CLOW

Clow Corporation
Engineered Products Division

40 Chestnut Avenue
Westmont, IL 60559

312 789-8900

PURGE AND VENT VALVE OPERABILITY

QUALIFICATION ANALYSIS

Report No. 7-25-85

PREPARED FOR

COMMONWEALTH EDISON CO.
LASALLE COUNTY STATION
UNITS #1 & #2

by

Steve Nondahl

July 1985

Work performed under Commonwealth Edison Purchase Order Number 289825 Rev.A
Project No. 6854-30

Clow Job Numbers: 84-2842-01 and -02(N)

This report covers valve tag numbers:

1VQ026	1VQ034	2VQ026	2VQ034
1VQ027	1VQ036	2VQ027	2VQ036
1VQ029	1VQ040	2VQ029	2VQ040
1VQ030	1VQ042	2VQ030	2VQ042
1VQ031	1VQ043	2VQ031	2VQ043

REV. NO.	REV. DATE	REV. BY	CHECKED BY	Q.A. BY	DESCRIPTION OF CHANGES AND PAGES REVISED
0	7-25-85	---	---	--	Original Issue

CERTIFICATION

This is to certify that all valves (Tag Nos. 1VQ026, 1VQ027, 1VQ029, 1VQ030, 1VQ031, 1VQ034, 1VQ036, 1VQ040, 1VQ042, 1VQ043, 2VQ026, 2VQ027, 2VQ029, 2VQ030, 2VQ031, 2VQ034, 2VQ036, 2VQ040, 2VQ042, 2VQ043) have been evaluated for operability under the installed conditions indicated in Commonwealth Edison Co. Purchase Order 289825, Rev. A and accompanying specifications as amended by Clow exceptions. 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.

Paper written and analyses by Steven R. Nondahl
Steven Nondahl
Design Eng. Mgr., Nuclear
Clow Corp.

Reviewed by William J. Allen
William J. Allen
Design Engineer, Nuclear
Clow Corp.

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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 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 (see Appendix A). This paper addresses those items which may be answered by the valve manufacturer based on the conditions provided by the pla operator for the postulated loss of coolant accident.

This paper describes the design of 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 is 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 setups 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 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^{0}, other angles, short radius, long radius), tees, reducers), a worst case had to be selected for evaluation. A 90^{0} 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 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 the scale models of the 24" and 48" valves 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 summary of this test). 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 (in the test all torques tended to close the valve) as measured in model tests.

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 listed in 7.0 References. It is advised that the documents listed in 7.0 be available for reference during review of this report.

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 Senitivity 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 hydrostatic end load only)
5. Actuator mounting structure
   a. Adaptor flange
   b. Bolting

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

It should be noted that for this application one generic report has been provided for the 8" valve (PEI-TR-83-24) and one for the 26" valve (PEI-TR-852200-1). In addition, a summary report for the 8" and 26" valves shows how the generic reports and the actuator and solenoid valve qualification reports encompass Sargent & Lundy/CECO spec. requirements.

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 an analysis of the torque characteristics of

the subject valve. 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 air supply, or maximum backpressure for pneumatic actuators if applicable).

The following method is used to show operability:

1. Determine the 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 the predicted torque values for all disc angles based on 2 and 3 above.
5. Provide a tabulation or plot of actuator output torque for all actuator angles.
6. Show that the 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 preceding calulations, the following assumptions are employed:

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

b. The pressure downstream of the valve is atmospheric. In the elbow experiment it was noted that downsteam 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) used as a basis for the analysis.

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 exceeds that needed to assure valve closure for the maximum aerodynamic torques which could occur under LOCA conditions.

2.0 DESIGN OF VALVE AND ACTUATOR ASSEMBLY

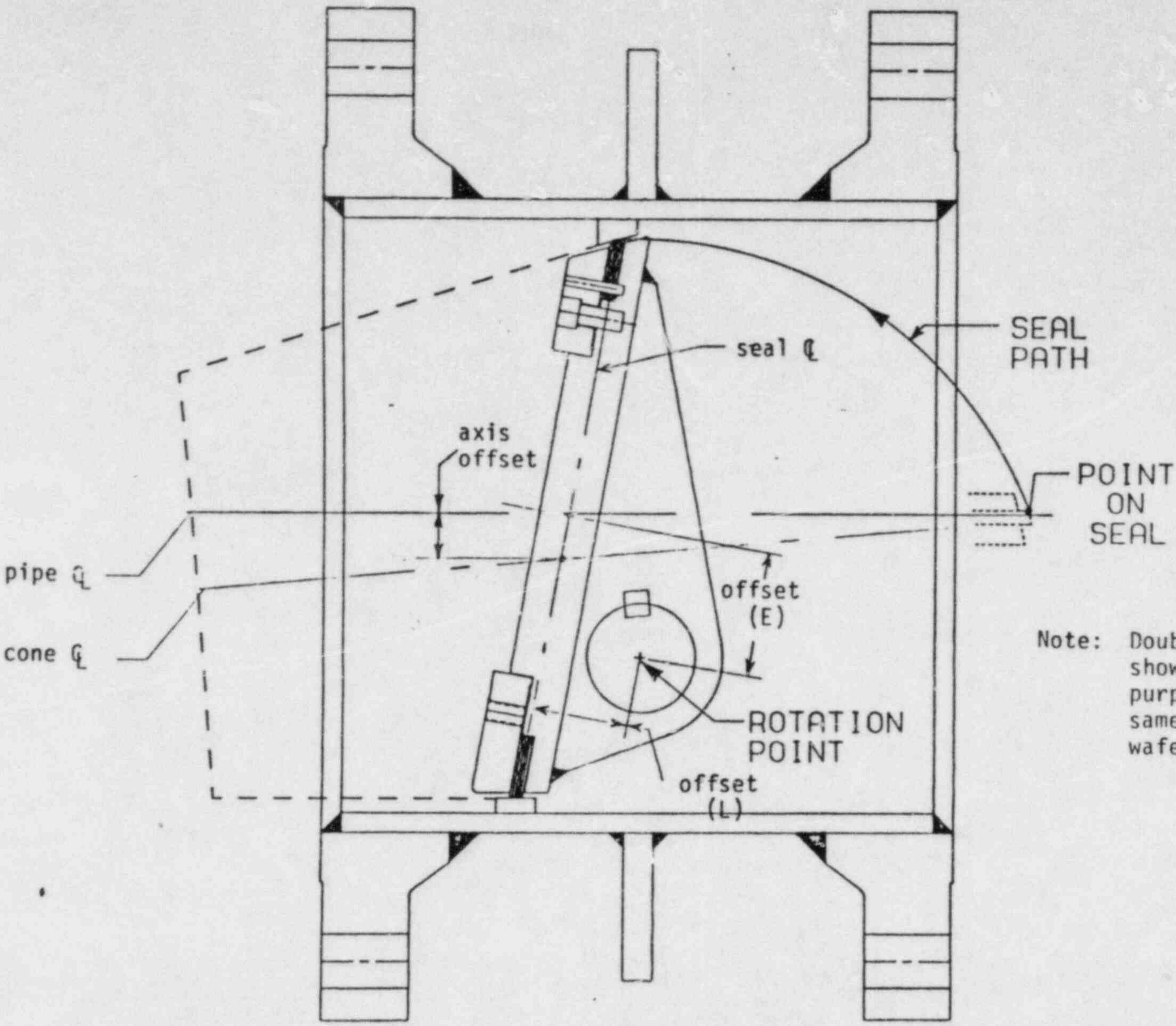
2.1 Valve Design

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 offsets apply to wafer style.

FIGURE 1 - TRI⌊⌋ENTRIC 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-sea 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.

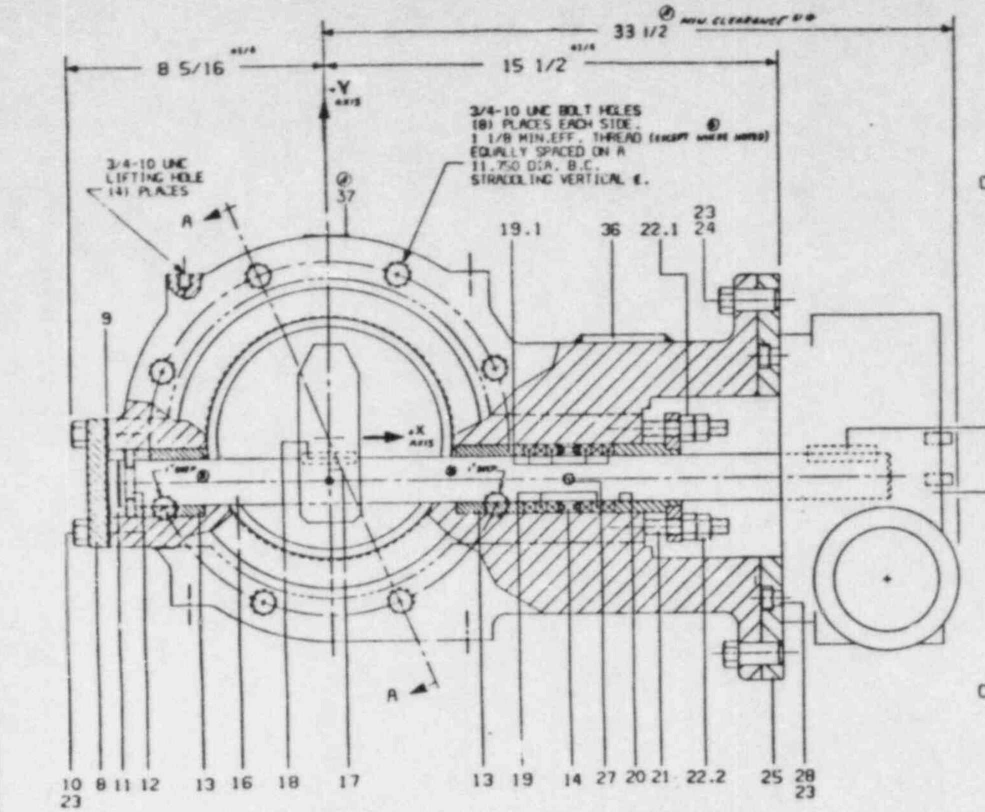
All of these features have been incorporated into the lugged wafer body that results in a very rugged and sturdy valve design capable of meeting or exceeding all the requirements set forth in the specification.

2.1.2 Materials

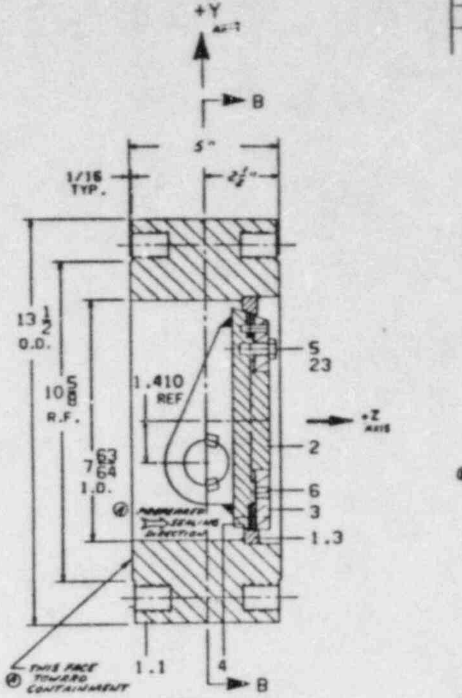
A complete list of valve component materials used on Commonwealth Edison Co. Purchase Order 289825 may be found on the General Arrangement Drawings (D-0804 and D-0805) which follow this section.

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 SA479 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 GR70) 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. 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 Klinger K-61 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 seal and therefore, results in application of higher normal stresses to the seal for any given seating torque.



SECTION B-B

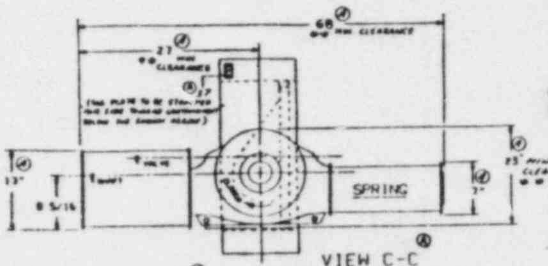


SECTION A-A

■ DENOTES SAFETY RELATED NON-PRESSURE BOUNDARY PARTS.
△ DENOTES PRESSURE RETAINING PARTS.
✱ DENOTES RECOMMENDED SPARE PARTS.
● DENOTES PARTS *GREATER THAN 50 LBS.*

	ITEM	DESCRIPTION	QTY	MATERIAL	
	37	TAG PLATE	1	[illegible]	[illegible]
	36	NAME PLATE	1	[illegible]	[illegible]
■	33	PARALLEL KEY	1	C-1045 COLD DRAWN	[illegible]
	28	SOC.HD.CAP SCRW	4	[illegible]	[illegible]
△	27	PIPE PLUG	1	[illegible]	[illegible]
●■	26	ACTUATOR	1	COMM.	[illegible]
	25	ADAPTOR PLATE	1	[illegible]	
	24	HEX HD.CAP SCRW	4	[illegible]	[illegible]
	23	LOCKWIRE	A/R	[illegible]	
	22.2	JAM NUT	4	COMM.	
△	22.1	NUT	4	[illegible]	
△	21	STUD	4	[illegible]	
△	20	GLAND	1	[illegible]	[illegible]
✱	19.1	PACKING	4	[illegible]	[illegible]
✱	19	PACKING	2	[illegible]	[illegible]
■	18	PARALLEL KEY	2	C-1045 COLD DRAWN	[illegible]
	17	DOWEL PIN	1	COMM. ALLOY STEEL	[illegible]
△	16	DRIVE SHAFT	1	[illegible]	[illegible]
	14	LANTERN RING	1	[illegible]	[illegible]
✱	13	BEARING	2	CARBON	[illegible]
✱	12	ANNULAR KEY	1	[illegible]	
	11	SPACER	1	[illegible]	[illegible]
△	10	HEX HD.CAP SCRW	4	[illegible]	[illegible]
✱	9	GASKET	1	[illegible]	[illegible]
△	8	COVER PLATE	1	[illegible]	
	6	SOC. SET SCRW	2	COMM.	
■	5	HEX HD.CAP SCRW	12	[illegible]	[illegible]
✱	4	LAMINATED SEAL	1	[illegible]	[illegible]
■	3	CLAMP RING	1	[illegible]	
△	2	DISC	1	[illegible]	
■	1.3	VALVE SEAT	1	[illegible]	
●△	1.1	VALVE BODY	1	[illegible]	

DESCRIPTION	ITEM	SIZE	TORQUES REQUIRED
DISC BOLTS	5	3/8-16 UNC	18-22 FT-LB.
COVER PLATE BOLTS	10	1/2-13 UNC	40-60 FT-LB.
GLAND NUTS	22.1	1/2-13 UNC	10-70 FT-LB.
ADAPTOR PLATE MTG BOLTS	24	3/4-10 UNC	125-150 FT-LB.
ACTUATOR MOUNTING BOLTS	28	7/8-9 UNC	200-250 FT-LB.



