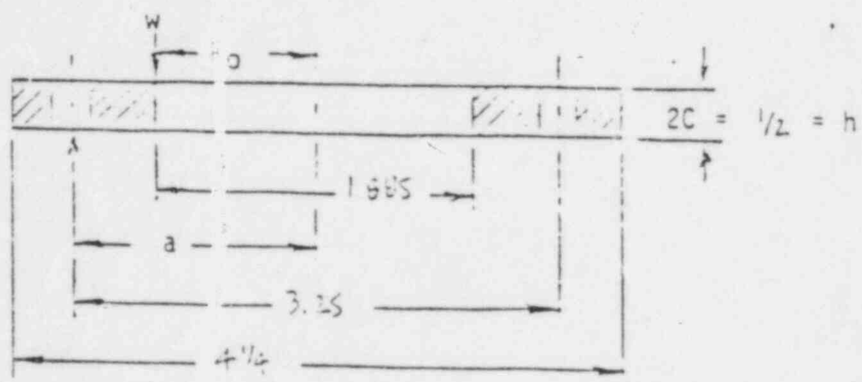
$= 1034$ lb.

$S = \text{stud stress} = \dfrac{\text{load } L}{(\#\text{ bolts})(\text{stress area})}$

$S = \dfrac{1034}{(4)(.126)} = 2052$ PSI

Allowed stress = 25,000 PSI

For flange loading the worst case is assumed with full load concentrated at flange bore center.



W = load/inch circum. = $1034/1.885\pi$ = 175 lb/in cir.

$R_o = 1.885/2 = .942$ $\quad$ $a = 3.25/2 = 1.625$

$R_{o/a} = .942/1.625 = .58$

From "Formulas For Stress And Strain", by Roark
(5th Edition, P. 334, Table 24, Case 1 and 1a)

$Km = .818$ (interpolating for $R_{o/a} = .58$)

$M = KmWa = (.818)(175)(1.625) = 233$ in=lb

$Max = MC/I$ where $I = \frac{bh^3}{12}$

$b = 4.25 - (2 \times .5625) - 1.885$

$= 1.24$

$$I = \frac{(1.24)(.5^3)}{12} = .013$$

$$Max = \frac{(233)(.25)}{.013} = 4481 \text{ PSI}$$

Using this simplified approach it can be seen the stress is well below code values.

(A)

(7) Pipe plug (for packing leakoff)
&
(8) Pipe plug (for "O" ring leakoff)

Table NC-3132-1 of Section III of the Code references ANSI B16.11, "Forged Steel Fittings, Socket Welding and Threaded". The 1980 Edition of this standard states in Table 1 that for 1/8 to 4 inch threaded plugs and bushings, "plugs and bushings are not identified by pressure class. They may be used for ratings up through pressure class 6000."

Therefore, these pipe plugs are more than suitable for the intended service.

B. Analysis of non-pressure retaining accessories requiring calculations to show operability is as follows:

(2) Valve disc parallel keys and (5) valve shaft to actuator parallel key:

The calculations of parallel key strengths to determine torque to yield in shear are as follows:

(2) Con't.
&
(5) Valve disc keys (two of them) have dimensions of 3/8" square x 2 1/4" long. Yield strength is 75,000 PSI. Shear yield = (.6) tensile yield
= (.6)(75,000)
= 45,000 PSI

Shaft radius is .75" = R
Key shear area is 3/8" x 2" = .75 in^2

Load to yield key is .75 in^2 x 45,000 PSI = 33,750 lb.

33,750 lb. x .75" R = 25,300 in-lb = 2,100 ft-lb for one key. For two keys, then 4,200 ft-lb would be required to yield both keys.

The key in the actuator is 3/8 square x 3" long. In a similar manner as above, 3,200 ft-lb is required to yield actuator key in shear.
Actuator output torque for valve seating is 756 ft-lb.

(6) Calculation to determine stress level in the valve shaft to ear dowel pin:

Dowel pin size: 3/8" dia. x 1" lg, comm alloy steel, 110,000 min tensile, 24-30 RC

Shear bearing area: Approximate by a circular area

For a circle, $\tau max = 4/3\ \tau$ average

F = displacing force on shaft = pressure x $\pi/4$ (shaft dia)^2

thus $F = \pi \dfrac{(1.5)^2}{4}$ x 285 PSI (CWP) x 1.5 (Hydro) = 755 lb

$\tau av = \dfrac{F}{D^2\ \pi/4}$ (D = dowel pin dia.)

$\tau max = \dfrac{16F}{3\pi(D)^2} = \dfrac{16(755)}{3\pi(3/8)^2}$

= 9,115 PSI

$\tau all = .6\gamma all$ = 66,000 PSI

thus 9,115 (τmax) < 66,000 PSI (τall) and loading on pin is acceptable.

(6) Con't.

For items (1), (3), (4), and (7) of Section IV B of this report, the analysis is covered in report PEI-TR-833700-1 by Patel Engineers. This report constitutes the "Design Report" required by the code.

REPLY TO BECHTEL COMMENTS ON DOCUMENT # DR-82-2739(N), REV. 0

COMMENT #1

I Design Code & Specifications
Refers to Section II - There is no Section II, unless the following Section III is Section II.

Reply: This is an editorial error and will be corrected.

COMMENT #2

III-A Should correctly reference the Valve Drawing D-0741C

Reply: Correct reference to the valve drawing is reconciled in Addendum I to DR-82-2739(N), Rev. 0.

COMMENT #3

V-A (1)a The thickness per ANSI B16.34 is exclusive of the corrosion allowance of 0.080. The drawing shows a body wall of .380". The design report states actual wall = 1.44".

The above must be reconciled.

Reply: Care must be taken in the interpretation of just what these numbers on the drawing are. Drawing D-0741C states that the minimum thickness of the body wall with corrosion allowance is .380". This is only the minimum. Due to the nature of the design of the valve, the actual body wall on the machining drawing comes out to be, conservatively, 1.44". However, minimum body wall with corrosion allowance should be .390" and not .380". This was an error. The arrangement drawing will be changed accordingly.

COMMENT #4

V-A (2) The disc thickness specified on the drawing is .450 inclusive of corrosion allowance. The calculation shows .520". The difference between the two must be reconciled.

Reply: The calculation does not show a thickness of .520". It shows a calculated thickness of .370". .370" + C.A. is .450" which is reflected on the arrangement drawing. The disc is machined at .520" for conservatism. The report is correct.

COMMENT #4

V-A (3) For this type of construction C = 0.20
The thickness required:

$$t = d\sqrt{\frac{CP}{S}} + C.A.$$

$$= 3.25\sqrt{\frac{(.2)(150)}{17500}} + .080$$

$$= 3.25\sqrt{\frac{30}{17500}} + .080$$

$$= .134 + .080$$
$$.215"$$

C = 0.20
d = B.C. Diameter

S = Allowable Stress = 17500 PSI

C.A. = Corrosion Allowance

P = Design Pressure = 150 PSI

The calculated thickness is correct, but the individual factors in the formula must coincide with those given in the Code. Also the drawing shows a cover plate thickness of .229". This must be corrected.

Reply: The calculations listed in the above Bechtel Comment are incorrect. Figure NC 3325.1(i) clearly shows the correct flange model, and clearly states that C = .17. Additionally, the design pressure (P) for the valve is not 150 PSI. The design is Class 150, which implies a cold working pressure of 285 PSI. The calculations in the report further assume hydrotest pressure (1.5 x 285 PSI) for conservatism. In other words, the calculations in the report are correct. However, the minimum cover plate thickness on the drawing will be changed to agree with what is in the report. A different equation was used to calculate the thickness when the drawing was made. Again, for conservatism, a 5/8" plate thickness was selected for the detail (making min thickness requirements almost academic.)

COMMENT #4

V-A (6) In computing the maximum stress

$$I = \frac{(1)^{*}\ (.5)^3}{12} = .0104$$

$$\text{Max } S = \frac{Mc}{I} = \frac{(233)(.25)}{.0104} = 5600 \text{ PSI}$$

*Note that the calculated moment is per inch of circumference.

Reply: In talking with Al Meyers of Bechtel Power, it was pointed out that this comment was meant for information only. No change to the report is required.

COMMENT:

Delete or amplify upon the final sentence regarding packing friction. It is argumentative and inconsequential. (This applies to Page 8, middle of page.)

Reply: Sentence will be deleted.

Steven M. Nondahl

Steven M. Nondahl
Design Engineer

cc: J. Krueger

BECHTEL POWER COPORATION	JOB 10394
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1 ☒ Work may proceed.
2 ☐ Submit final document. Work may proceed.
3 ☐ Revise and resubmit. Work may proceed subject to resolution of indicated comments.
4 ☐ Revise and resubmit. Work may not proceed.
5 ☐ Permission to proceed not required.

Permission to proceed does not constitute acceptance or approval of design details, calculations, analyses, test methods, or materials developed or selected by the supplier, and does not relieve supplier from full compliance with contractual obligations or release any "holds" placed on the contract.