VIEW C-C

ACTUATOR ORIENTATION FOR ALL VALVES IS AS SHOWN.

DESIGN INFORMATION

1. CODE REFERENCE: CLASS 2, SECTION III OF ASME BOILER AND PRESSURE VESSEL CODE, 1983 EDITION INCLUDING ADDENDA THROUGH SUMMER, 1984.
2. FLANGE BOLTING DIMESIONS PER 8" - CLASS 150 B16.5 WITH 1/16" RAISED FACE.
3. DESIGN PRESSURE:
   BODY: 285 PSIG, RATING CLASS: 150
   DISC: *60* PSID
4. HYDROSTATIC PRESSURE TESTS:
   SHELL TEST: 450 *PSIG*
   SEAT TEST:
5. VALVE WEIGHT: 275 LBS. (APPROX. W/O ACTUATOR)
6. ACTUATOR WEIGHT: *430* LBS. (APPROX.)
7. *DESIGN TEMP: 340 °F*
8. MINIMUM THICKNESSES:
   BODY WALL: *.59*
   ABOVE DIMENSION INCLUDES *.08* MIN. COR. ALLOW.
   BEARING NECK: *.158*
   (THE MINIMUM WALL BETWEEN THE BOLT HOLE AND BEARING BORE)

*** - MINIMUM CLEARANCE FOR COMPONENT ASSY./DISASSEMBLY PHYSICAL PARTS DIMENSIONS ARE EQUAL TO OR SMALLER.*

JOB INFORMATION

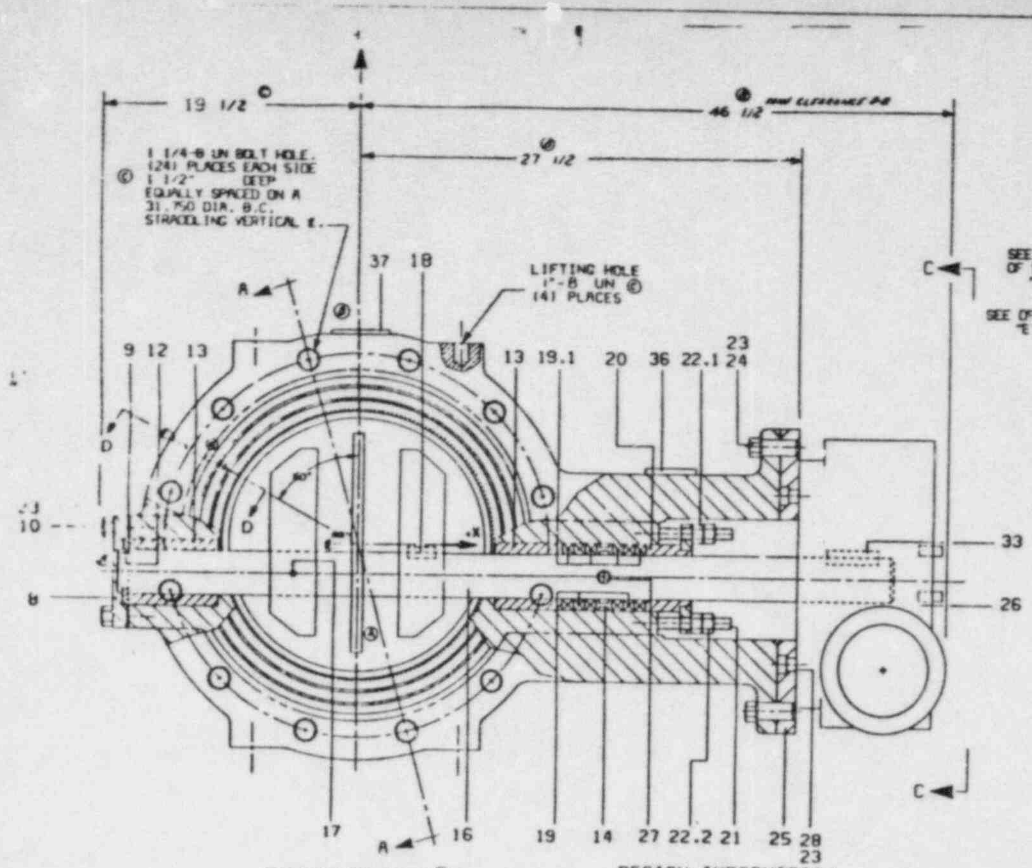
CUSTOMER: COMMONWEALTH EDISON CO.
LASALLE COUNTY STATION UNIT 1&2
PO. NO.: 289825
SPEC. NO.: SARGENT & LUNDY T-3750
CLOW JOB NO.: 84-2842-02(N)
SERVICE: VQ CONTAINMENT ISOLATION

	CLOW SERIAL NO.(S)	VALVE TAG NO.(S)	C.G. DIMENSIONS X	Y	Z
UNIT #1	84-2842-02(N)-01	1VQ042	[illegible]	[illegible]	-1.75
	84-2842-02(N)-02	1VQ043			
UNIT #2	84-2842-02(N)-03	2VQ042	[illegible]	[illegible]	-1.75
	84-2842-02(N)-04	2VQ043			

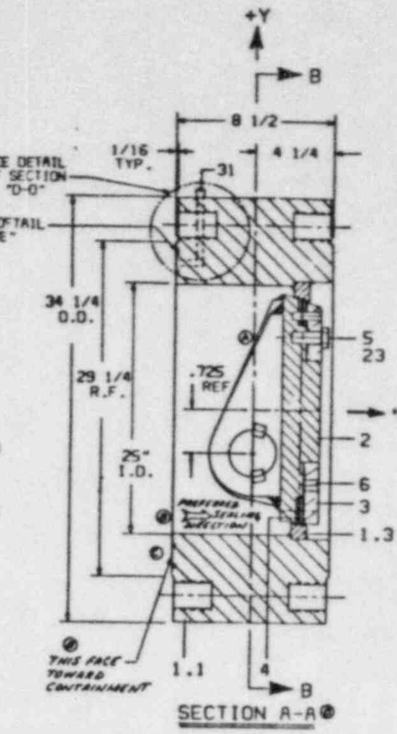
MATERIAL N/A	DO NOT SCALE PRINT	DRAWN Hung 10-31-84	CLOW
HEAT TREAT N/A	TOLERANCES	CHECKED J.K. 10-31-84	TITLE 8" WAFER STOP VALVE ASSEMBLY
N/A	4	APPROVED I.R. 11-16-84	D-0804
SCALE NTS			JOB NO. 84-2842-02

FIGURE 2 - 8" AIR OPERATED VALVE ASSEMBLY AND MATERIALS

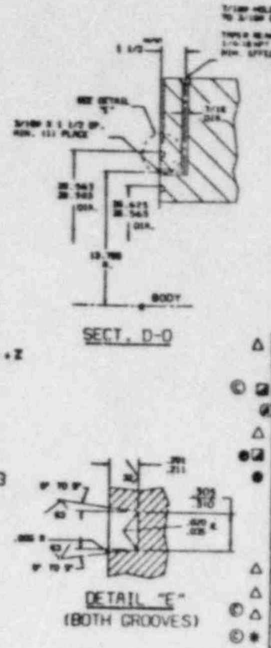
	CR	DESCR		
A	1838	ADDED DISC RIB TO SECT. A-A & SECT B-B	[illegible]	
B	1894	REVISED PER SCR # 1894	[illegible]	
C	1952	REVISED PER SCR # 1952	[illegible]	
D	1972	REVISED PER SCR # 1972	[illegible]	



SECTION B-B



SECTION A-A



■ DENOTES SAFETY RELATED NON-PRESSURE BOUNDARY PARTS.
△ DENOTES PRESSURE RETAINING PARTS.
✱ DENOTES RECOMMENDED SPARE PARTS.
● DENOTES PARTS GREATER THAN 50 LBS.

	ITEM	DESCRIPTION	QTY	MATERIAL	REMARKS
△	31	PIPE PLUG	1	[illegible]	[illegible]
	36	NAME PLATE	1	[illegible]	[illegible]
© ■	33	PARALLEL KEY	1	C-1045 COLD DRAWN	80,000 PSI MIN. TENSILE
	28	SOC.HD.CAP SCRW	8	[illegible]	TO BE LOCKWIRED TOGETHER
△	27	PIPE PLUG	1	[illegible]	[illegible]
● ■	26	ACTUATOR	1	COMM.	[illegible]
●	25	ADAPTOR PLATE	1	[illegible]	
	24	HEX HD.CAP SCRW	4	[illegible]	TO BE LOCKWIRED TOGETHER
	23	LOCKWIRE	A/R	[illegible]	
	22.2	JAM NUT	4	COMM.	
△	22.1	NUT	4	[illegible]	
△	21	STUD	4	[illegible]	
© △	20	GLAND	1	[illegible]	[illegible]
© ✱	19.1	PACKING	4	[illegible]	[illegible]
✱	19	PACKING	2	[illegible]	JOHN CRANE STYLE [illegible]
© ■	18	PARALLEL KEY	2	C-1045 COLD DRAWN	80,000 PSI MIN. TENSILE
	17	DOWEL PIN	1	[illegible] ALLOY STEEL	[illegible]
● △	16	DRIVE SHAFT	1	[illegible]	COND. H-1100
	14	LANTERN RING	1	[illegible]	[illegible]
✱	13	BEARING	2	CARBON	[illegible]
✱	12	ANNULAR KEY	1	[illegible]	
	37	TAG PLATE	1	SST	[illegible]
△	10	HEX HD.CAP SCRW	4	[illegible]	TO BE LOCKWIRED TOGETHER
© ✱	9	GASKET	1	[illegible]	ONE PC.
△	8	COVER PLATE	1	[illegible]	
	6	SOC. SET SCRW	2	COMM.	
■	5	HEX HD.CAP SCRW	24	[illegible]	TO BE LOCKWIRED TOGETHER
✱	4	LAMINATED SEAL	1	[illegible]	[illegible] ONE PC.
● ■	3	CLAMP RING	1	[illegible]	
● △	2	DISC	1	[illegible]	
■	1.3	VALVE SEAT	1	[illegible]	[illegible]
● △	1.1	VALVE BODY	1	[illegible]	

DESIGN INFORMATION

1. CODE REFERENCE: CLASS 2, SECTION III OF ASME BOILER AND PRESSURE VESSEL CODE, 1983 EDITION INCLUDING ADDENDA THROUGH SUMMER, 1984.
2. FLANGE BOLTING DIMENSIONS PER AWWA C-207, CLASS E WITH 1/16" RAISED FACE (SERRATED)
3. DESIGN PRESSURE:
   BODY: 285 PSIG - RATING CLASS: 150 LB.
   DISC: 60 PSID
4. HYDROSTATIC PRESSURE TESTS:
   SHELL TEST: 450 PSIG
   SEAT TEST: [illegible]
5. VALVE WEIGHT: 1800 LBS. (APPROX. W/O ACTUATOR)
6. ACTUATOR WEIGHT: 1515 LBS. (APPROX.)
7. DESIGN TEMP. : 340°F - OPERATING TEMP. : 135°F
8. MINMUM THICKNESSES:
   BODY: 0.69" (INCLUDED 0.08" FOR MIN. COR. ALLOW.)
   BEARING NECK: .23 (THE MIN. WALL BETWEEN THE FLANGE HOLE & BODY BEARING BORE.

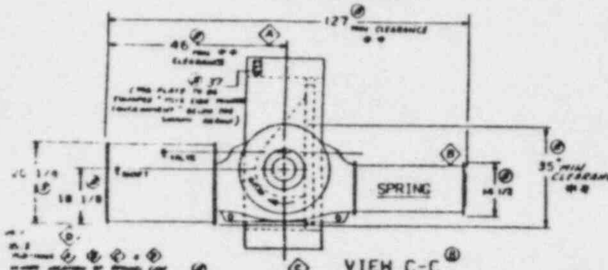
- MIN CLEARANCE FOR COMPONENT ASSY/DISASSEMBLY. PHYSICAL PARTS DIMENSIONS ARE EQUAL TO OR SMALLER.