REVIEWED	A	C	E	J	M	N	P	Q
		✓					LB	

BY [illegible signature] DATE 1-20-84

1-10-84

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JOB 10394
BECHTEL POWER CORPORATION
San Francisco

10394-M114-1-21-2

bcc: J. L. Carton/C. McBride w/o
A. Teller w/o
C. B. Hogg w/a
G. Zito w/a
J. Cravotto w/o
R. W. Fosse w/a
S. L. Case w/a
K. Jindal w/a
K. Jue/C. Shelton w/o
R. Raghavan w/a
H. Phipps w/a
C. McCullar/M. Osborne w/a

Attachment No. 4

Bechtel Power Corporation

Engineers—Constructors

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

November 24, 1982

IN OUT PI0556

10394-BLE - 1417

NOV 2

IN OUT

Mr. G. M. McHugh, Jr.
Deputy Manager - Nuclear Engineering Department
Boston Edison Company (NUCLEAR)
25 Braintree Hill Office Park
Braintree, Massachusetts 02184

Subject: Pilgrim Station 600 Unit No. 1
Job 10394, Boston Edison Company
Sub 105 - BECo P.O. 62812
Valve Betterment Program

Reference: Letter 10394-BLE-1388 dated
September 27, 1982

Dear Mr. McHugh:

As part of the valve betterment program, we performed an analysis to determine the size of replacement valves for the existing 20" valves used in the primary containment purge, vent, and vacuum breaker lines. Based on this analysis, we recommend that the existing valves be replaced with 8" size butterfly valves.

The analysis was performed to determine the revised operating points for the drywell and torus supply air fans (VSF-205 A & B), and standby gas treatment system exhaust fans (VEX-210 A & B) using 10" and 8" replac[illegible]alves. Available fan performance characteristics were u[illegible]his analysis. Based on the revised flow rates, the time r[illegible] purge/vent the primary containment was calculated. [illegible]s are shown on Enclosure 1. Four (4) times the free vol[illegible]well and torus (approximately 1 million cubic feet), a[illegible]updated FSAR, was assumed as the volume of air/gas to b[illegible]ed. Enclosure 2 shows the fan and system curves ind[illegible]xpected operating points for fans VSF-205A & B, and VEX-2[illegible] & B [illegible]ler the two operating modes stated in Enclosure 1.

As requested, we have initiated the paperwork required to release the calculation to BECo.

158/4

In accordance with Branch Technical Position CSB6-4, Revision 2-July 1981, evaluation of radiological consequences will not be required for 8" valves.

Our investigation indicates that 8 inch valves are not available from other projects which are under construction or have been cancelled. Therefore, we intend to solicit bids from the following vendors to procure ten (10)-8 inch replacement valves. Enclosure 3 is the data sheet of the valves we intend to ask for bids.

- Jamesbury
- Posi-Seal
- Allis-Chalmers
- Clow
- Pratt

We will forward our recommendation after the receipt of bids.

Very truly yours,

CH

C. B. Hogg
Project Engineer

R.G. CBH:KKJ:RR:jat

Enclosures: 1. Calculation Results (Table)
2. Fan Curves and System Curves
3. Valve Data Sheet

Partial Complete(s)
REPLY TO CHRON
No.
REQUIRES REPLY
☒ No ☐ Yes
Date Due
Requesting Group