JOB INFORMATION

CUSTOMER:
COMMONWEALTH EDISON CO.
LASALLE COUNTY STATION UNIT 1&2
P.O. NO.: 289825

SPEC. NO.: SARGENT & LUNDY T-3750

CLOW JOB NO.: 84-2842-01(N)
SERVICE: VQ CONTAINMENT ISOLATION



VIEW C-C

ACTUATOR SHOWN IN [illegible] POSITION

DESCRIPTION	ITEM	SIZE	TORQUES REQUIRED
DISC BOLTS	5	5/8-11 UNC	90-110 FT-LBS
COVER PLATE BOLTS	10	5/8-11 UNC	90-110 FT-LBS
GLAND NUTS	22.1	1/2-13 UNC	10-20 FT-LBS
ADAPTOR PLATE MTG BOLTS	24	1-8 UNC	225-250 FT-LBS
ACTUATOR MOUNTING BOLTS	28	3/4-10 UNC	125-150 FT-LBS

UNIT NO 1

CLOW SERIAL NUMBERS	VALVE TAG NO (S)	ACT POS.	C.G. DIMENSIONS X	Y	Z
84-2842-01(N)-01	1VQ026	C	17.345	-5.046	-5.110
84-2842-01(N)-02	1VQ027	B	17.345	-3.744	5.847
84-2842-01(N)-03	1VQ029	B	17.345	-3.744	5.647
84-2842-01(N)-04	1VQ030	A	17.345	5.020	3.487
84-2842-01(N)-05	1VQ031	D	17.345	2.842	-5.277
84-2842-01(N)-06	1VQ034	B	17.345	-3.744	5.647
84-2842-01(N)-07	1VQ036	D	17.345	2.842	-5.277
84-2842-01(N)-08	1VQ040	A	17.345	5.020	3.487

UNIT NO 2

CLOW SERIAL NUMBERS	VALVE TAG NO (S)	ACT POS.	C.G. DIMENSIONS X	Y	Z
84-2842-01(N)-09	2VQ026	D	17.345	2.842	-5.277
84-2842-01(N)-10	2VQ027	B	17.345	-3.744	5.647
84-2842-01(N)-11	2VQ029	B	17.345	-3.744	5.647
84-2842-01(N)-12	2VQ030	A	17.345	5.020	3.487
84-2842-01(N)-13	2VQ031	D	17.345	2.842	-5.277
84-2842-01(N)-14	2VQ034	B	17.345	-3.744	5.647
84-2842-01(N)-15	2VQ036	B	17.345	-3.744	5.647
84-2842-01(N)-16	2VQ040	A	17.345	5.020	3.487

MATERIAL: N/A	DO NOT SCALE PRINT	DRAWN: [illegible] 11-01-84	CLOW ENGINEERED PRODUCTS DIVISION
HEAT TREAT: N/A	TOLERANCES: DECIMAL ± .010; FRACTIONAL ± 1/8; ANGULAR ± [illegible]	CHECKED: [illegible] 11-1-84	TITLE: 26" WAFER STOP VALVE ASSEMBLY
QUANTITY: N/A; NO. REQ. PER: 16		APPROVED: I.R. 11-16-84	DRAWING NO.: D-0805 D
SCALE: NTS			JOB NO.: 84-2842-01(N)

FIGURE 3 - 26" AIR OPERATED VALVE ASSEMBLY AND MATERIALS

2.1.3 Operation

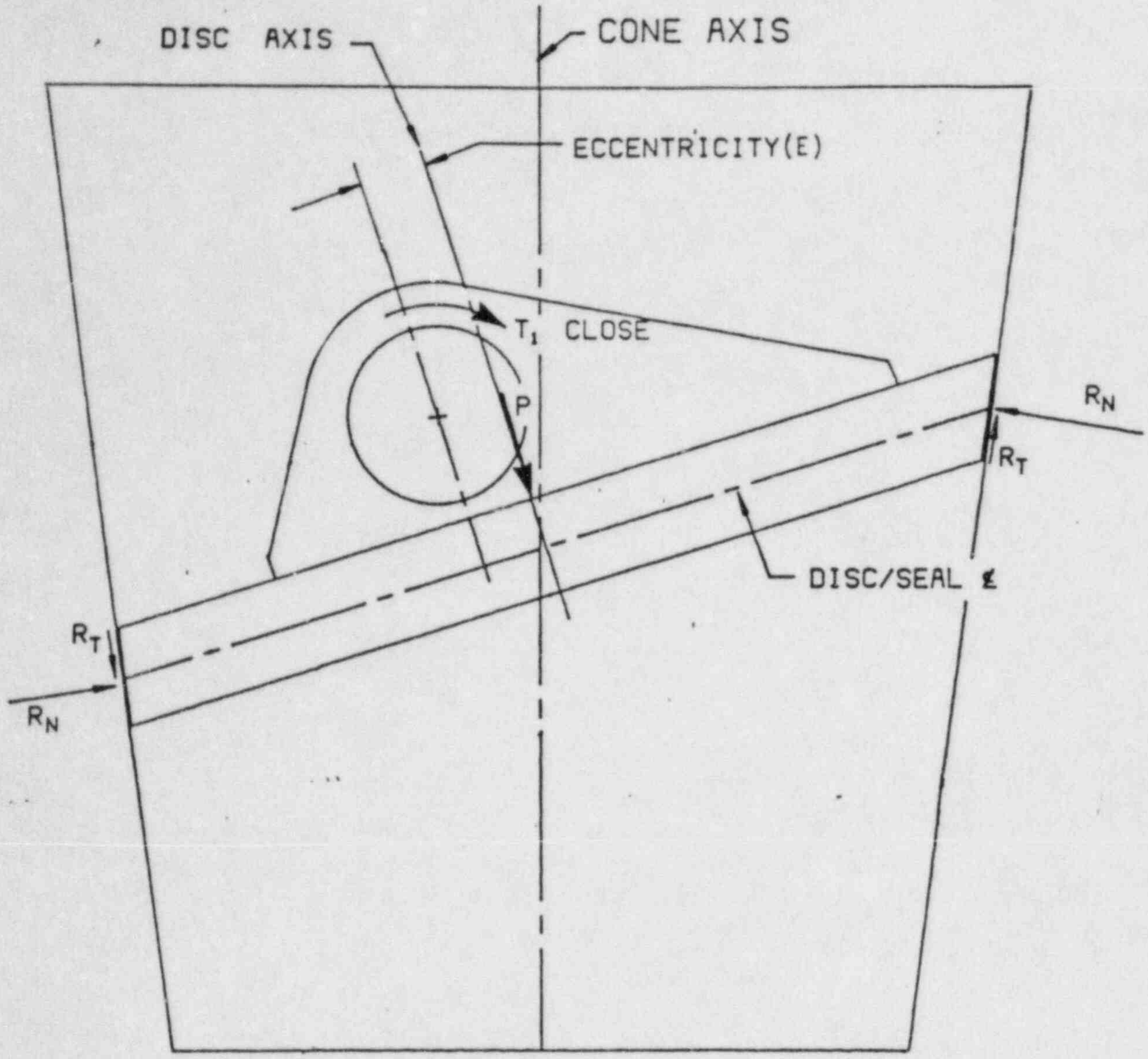
The operation of the Tricentric valve is extremely simple since there are only 2 moving parts, the disc assembly and the shaft. The valve operates by changing the position of the disc relative to the seat. This is accomplished through the application of a torque on the valve shaft through the entire operating range of 90 degrees. (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 the 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 forces the packing exerts on the shaft. These forces are a result of 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. (Figure 4) 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 Figure 5, 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 at the point where R_N is a minimum, a sufficient loading must be applied 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 4

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.. Typically 1.2 to 1.5 times the PAM torque is required to unseat the valve.

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) the running torque was approximately 1000 in.lbs. This is seen in Figure. 8 Run 1 and Fig. 15 Run 8 (of the Vought Report) 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. Torques for the 8" valve on this order would be about the same. For the 26" valves these torques are expected to be less than 10% of the actuator output torque.

2.2 Actuator Design (Pneumatic Spring Return)

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 5.

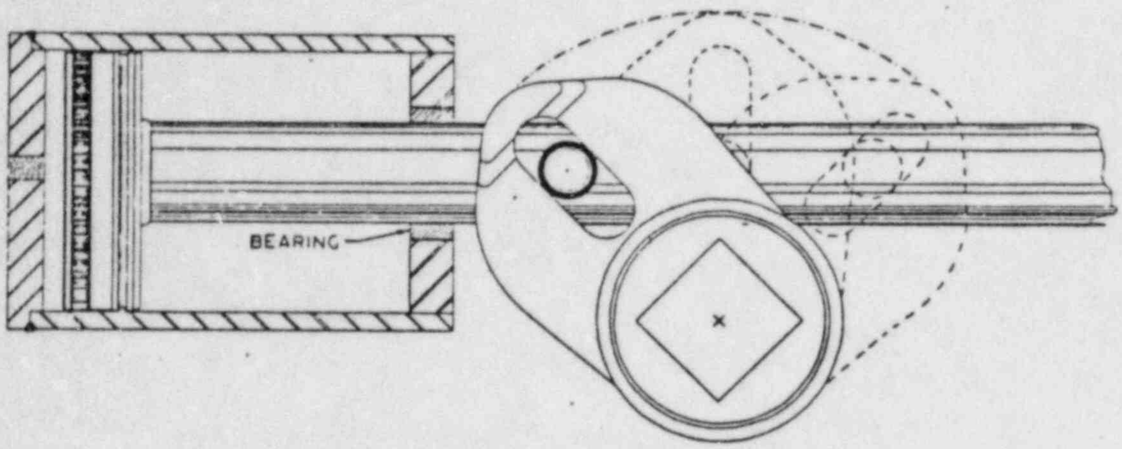


FIGURE 5 - ACTUATOR SCOTCH YOKE DESIGN

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^{o} 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^{o}.

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

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

Where:

T = Torque in in-lb

F = Resultant total force in pounds = P X A

$\frac{MA}{\text{Cos}.\ 45^{\circ}}$ = 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 6.

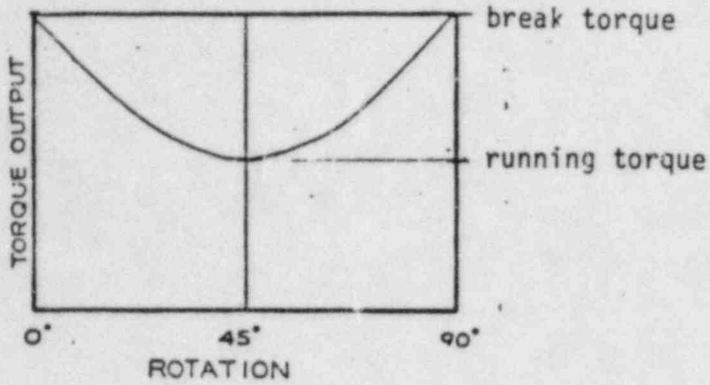


FIGURE 6 - Typical torque output for double acting scotch yoke actuator.

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.

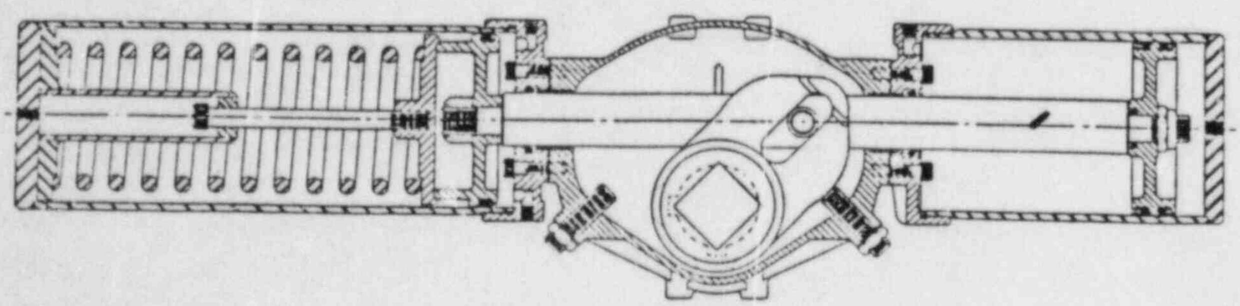


FIGURE 7 - Fail safe, spring return actuator design

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.