PI0556

CALCULATION RESULTS

MODE 1

SIMULTANEOUS PURGING & VENTING OF DRYWELL AND TORUS

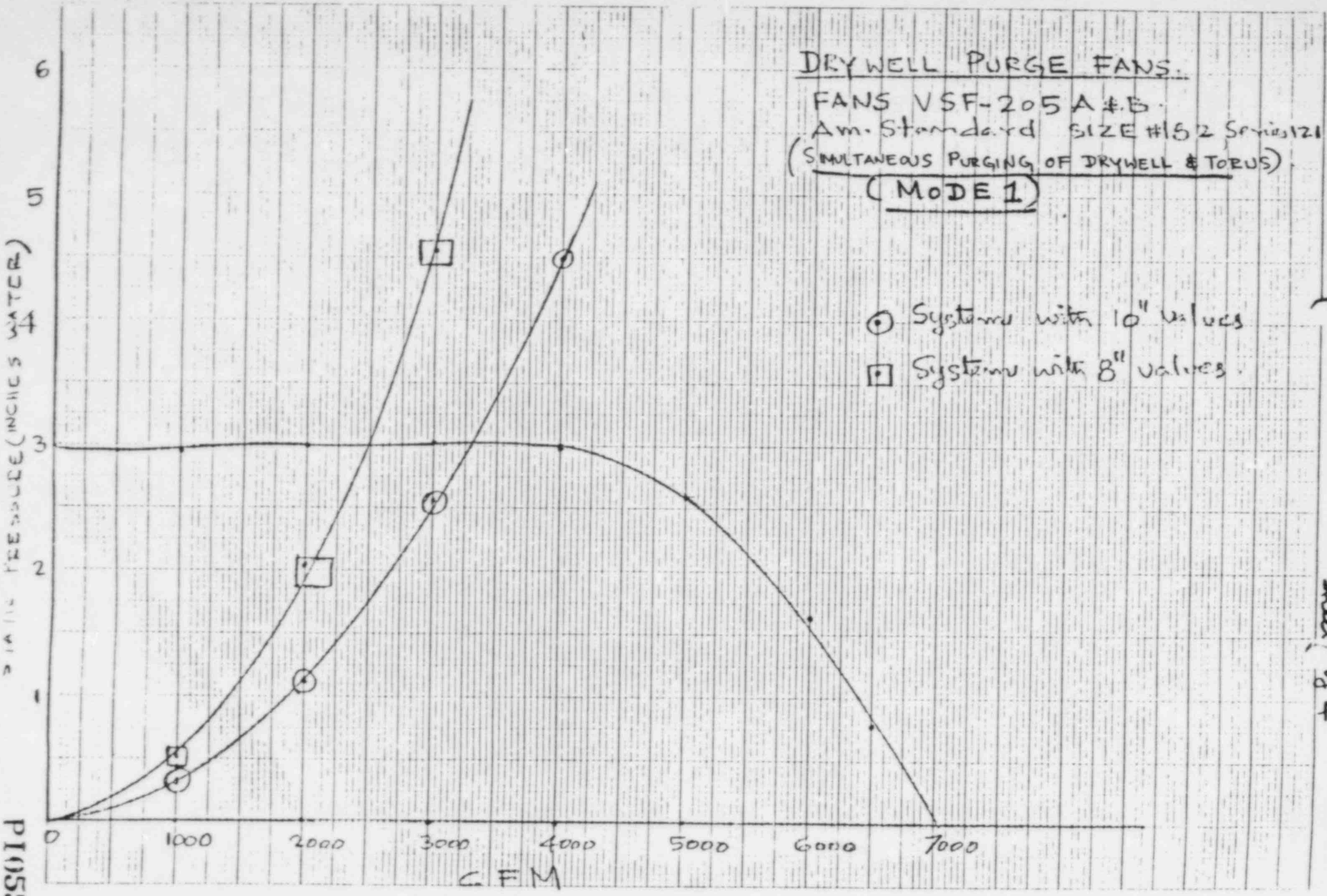
10" Isolation Valves		8" Isolation Valves	
Expected Flow Rate	Time req'd to purge/vent	Expected Flow Rate	Time Req'd to purge/vent
3300 cfm	5-1/2 hours	2500 cfm	7-1/2 hours

MODE 2

INDIVIDUAL PURGING & VENTING OF DRYWELL AND TORUS

10" Isolation Valves		8" Isolation Valves	
Expected Flow Rate	Time Req'd to purge/vent	Expected Flow Rate	Time Req'd to purge/vent
2400 cfm	4 hours ea.	1450 cfm	6-3/4 hours ea.

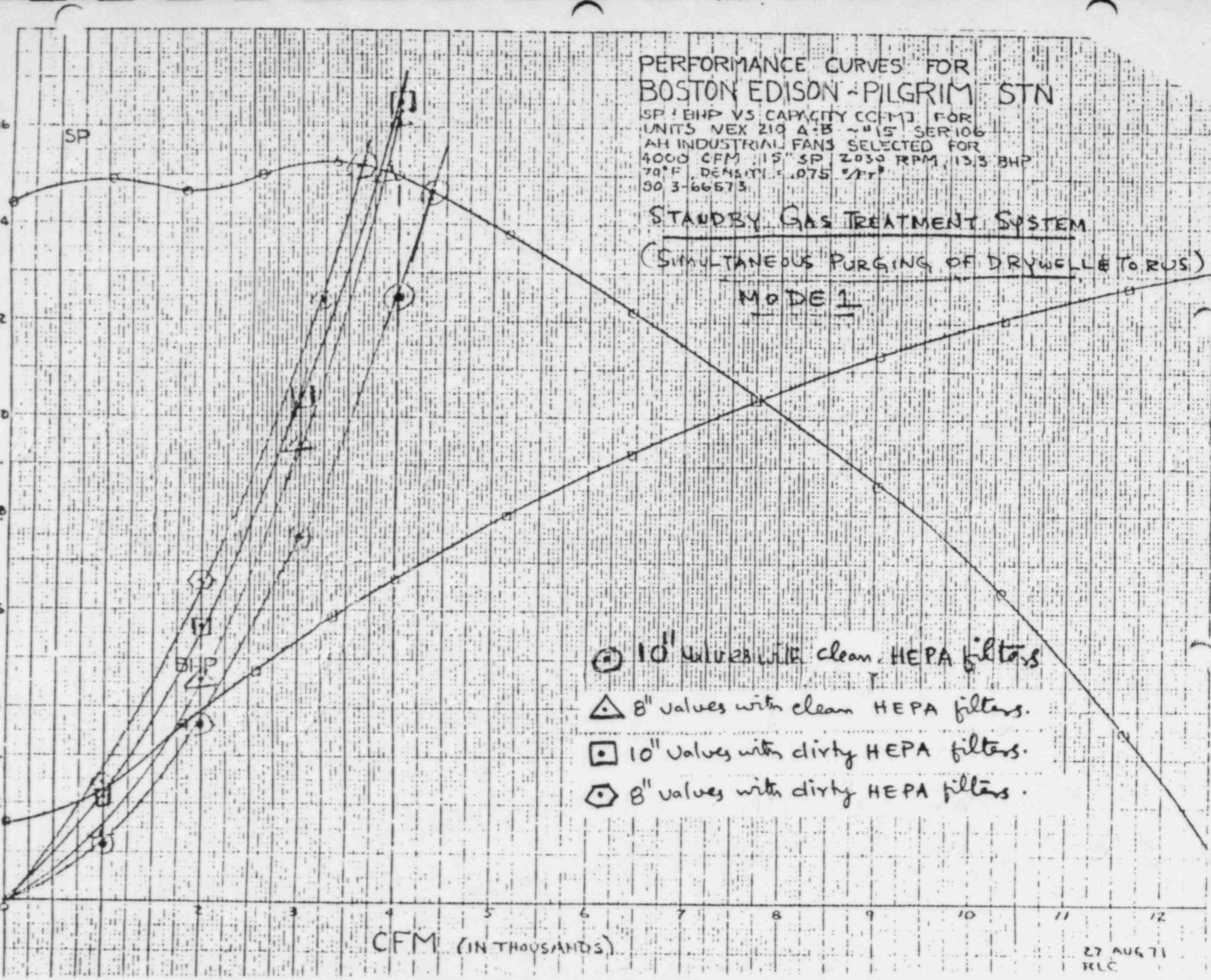
Enclosure 2 CO BLL - 1417
Sheet (of 4



PI0556

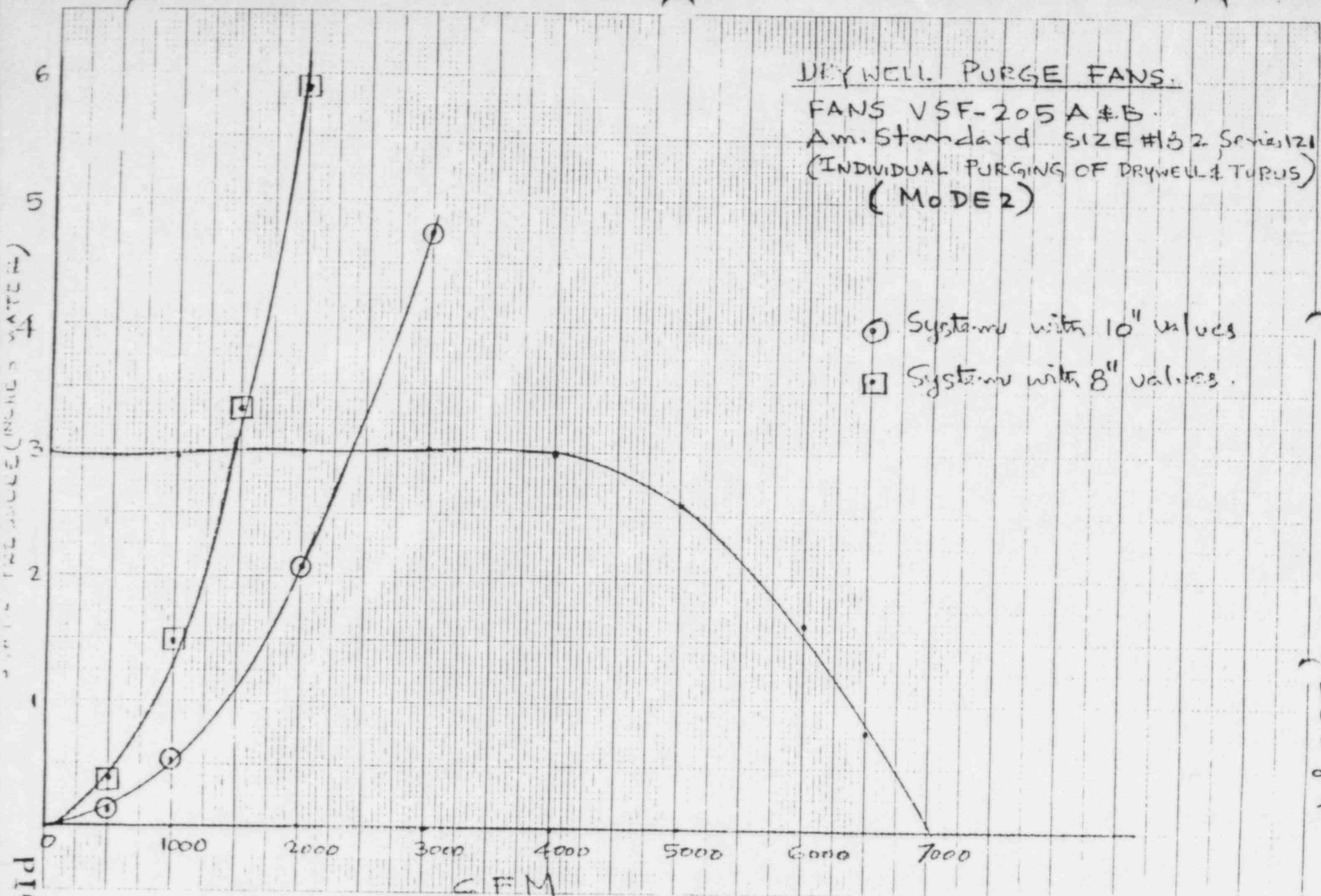
46 1322

Enclosure 2 to BLE-4/7
Sheet 2 of 4



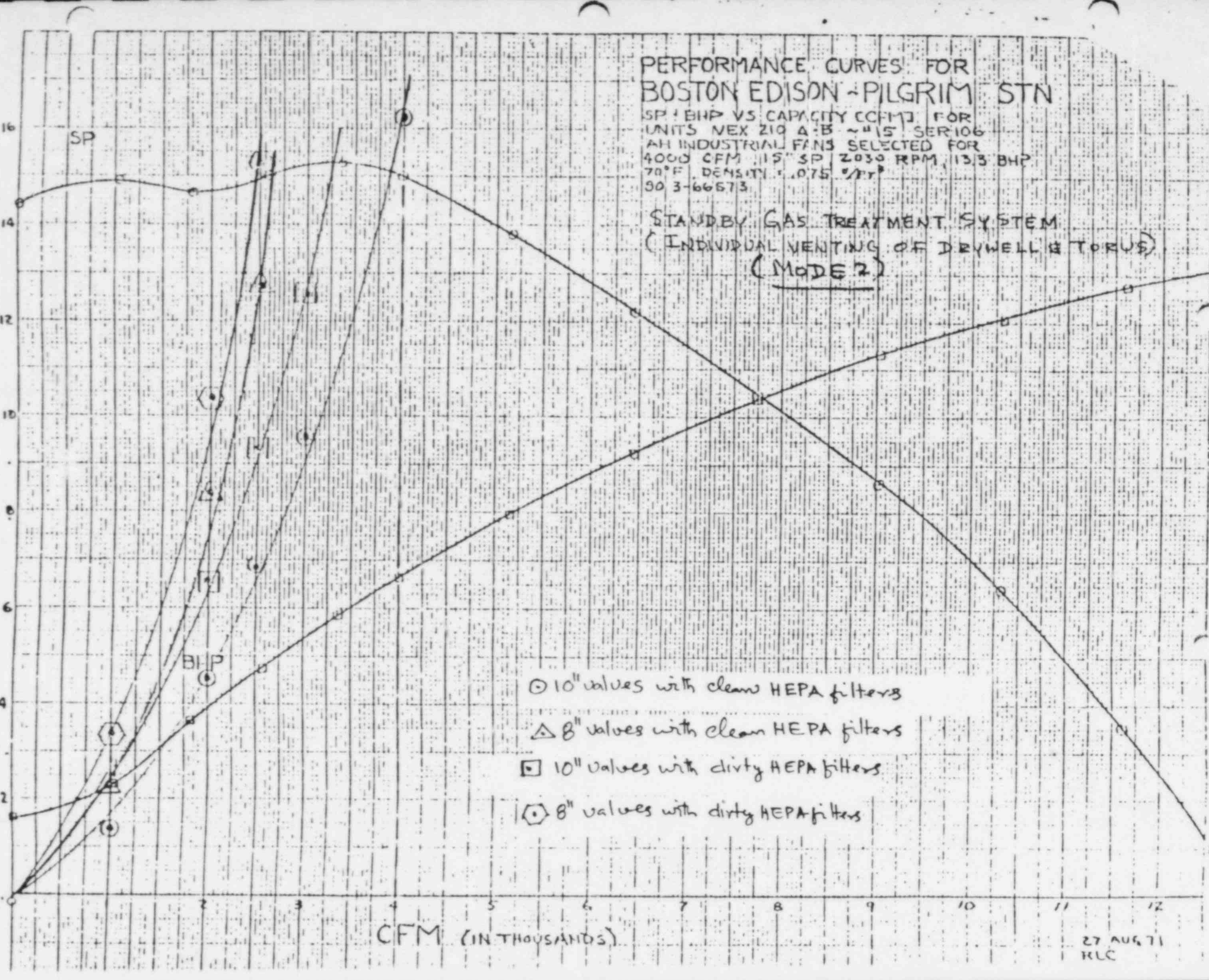
PI0556

Enclosure 2 to BLE-1417
Sheet 3 of 4



PI0556

Enclosure 2 to BLE-147
Sheet 4 of 4



PI0556

VALVE NO.	AO - 5035 A/B 5036 A/B	AO - 5042 A/B 044 A/B
SERVICE	Containment Isol. Purge and Vent Sys.	Containment Isol. Purge and Vent Sys.
VALVE TYPE	Butterfly	Butterfly
LINE OR EQUIPMENT REF.		
VALVE CLASS	HBB	HBB
SIZE	8"	8"
COMMODITY *	Air and Nitrogen	Air and Nitrogen
DESIGN/MAX. PRESS.	56 PSIG	56 PSIG
DESIGN/MAX. TEMP.	350°F	350°F
FLOW - NORMAL / MAX. **	2000 / 4000 SCFM	2000/4000 SCFM
VALVE RATING	150# ASME	150# ASME
TYPE ENDS/RATING	150# RF FLANGES	150# RF FLANGES
BODY MATERIAL	CARBON STEEL	CARBON STEEL
TRIM MATERIAL		
SEAT FACINGS		
PACKING	Grafoil	Grafoil
TYPE BONNET	—	—
TYPE OF SEATS		
TYPE OF DISC		
ACTUATOR		
OPER. DIFF. PRESS (MAX)	56 PSIG	56 PSIG
MAX. SEC. TO OPEN/CLOSE	5 sec	5 sec
LOCATION @ AMB. TEMP	104°F	104°F
IN CONTAINMENT - YES/NO	NO	NO
SAFETY RELATED - YES/NO	Yes	YES
SEISMIC CAT. 1 - YES/NO	YES	YES