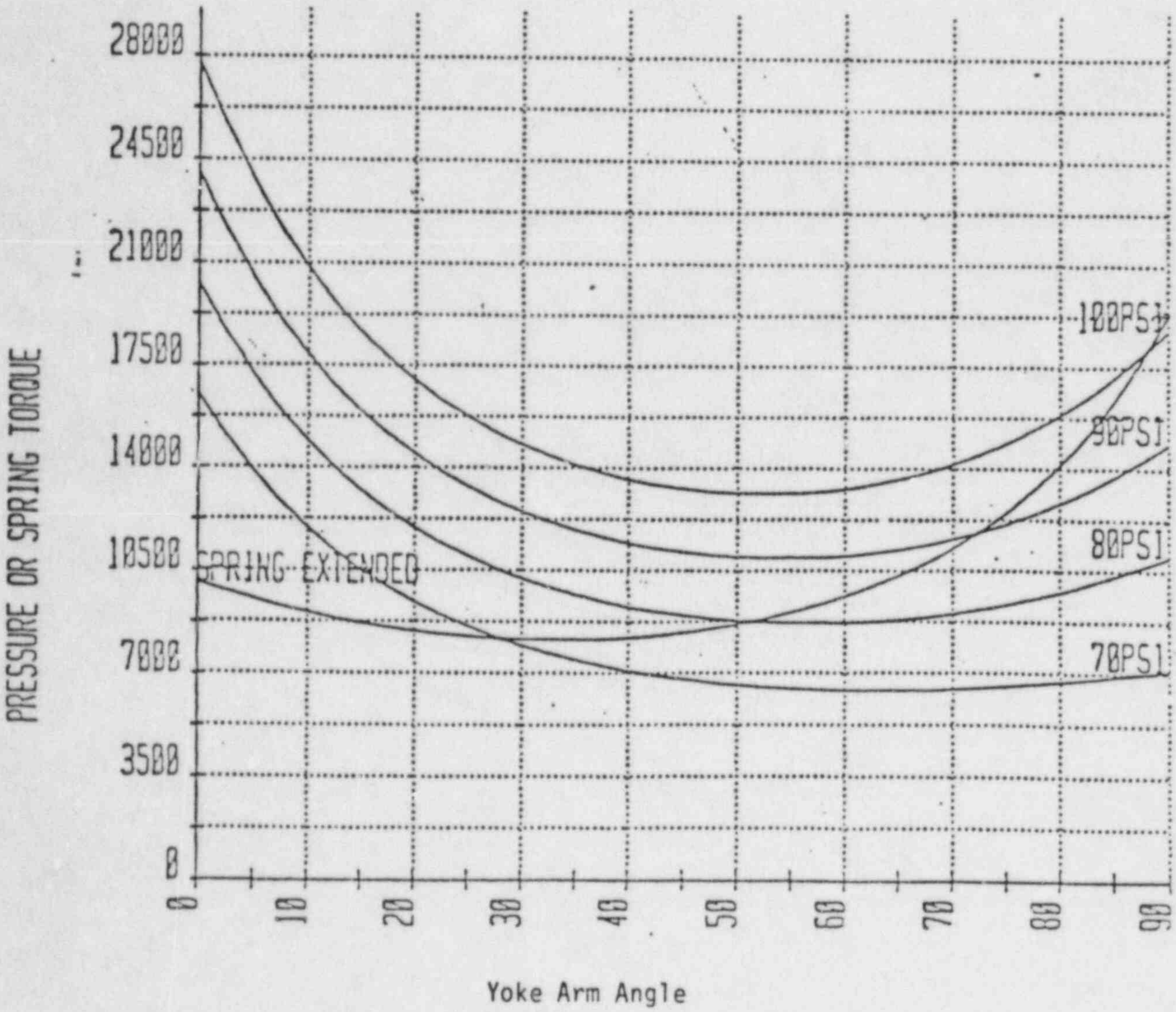


FIGURE 8 - Typical torque output curves for a spring return actuator

2.2.2 Actuator Design Materials

The Bettis actuators used for this job are the T 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. These actuators incorporate use of special materials for nuclear service as listed below.

Special Material:

Grease - Molykote 44 (medium grade)

Seals - Ethylene Propylene (certified to 1.4 x 10^8 rads)

Internal cylinder coating - Molybdenum disulfide

Yoke pin and rollers - Ryton coated

It should also be noted that since these units are of the fail safe type, the spring is a critical safety component. All springs supplied on this order were 100% magnaflux inspected to insure the spring quality.

2.2.3 Actuator and Valve Operation (Pneumatic/Spring Return)

2.2.3.1 Actuators and Accessories Supplied

A complete list of all accessories used on each pneumatically actuated 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 120 VAC 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 used is a NPL8316E34E. This valve is designated for use in nuclear power applications which consists of providing IEEE compliance and a waterproof solenoid enclosure. It is a high flow valve which has 1 in. NPT ports and a 1 in. orifice. All elastomeric materials of construction are Ethylene Propylene material.

Limit switches are also provided, mounted on the actuator to indicate full open or closed position. One of each model no. switch is provided, one set for the open position and the other set for the closed position. The switch model nos. are Namco EA 180-31302 and EA 180-32302 which are DPDT switches with 2 normally open and 2 normally closed contacts and are quick make-quick break type. The switches are of the spring return type with one model being CW

operation and the other CCW operation. Both switches use the same lever arm which is a Namco model EL-060-53300.

TABLE 1

PNEUMATIC ACTUATED UNITS

VALVE SIZE (IN)	TAG NOS.		CLOW JOB NO.	BETTIS ACTUATOR MODEL NO.	FAIL-SAFE ROTATION (viewed from top of unit)	FAIL-SAFE VALVE POSITION	ACTUATOR ACCESSORIES: ASCO SOLENOID VALVE MODEL NO.	ACTUATOR ACCESSORIES: NAMCO LIMIT SWITCHES AND LEVER ARM MODEL NOS. (2 closed position switches) (2 open position switches)
26"	1VQ026 1VQ027 1VQ029 1VQ030 1VQ031 1VQ034 1VQ036 1VQ040	2VQ026 2VQ027 2VQ029 2VQ030 2VQ031 2VQ034 2VQ036 2VQ040	84-2842-01(N)	NT820-SR3	CW	Close	NP8316E34E (Qty. 1)	EA 180-31302 L.S. EA 180-32302 L.S. EL 060-53300 L.A.
8"	1VQ042 1VQ043 2VQ042 2VQ043		84-2842-02(N)	NT312-SR3	CW	Close	NP8316E34E (Qty. 1)	EA 180-31302 L.S. EA 180-32302 L.S. EL 060-53300 L.A.

2.2.3.2 Pneumatic Actuator Output Torques

The torque plots provided in this section represent the calculated output torque of the actuators for the spring and various supply pressures shown. The guaranteed output torques that Bettis provides is for the yoke arm at 0 degrees and the spring fully extended. The ratio of guaranteed torque to calculated torque is shown below for the two actuator sizes used.

TABLE 2

ACTUATOR MODEL	GUARANTEED TORQUE/CALCULATED TORQUE	%
NT820-SR3	129,035/136,934	94
NT312-SR3	13,400/13,900	96

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 from the actuator manufacturer.

FIGURE 9A CALCULATED TORQUE DATA

T820-SR3

	SPRING	@ 80	@ 90
0	129,035	151,954	192,915
10	108,197	114,149	147,255
20	96,317	92,864	121,492
30	90,326	80,243	106,375
40	88,888	72,843	97,848
50	91,621	69,066	94,087
60	98,764	68,405	94,686
70	111,824	70,697	99,794
80	134,264	76,182	110,414
90	173,968	85,343	128,801

FIGURE 9B CALCULATED TORQUE PLOT

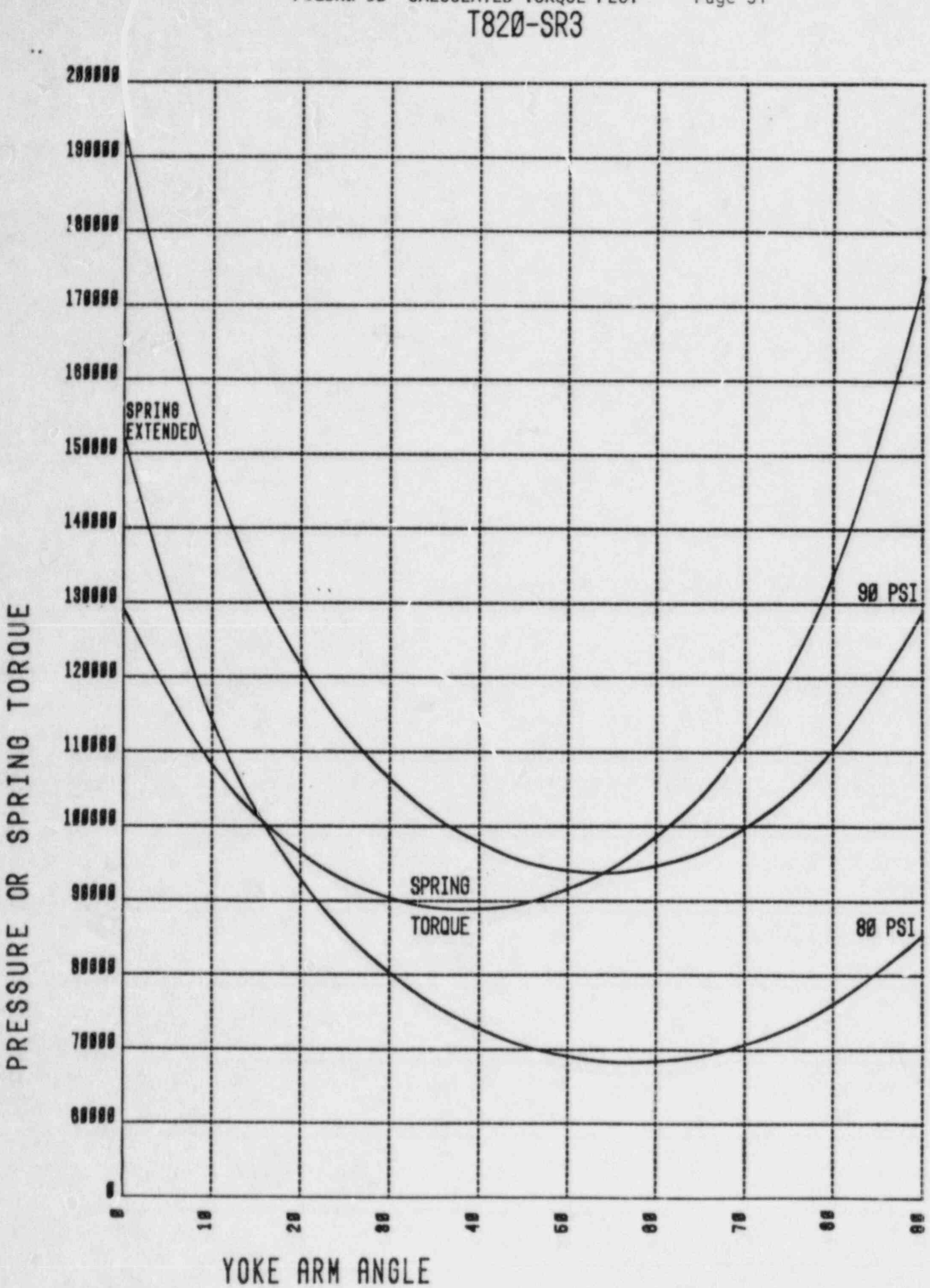


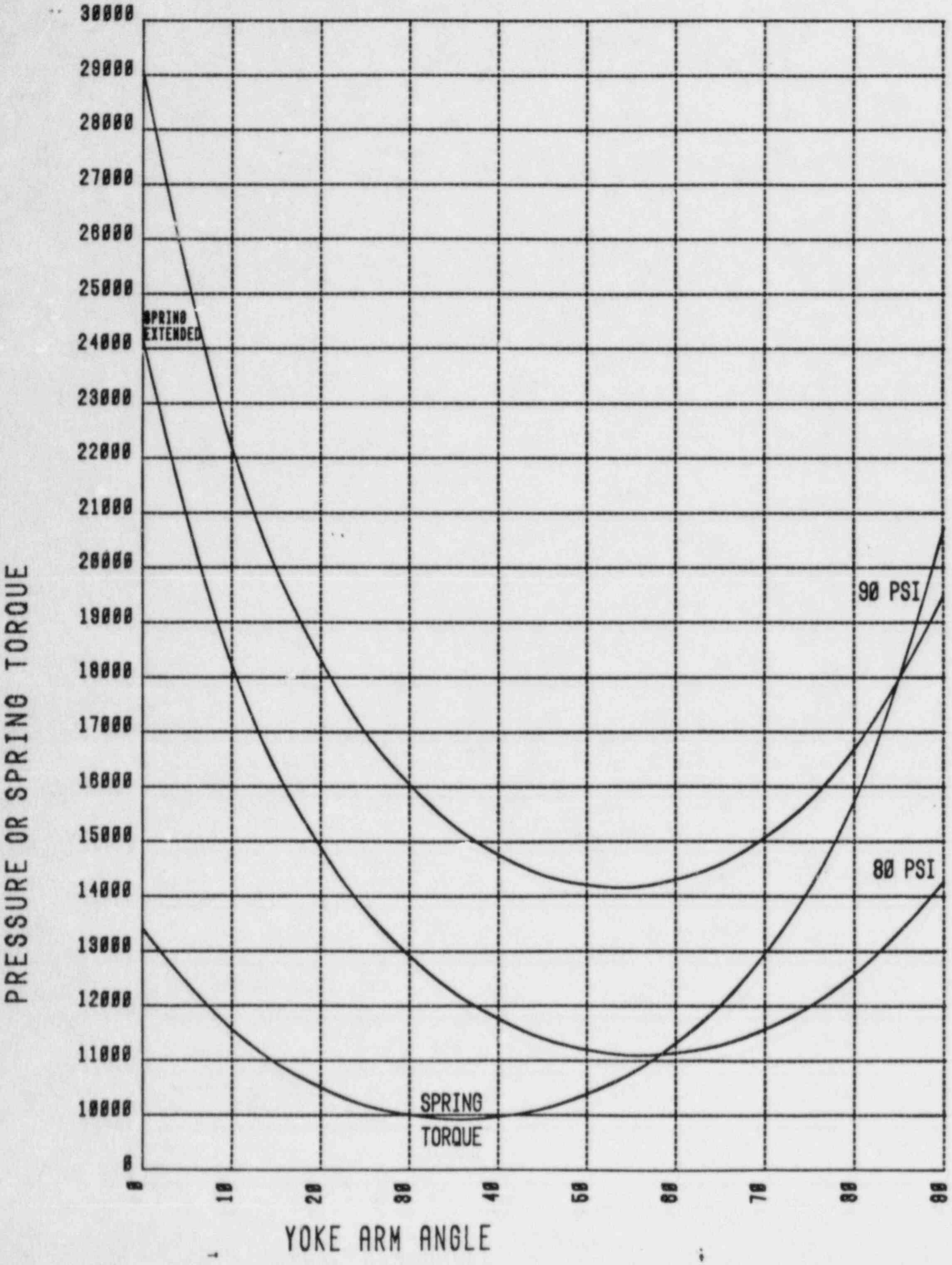
FIGURE 10A CALCULATED TORQUE DATA

T312-SR3

	SPRING	@ 80	@ 90
0	13,400	24,169	29,085
10	11,568	18,233	22,206
20	10,504	14,890	18,325
30	10,002	12,912	16,048
40	9,968	11,764	14,765
50	10,393	11,197	14,200
60	11,327	11,140	14,294
70	12,969	11,577	15,069
80	15,762	12,571	16,679
90	20,714	14,252	19,467

FIGURE 10B CALCULATED TORQUE PLOT

T312-SR3



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 80 - 110 psig air supply with a maximum flow rate of approximately 70 SCFM. There was no flow through the valve during this test.