THROTTLING SERVICE - YES/NO	No	No
TORQUE BACK SEATING - YES/NO	—	—
TORQUE OUTPUT MAX (MAX)		
REMOTE POSITION INDICATOR	LIMIT SWITCH	LIMIT SWITCH
FAIL SAFE MODE: Open/Clsd/Lkd	Fail close	Fail Close
Class 1 (IEEE)	YES	YES
Active Yes/No	Yes	Yes
Air Supply Pressure	70-100 PSIG	70-100 PSIG
Conservative leak dir. (By Supplier)		
Solenoid Voltage	120 VAC	120 VAC
MANUFACTURER		
MODEL OR FIG. NO.		
VENDOR		
P/O (& ITEM) NO.	1.1	1.1
FOREIGN PRINT NO.		
WELD END DWG. REFERENCE		
P & I DIAGRAM REF.		
LOCATION DWG. REF.		

REMARKS

* Air and Nitrogen may be saturated with steam.
** MAX. values are approx. only. Supplier will calculate actual values as reqd by Para. 6.4 of App 3. to Des. Spec. 8031-P-144.

REV.	DESCRIPTION	ENG	DR	CHK	SUPV	MATL	APPROVALS	DATE
0	ISSUED FOR USE	MJO						

PI-9556

VALVE DATA SHEET

~~UNIT NO 1~~

PILGRIM ~~STATION~~ STATION NO. 600
BOSTON EDISON COMPANY

BECHTEL
TPO

JOB NO. 10394

Requisition No. 10394-P-

REV. 0

SHEET 1 OF 2

3-19-74

Enclosure ... RIF - 1417

VALVE NO.	AO-5040 A/B		
SERVICE	Containment Isol. Purge and Vent Sys.		
VALVE TYPE	Butterfly		
LINE OR EQUIPMENT REF.			
VALVE CLASS	HBB		
SIZE	8"		
COMMODITY *	Air and Nitrogen		