TABLE 3

TAG NO. OF VALVE	VALVE SIZE (INCH)	BETTIS ACTUATOR MODEL NO.	OPENING TIME* SEC.	CLOSING TIME SEC.
1VQ026 / 2VQ026	26	NT820-SR3	25.60/23.75	4.71/3.99
1VQ027 / 2VQ027	26	"	33.30/29.29	3.91/4.19
1VQ029 / 2VQ029	26	"	40.68/24.93	3.95/4.23
1VQ030 / 2VQ030	26	"	34.50/27.20	3.65/3.93
1VQ031 / 2VQ031	26	"	25.53/24.17	4.34/4.24
1VQ034 / 2VQ034	26	"	34.09/23.57	4.48/4.59
1VQ036 / 2VQ036	26	"	27.82/24.25	3.99/4.12
1VQ040 / 2VQ040	26	"	30.20/23.58	4.10/4.31
1VQ042	8	NT312-SR3	3.0	1.0
1VQ043	8	"	3.0	1.0
2VQ042	8	"	3.0	1.0
2VQ043	8	"	3.0	1.0

* Opening times were restricted by Clow test set up (Air hose used had approximately 3/8" I.D.)

For comparison, a description of operating times for a valve Serial No. 80-8170-03-01 during a LOCA and Seismic Simulation Test is given in the Vought Corp. Report (Reference 7.0). The Vought Test demonstrated when there was flow through the valve, the aerodynamic torque aided closure thus reducing closing time.

3.0 VALVE OPERATING AND INSTALLATION REQUIREMENTS

3.1 Valve Operating Conditions

The normal and accident operating conditions for the subject valves are taken from Sargent & Lundy Eng. Specification T-3750, Data Tables No. DT-925 and DT-926 and Data Sheets AO-53 Rev. A and AO-54 Rev. A. Leakage requirements are per Spec. T-3750 Paragraph 305.3. This data is presented in summarized form in Tables 4 thru 6.

TABLE 4

Seismic Loadings For All Valves

Condition	Loading Condition	Acceleration Values (g) Horiz.	Vert.
Normal operation	gravity load only (no seismic acceleration)	0.0	1.0
Emergency	All loads per Data Tables DT-925 & DT-926	4.5	4.5

g = Acceleration as a fraction of the acceleration due to gravity.

TABLE 5

Pressure Differentials Applied to Valves

VALVE SIZE	VALVE TAG NO.		NORMAL OPERATING PRESSURE (PSIG)	OPER. TEMP. RANGE (°F)	DESIGN DIFFERENTIAL PRESSURE (PSiG)	NORMAL FLOW SCFM	FAILURE MODE
26"	1VQ026 1VQ027 1VQ029 1VQ030 1VQ031 1VQ034 1VQ036 1VQ040	2VQ026 2VQ027 2VQ029 2VQ030 2VQ031 2VQ034 2VQ036 2VQ040	-2 to 2	40-340	45 *	11,000	closed
8"	1VQ042 1VQ043 2VQ042 2VQ043		-2 to 2	40-340	45 *	1,100 #	closed

* Max for flow analysis is 45 PSIG, max for pressure retention is 60 PSIG.
# Estimated based on flow thru 26" valve since no rate is specified in DT-926

TABLE 6

Allowed Seat Leakage Rates

VALVE SIZE	VALVE TAG NO.		ALLOWED LEAKAGE
26"	1VQ026 1VQ027 1VQ029 1VQ030 1VQ031 1VQ034 1VQ036 1VQ040	2VQ026 2VQ027 2VQ029 2VQ030 2VQ031 2VQ034 2VQ036 2VQ040	.043 SCFM at 2 and 50 PSIG (1) for pneumatic test ---- .867 cc/min at 66 PSIG (2) for hydrostatic test
8"	1VQ042 1VQ043 2VQ042 2VQ043		.013 SCFM at 2 and 50 PSIG (1) for pneumatic test .267 cc/min at 66 PSIG (2) for hydrostatic test

(1) 50 PSIG is 110% of design differential for air flow
(2) 66 PSIG is 110% of design differential for pressure retention

3.2 Valve Installation Configurations

In addition to the pressure and flow conditions specified in 3.0, the valve performance is effected by the as installed orientation. Upstream and downstream, tees, elbows, reducers, and other valves can effect 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 Cygna prints are summarized in Figures 11 thru 21 with appropriate print references.

A summary of valve function, Cygna drawing number, flow direction, and a cross reference between the valve tag number, Clow serial number, and figure number is given in Tables 7 and 8 . It should be noted that the disc orientations are <u>not</u> shown in plan or elevation views in a true perspective. The orientation approximates the true prespective, while the views A-A or B-B give the true orientation. The plan or elevation views are shown only to give an idea as to the position of the valve shaft and the direction the valve opens. The opening direction can be clearly determined by comparing Figures 11 thru 21 with the valve detail drawings Figures 2 and 3.

TABLE 7

VALVE IDENTIFICATION AND INSTALLATION DRAWING CROSS REFERENCE
FOR
LASALLE COUNTY STATION UNIT #1 PURGE AND VENT VALVES

VALVE TAG NO.	CLOW SERIAL NUMBER	VALVE SIZE (IN)	NORMAL FUNCTION	CYGNA PIPING DRAWING * NUMBER/REV.	FLOW DIRECTION NORMAL	FLOW DIRECTION LOCA	FIGURE NO.
1VQ026	84-2842-01(N)-01	26	Intake	ECN-ME-001-LS-14/0	Toward 027	From penetration M-66	
1VQ027	84-2842-01(N)-02	26	Intake	ECN-ME-001-LS-14/0	Toward penetration M-66	From penetration M-66	
1VQ029	84-2842-01(N)-03	26	Intake	ECN-ME-001-LS-15/0	Toward 030	From penetration M-20	
1VQ030	84-2842-01(N)-04	26	Intake	ECN-ME-001-LS-15/0	Toward penetration M-20	From penetration M-20	
1VQ031	84-2842-01(N)-05	26	Exhaust	ECN-ME-001-LS-9/0	From penetration M-67	From penetration M-67	
1VQ034	84-2842-01(N)-06	26	Exhaust	ECN-ME-001-LS-12/0	From penetration M-21	From penetration M-21	
1VQ036	84-2842-01(N)-07	26	Exhaust	ECN-ME-001-LS-13/0	From South to North	From South to North	
1VQ040	84-2842-01(N)-08	26	Exhaust	ECN-ME-001-LS-11/0	From penetration M-67	From penetration M-67	
1VQ042	84-2842-02(N)-01	8	Intake	ECN-ME-001-LS-14/0	From valve 042 toward valve 043	From penetration M-66	
1VQ043	84-2842-02(N)-02	8	Intake	ECN-ME-001-LS-14/0	From valve 042 toward valve 043	From penetration M-66	

* Cygna Job No. 85007 (other drawings referenced include ECN-ME-001-LS-10 Rev. 0 and ECN-ME-001-LS-18 Rev. 0).

TABLE 8

VALVE IDENTIFICATION AND INSTALLATION DRAWING CROSS REFERENCE FOR LASALLE COUNTY STATION UNIT #2 PURGE AND VENT VALVES

VALVE TAG NO.	CLOW SERIAL NUMBER	VALVE SIZE (IN)	NORMAL FUNCTION	CYGNA PIPING DRAWING * NUMBER/REV.	FLOW DIRECTION NORMAL	FLOW DIRECTION LOCA	FIGURE NO.
2VQ026	84-2842-01(N)-09	26	Intake	ECN-ME-002-LS-13/0	Toward 027	From penetration M-66	
2VQ027	84-2842-01(N)-10	26	Intake	ECN-ME-002-LS-13/0	Toward penetration M-66	From penetration M-66	
2VQ029	84-2842-01(N)-11	26	Intake	ECN-ME-002-LS-14/0	Toward 030	From penetration M-20	
2VQ030	84-2842-01(N)-12	26	Intake	ECN-ME-002-LS-14/0	Toward penetration M-20	From penetration M-20	
2VQ031	84-2842-01(N)-13	26	Exhaust	ECN-ME-002-LS-8/0	From penetration M-67	From penetration M-67	
2VQ034	84-2842-01(N)-14	26	Exhaust	ECN-ME-002-LS-11/0	From penetration M-21	From penetration M-21	
2VQ036	84-2842-01(N)-15	26	Exhaust	ECN-ME-002-LS-12/0	From South to North	From South to North	
2VQ040	84-2842-01(N)-16	26	Exhaust	ECN-ME-002-LS-10/0	From penetration M-67	From penetration M-67	
2VQ042	84-2842-01(N)-03	8	Intake	ECN-ME-002-LS-1[illegible]/0	From valve 042 toward valve 043	From penetration M-66	
2VQ043	84-2842-01(N)-04	8	Intake	ECN-ME-002-LS-13/0	From valve 042 toward valve 043	From penetration M-66	

* Cygna Job No. 85007 (other drawings referenced include ECN-ME-002-LS-9 Rev. 0, ECN-ME-002-LS-16 Rev. 0, and Sargent & Lundy Drawing No. M-138 Sheet 1 Rev. AC.)

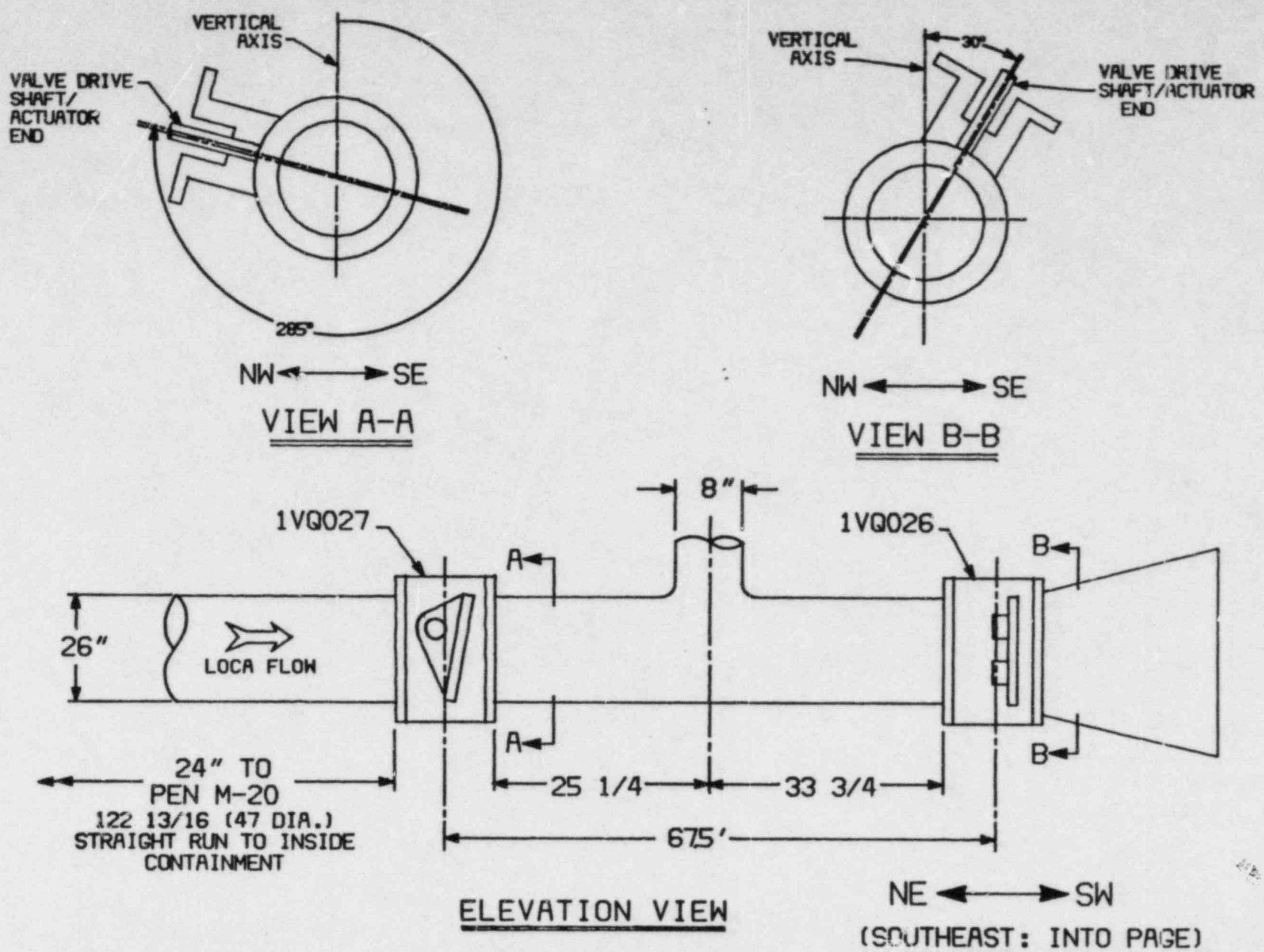


FIGURE 11

VALVES 1VQ027 & 1VQ026 AS INSTALLED PIPING CONFIGURATION.
ELEVATION VIEW LOOKING SOUTHEAST PER DRAWING ECN-ME-001-LS-14.

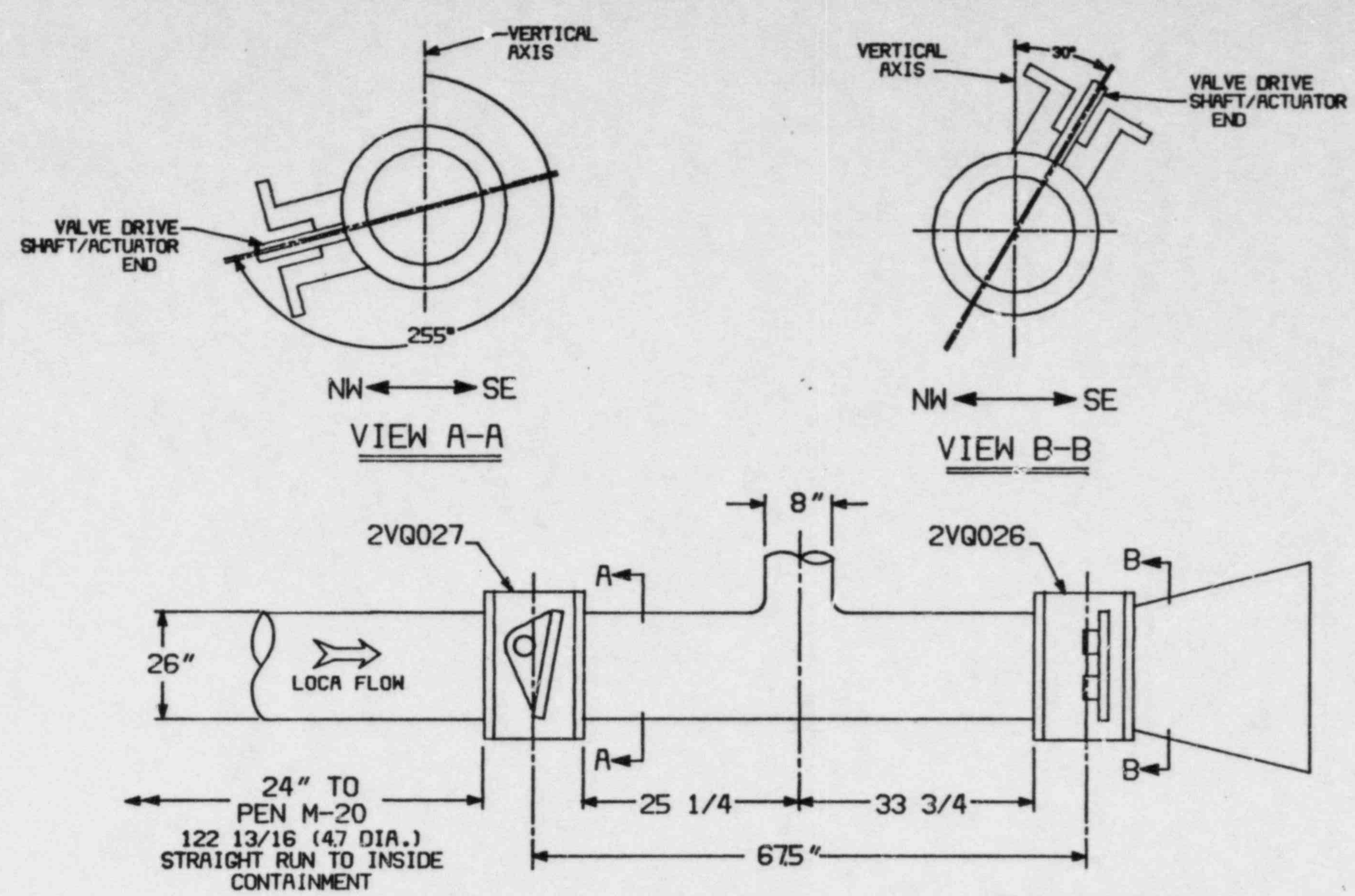


FIGURE 12

VALVES 2VQ027 & 2VQ026 AS INSTALLED PIPING CONFIGURATION.
ELEVATION VIEW LOOKING SOUTH EAST PER DRAWING ECN-ME-002-LS-13.

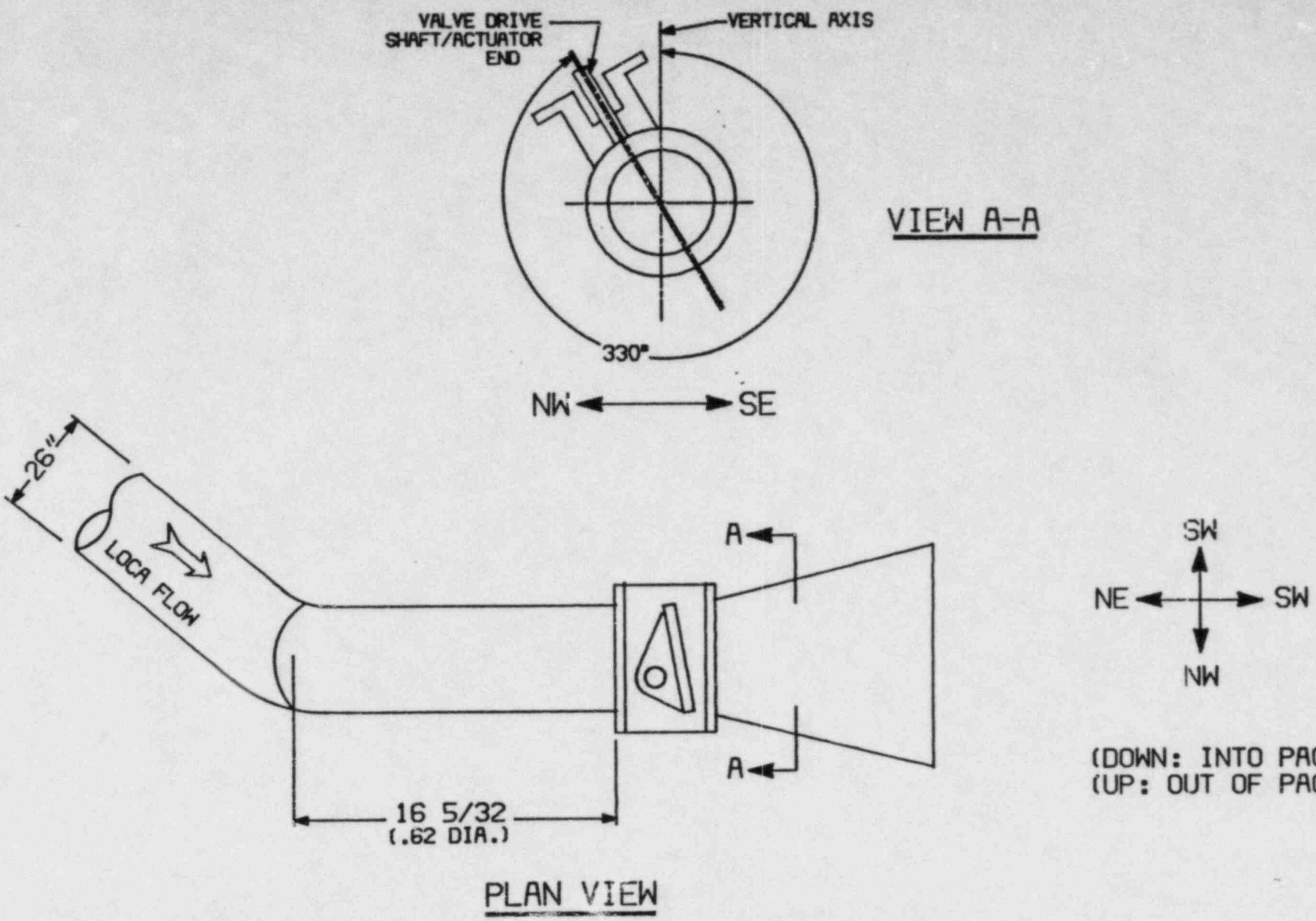


FIGURE 13

VALVES 1VQ029 & 2VQ029 AS INSTALLED PIPING CONFIGURATION. PLAN VIEW LOOKING DOWN PER DRAWINGS ECN-ME-001-LS-15 AND ECN-ME-002-LS-14.

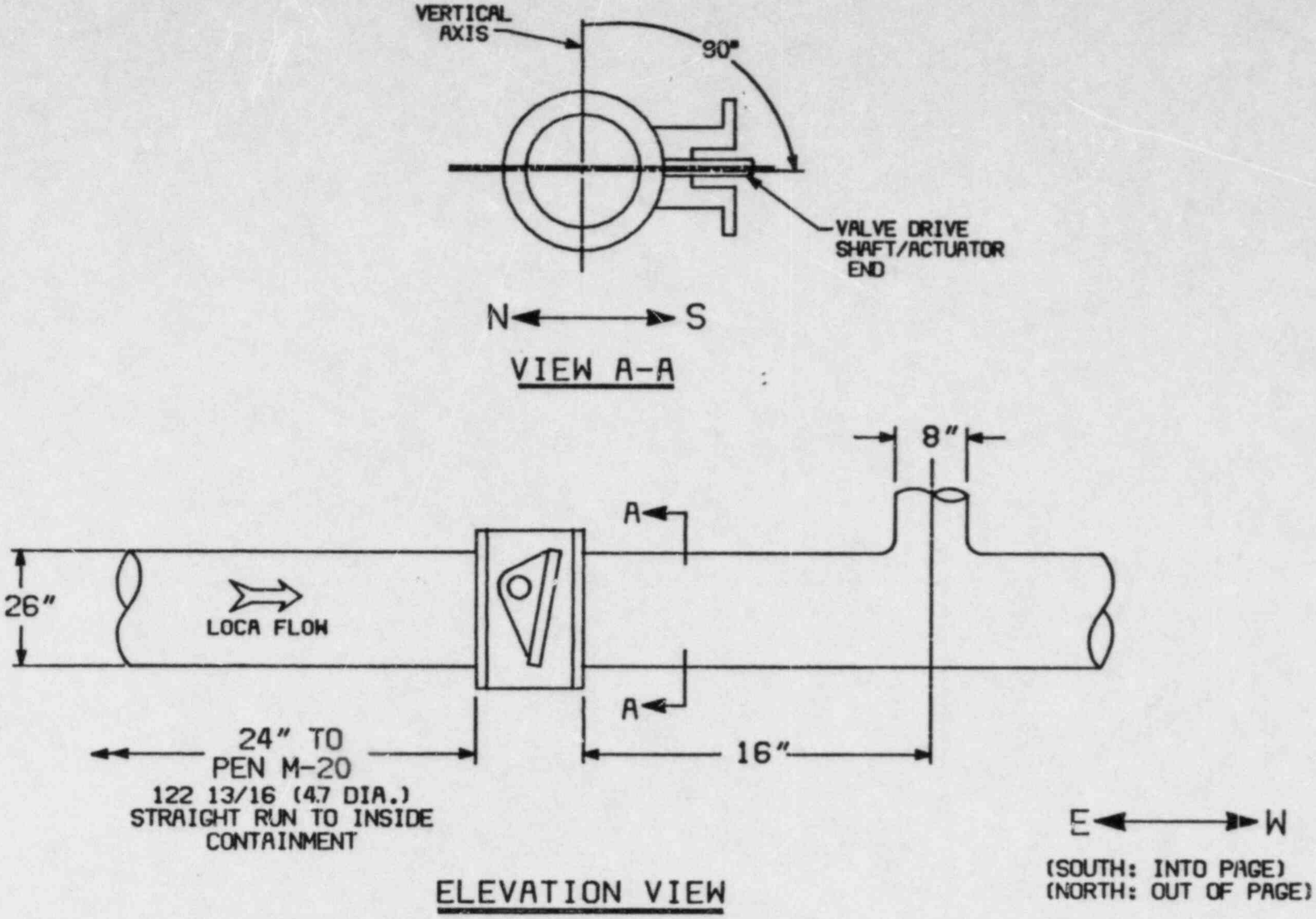


FIGURE 14

VALVES 1VQ030 & 2VQ030 AS INSTALLED PIPING CONFIGURATION.
ELEVATION VIEW LOOKING SOUTH PER DRAWING ECN-ME-001-LS-15
AND ECN-ME-002-LS-14.