DESIGN/MAX. PRESS.	56 psig		
DESIGN/MAX. TEMP.	350 °F		
FLOW - Normal / Max **	2000 / 4000 SCFM		
VALVE RATING	150# ASME		
TYPE ENDS/RATING	150# RF Flanges		
BODY MATERIAL	Carbon Steel		
TRIM MATERIAL			
SEAT FACINGS			
PACKING	Grafoil		
TYPE BONNET			
TYPE OF SEATS			
TYPE OF DISC			
ACTUATOR			
OPER. DIFF. PRESS (MAX)	56 psig		
MAX. SEC. TO OPEN/CLOSE	5 sec		
LOCATION @ AMB. TEMP	104 °F		
IN CONTAINMENT - YES/NO	No		
SAFETY RELATED - YES/NO	Yes		
SEISMIC CAT. 1 - YES/NO	Yes		
THROTTLING SERVICE - YES/NO	No		
TORQUE BACK SEATING - YES/NO	—		
TORQUE OUTPUT MAX (MAX)			
REMOTE POSITION INDICATOR	Limit Switch		
FAIL SAFE MODE: Open/Clsd/Lkd	Fail open		
Class 1 (IEEE)	Yes		
Active Yes/No	Yes		
Air Supply Pressure	70-100 psig		
Conservative Leak dir. (By supplier)			
Solenoid Voltage	120 VdC		
MANUFACTURER			
MODEL OR FIG. NO.			
VENDOR			
P/O I& ITEM NO.	1.2		
FOREIGN PRINT NO.			
WELD END DWG. REFERENCE			
P & I DIAGRAM REF.			
LOCATION DWG. REF.			
REMARKS	* Air and Nitrogen maybe saturated with steam. ** Max values are approx. only. Supplier will calculate actual values as req'd by para. 6.4 of App 3 to Design Spec. 8031-P-144.		