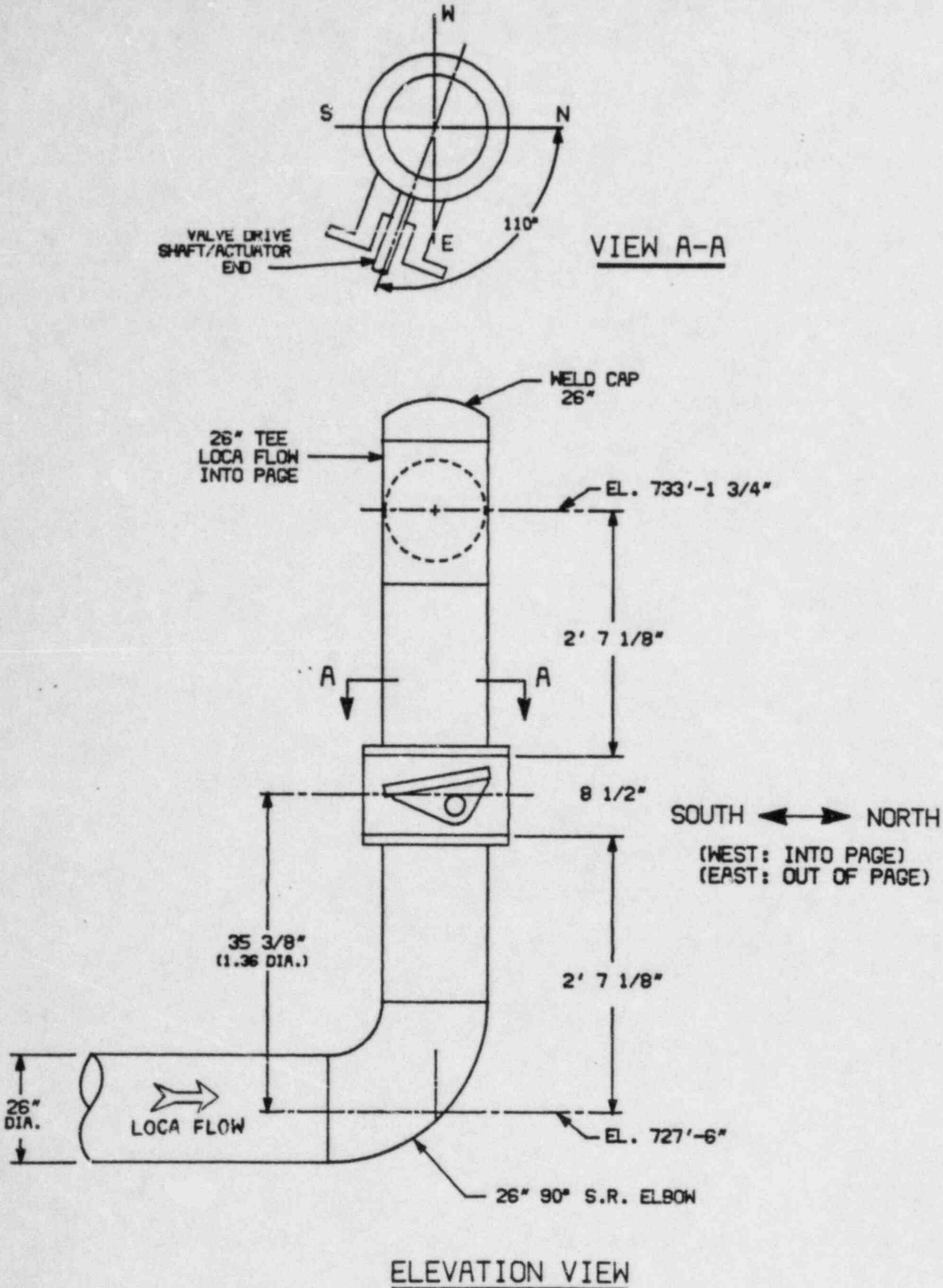


FIGURE 15

VALVE 1VQ031 & 2VQ031 AS INSTALLED PIPING CONFIGURATION. ELEVATION VIEW LOOKING WEST PER DRAWING ECN-ME-001-LS-9 AND ECN-ME-002-LS-8.

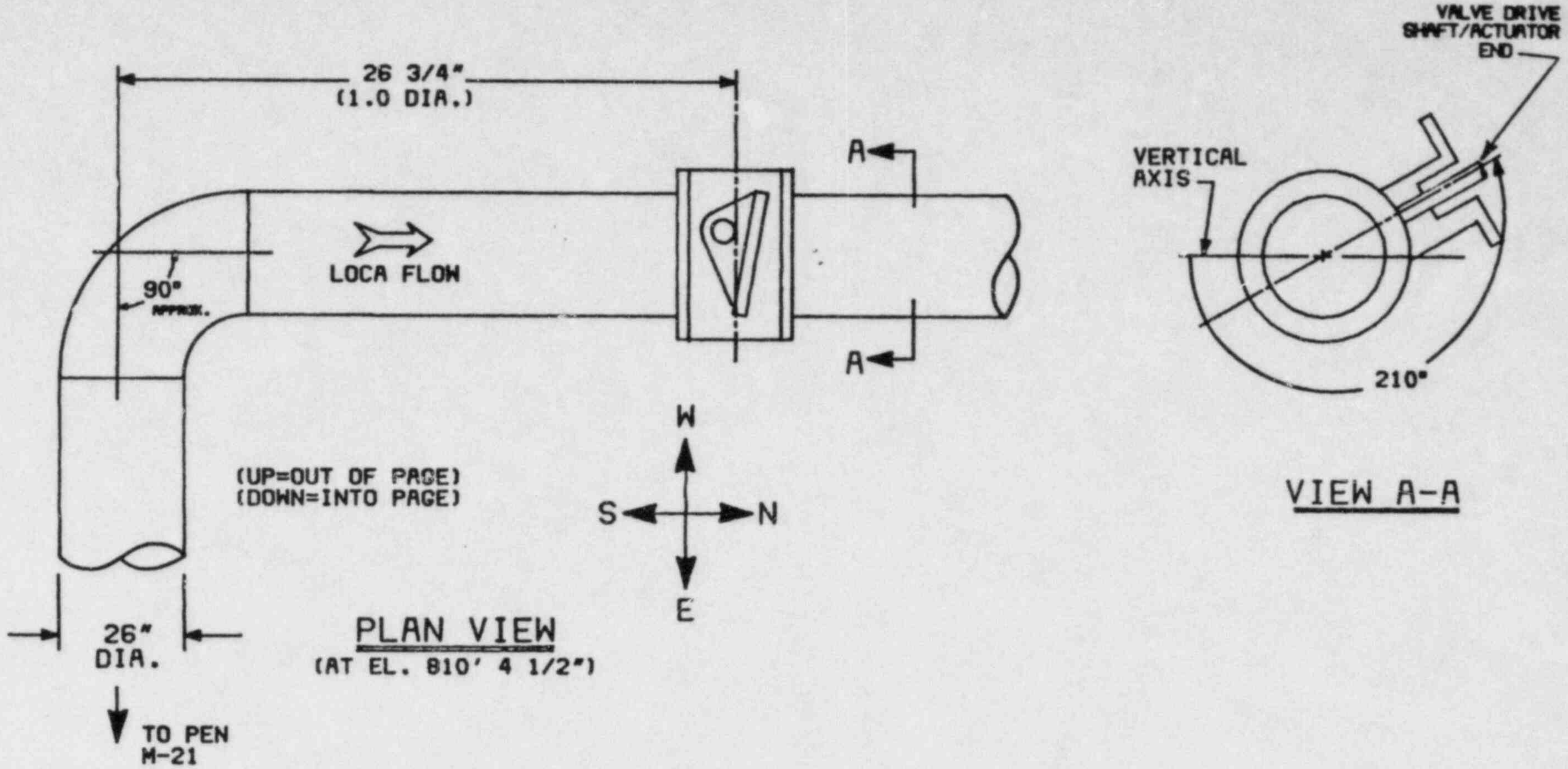


FIGURE 16

VALVE 1VQ034 AS INSTALLED PIPING CONFIGURATION.
PLAN VIEW LOOKING DOWN PER DRAWING ECN-ME-001-LS-12.

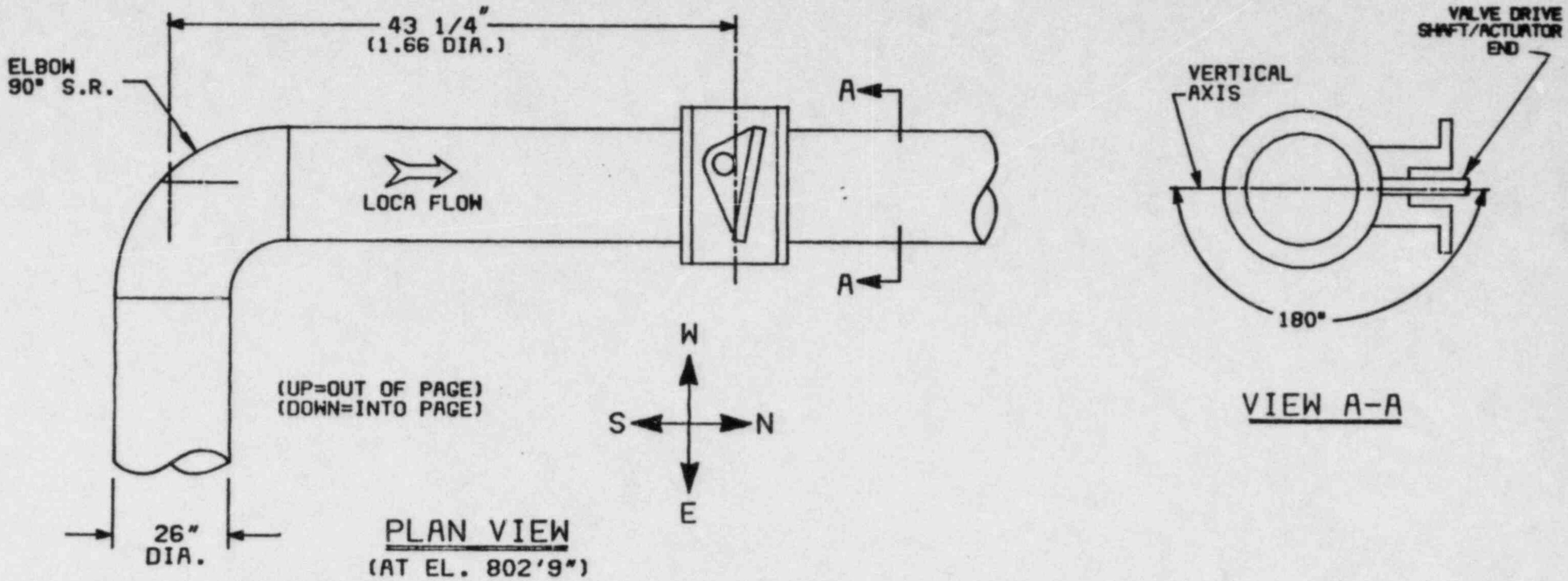


FIGURE 17

VALVE 2VQ034 AS INSTALLED PIPING CONFIGURATION.
PLAN VIEW LOOKING DOWN PER DRAWING ECN-ME-002-LS-11.

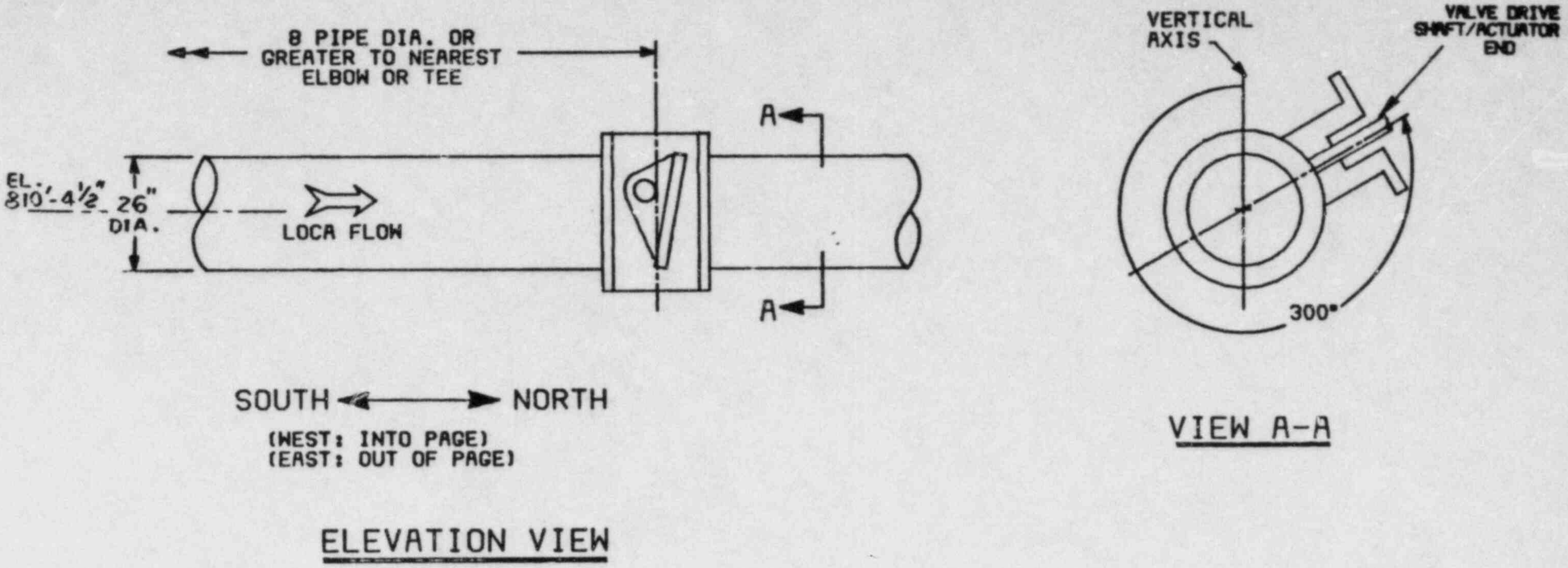


FIGURE 18

VALVE 1VQ036 & 2VQ036 AS INSTALLED PIPING CONFIGURATION. ELEVATION VIEW LOOKING WEST PER DRAWING ECN-ME-001-LS-13 AND ECN-ME-002-LS-12.