REV.	DESCRIPTION	ENG	DR	CHK	SUPV	MATL	APPROVALS	DATE
0	ISSUED FOR USE	MO						

PI0556

VALVE DATA SHEET

BECHTEL

TPO

~~UNIT NO. 1~~

PILGRIM ~~STATION~~ NO. 600

BOSTON EDISON COMPANY

JOB NO. 10394

Requisition No. 10394-P-

REV. 0

SHEET 2 OF 2

3-19-74

End of ... table — 1417

Attachment No. 5

References

(Note: Numbers in parentheses represent Bechtel Document No./Transmittal No. for those documents reviewed by Bechtel)

A. Seismic Analysis Reports

1. Technical Report PEI-TR-83-24, Rev. A. Seismic Qualification Analysis of Clow 8-Inch Wafer Stop Valve. (10394-M-119-1-13-1/SFP 26663)

2. Technical Report PEI-TR-833700-1, Addendum to PEI Technical Report PEI-TR-83-24 covering 8" - HBB-BF-AO-5035A, 5035B, 5036A, 5036B, 5042A, 5042B, 5044A, and 5044B. (10394-M-119-1-18-1/SFP 37237)

3. Technical Report PEI-TR-833700-1, Revision A (10394-M-119-1-18-2/SFP 26688)

B. Seismic Qualification Test Reports

1. Report No. 2-59700/1R-52972 "Simultaneous Static Seismic Load of Flow Interruption Capability Tests of a 12 Inch Valve for the Clow Corporation" (December 15, 1981). Application of 11.0 g biaxial static load to valve actuator during operation with choked air flow through the valve. (10394-M-119-1-6-1/SFP 26663)

2. Patel Report PEI-TR-83-29, Revision A (August 10, 1983) "Seismic Qualification of Clow Wafer Stop Valve Assemblies" including Addendum I and II. (10394-M-119-1-22-1/SFP 26663)

3. Bettis-37274, Nuclear Qualification Test Report, August 12, 1980. (10394-M-119-1-24-1/SFP 26601)

4. Namco-QTR-105, Qualification of EA180 Series Limit Switches, August 28, 1980. (10394-M-119-1-30-1/SFP 26663)

5. Asco-AQR67368, Rev. 0, Report on Qualification of Automatic Switch Co., (ASCO) Catalog NP-1 Solenoid Valves for Safety-Related Applications In Nuclear Power Generating Stations, March 2, 1982. (10394-M-119-1-21-1/SFP 26663)

C. Air Flow Tests

1. Final report on the Clow Valve Analysis Program CVAP (October 1981). Report covers methods of analysis, development of data base from model tests, and set-up of computer program to predict characteristics of full size valves. (10394-M-119-1-12-1/SFP 26601)

2. "Aerodynamic Torque And Mass Flow Rate for Compressible Flow Through Geometrically Similar Scale-Model Clow Valves in Series." (October, 1982) (10394-M-119-1-10-1/SFP 26601)

D. Other Reports and Information

1. Operating Instructions for Clow Tricentric Wafer Stop Valve covers installation, maintenance, and operating instructions for 82-2739(N) valves. (10394-M-119-1-25-1/SFP 26664)

2. Clow Test Report Projet No. 82-003 "Effects f Foreign Bodies on Tricentric Sealing" by Robert Sansone. (10394-M-119-1-8-1/SFP 26601)

3. Shell International Cycling Test (2/6/72) by M. Nijenhuis (Note: CLow produces Tricentric valves under license of Gebruder Adams of Bochum, West Germany.) (10394-M-119-1-50-1/SFP 26688)

E. Other References

1. Bechtel Power Corporation Design Specification 10394-P-119-1(Q), Rev. 0.

2. "A Water Table Investigation of Two-Dimensional Models of The Clow Corporation Tricentric Valve" by Dr. Robert F. Hurt, Engineering Consultant, Professor of Mechanical Engineering, Bradley University, Peoria, Illinois, September 14, 1979.

3. "A Parametric Study of A Butterfly Valve Utilizing The Hydraulic Anology" by Bruce A. Coers, Bradley University, 1983.

4. "Radiation Sensitivity Analysis of Laminated Valve Seals for Clow Corporation." Wyle No. 17629-01 (January 31, 1983).
(10394-M-119-1-7-1/SFP 26601)