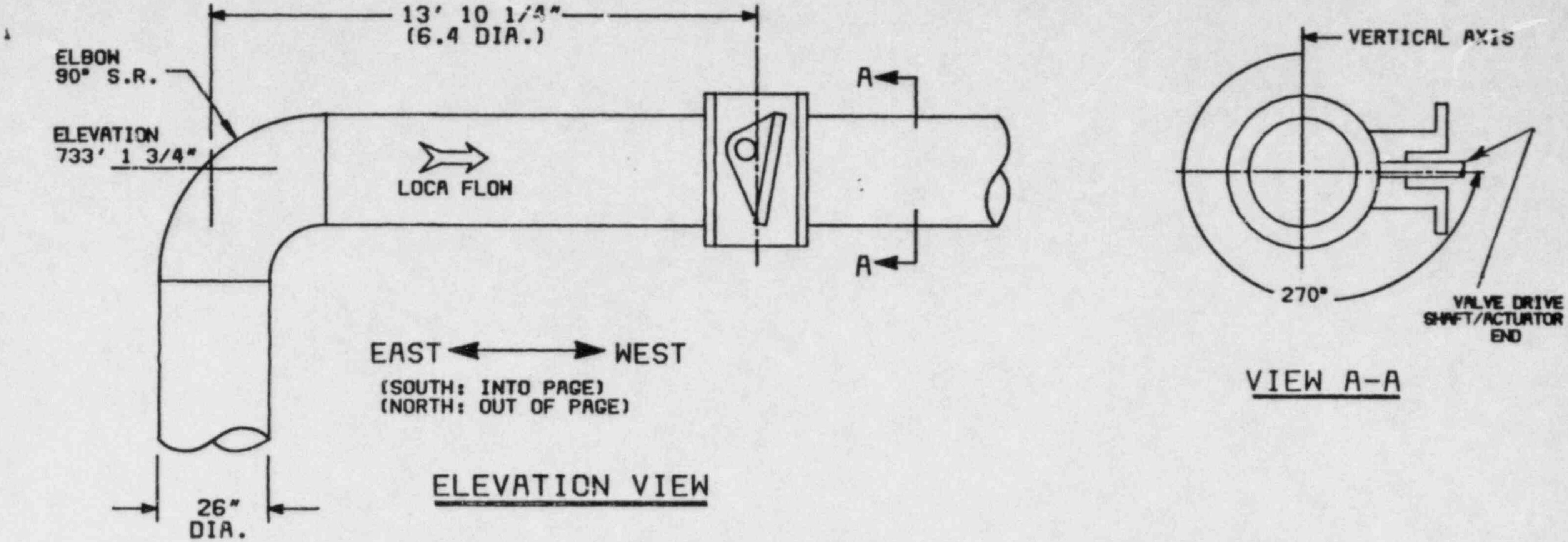


FIGURE 19

VALVE 1VQ040 & 2VQ040 AS INSTALLED PIPING CONFIGURATION.
ELEVATION VIEW LOOKING SOUTH PER DRAWING ECN-ME-001-LS-11
AND ECN-ME-002-LS-10.

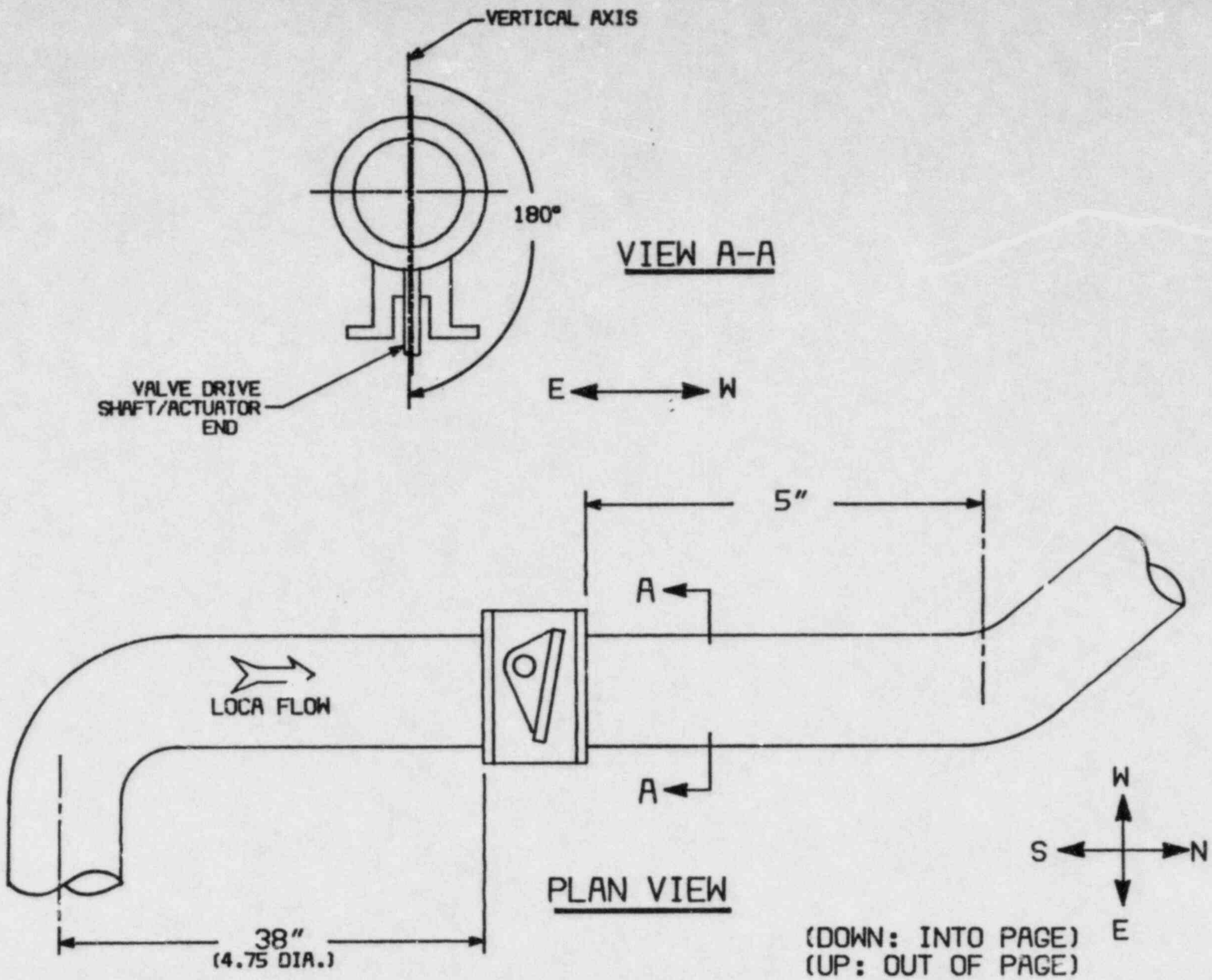


FIGURE 20

VALVES 1VQ042 & 2VQ042 AS INSTALLED PIPING CONFIGURATION. PLAN VIEW LOOKING DOWN PER DRAWINGS ECN-001-LS-14 AND ECN-002-LS-13.

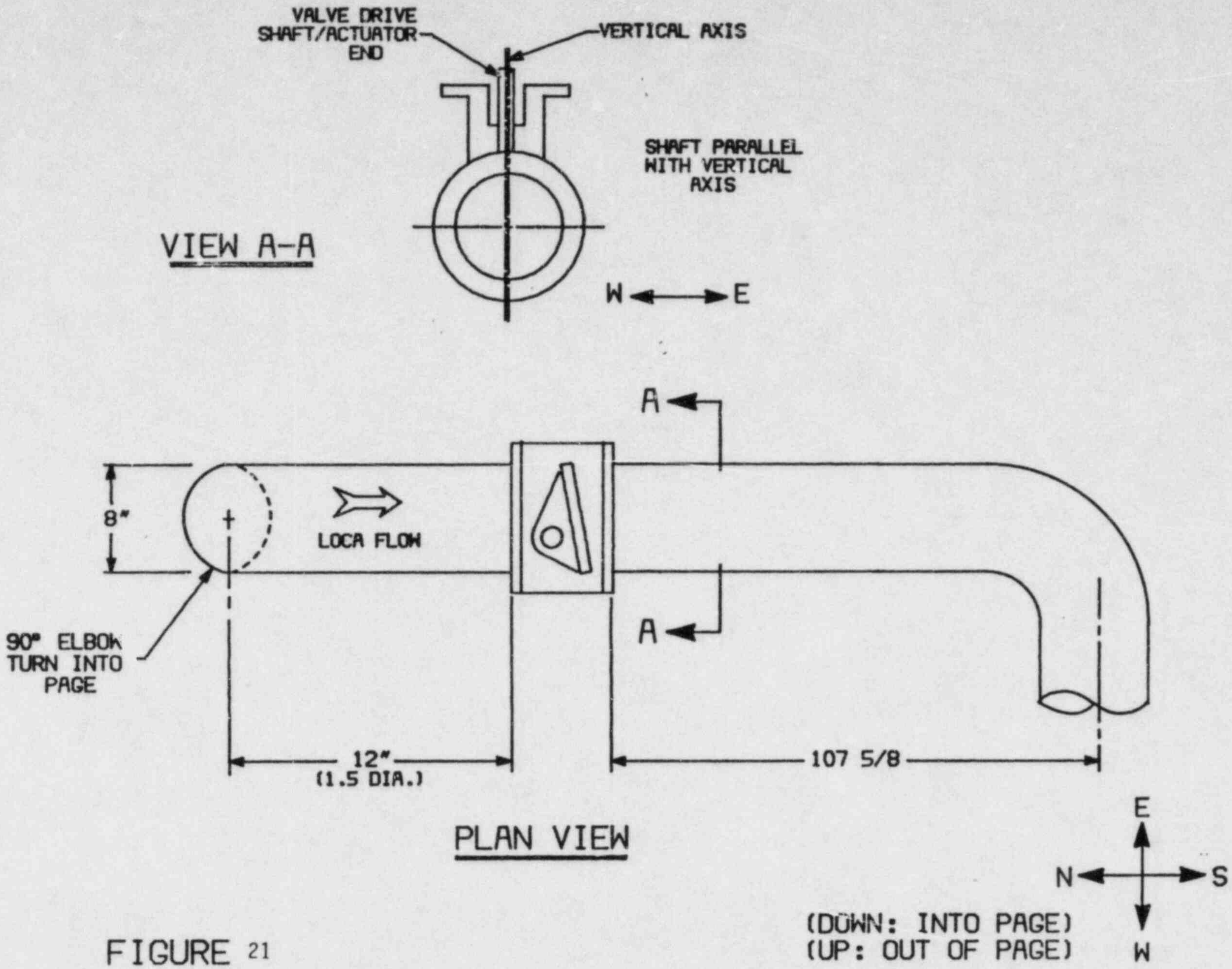


FIGURE 21

VALVES 1VQ043 & 2VQ043 AS INSTALLED PIPING CONFIGURATION. PLAN VIEW LOOKING DOWN PER DRAWINGS ECN-ME-001-LS-14 AND ECN-002-LS-13.

4.0 VALVE STRUCTURAL INTEGRITY UNDER SEISMIC AND OPERATIONAL LOADINGS

Operability of the subject valves has been verified by a combination of testing and analysis in accord with Patel Engineers Technical Proposal PEI-TR-85-22. Separate reports have been prepared and provided demonstrating suitability of valve components and the assembly for both seismic and environmental requirements. A listing is provided in the references (7.0) at the end of the report. This section gives a brief summary of the results of such tests and analyses in meeting the conditions as presented in Section 3.0. For specific information the referenced reports must be read.

4.1 Valve Frequency And Stress Analysis

Valve frequency and stress analysis was performed by Patel Engineers, Huntsville, Alabama for each valve size. 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 of use to justify its applicability and validity. The analyses were made for the seismic conditions in excess of those stated in Section 3.0 and for pressure and temperature in excess of those specified in Table 5 . This was done to provide a very conservative generic approach to showing qualification. For stress analysis, allowable stresses were compared to ASME Section III requirements. Tables 11 and 12 summarize the maximum stresses in the critical valve elements and how these relate to allowed values.

TABLE 9

Lowest Valve Resonant Frequencies
(Per Analysis)

VALVE SIZE	TYPE ACTUATOR	FREQUENCIES (Hz)
26"	Air operated	≥ 50
8"	Air operated	≥ 50

NOTE: Analysis was generic and assumed a greater actuator mass than is actually used thus true resonant frequency should be higher than indicated.

TABLE 10

Condition Applied For Generic Stress Analysis *

VALVE SIZE	VALVE TAG NOS.		BODY DESIGN PRESSURE (PSIG)	DISC DIFFERENTIAL PRESSURE (PSI)	DESIGN TEMP. °F	DESIGN SEATING TORQUE (in-lb)
26"	1VQ026	2VQ026	285	155	340°F	164,400
	1VQ027	2VQ027				
	1VQ029	2VQ029				
	1VQ030	2VQ030				
	1VQ031	2VQ031				
	1VQ034	2VQ034				
	1VQ036	2VQ036				
	1VQ040	2VQ040				
8"	1VQ042		285	75	350°F	16,643
	1VQ043					
	2VQ042					
	2VQ043					

* Used for Report PEI-TR-852200-1 (26") & PEI-TR-83-24 (8")

Table 11

Summary of Allowable Stresses
26" Valve (Loads Per Generic Report)

LOCATION	MATERIAL	ALLOWABLE STRESS (psi) (PER ASME SECTION III, TABLES I-7.1 THROUGH I-7.3) (for 7.0g Seismic Load)	STRESS VALUE (psi)	ELEMENT	STRESS RATIO
Valve Body	SA 516 GR.70	17500	6703	72	0.38
Disc	SA 516 GR.70	17500	3540	314	0.20
Drive Shaft	SA 564 Type 630 H-1100	34500	3044	331	0.87
Operator Adapter Plate	SA 516 GR.70	31500** 34200***	29120	361	0.93** 0.85***
Adapter Plate Bolts (7g)	SA 193 GR.B7	25000	$36986\sigma_n$ 20736τ	N/A	0.93
Cover Plate	SA 516 GR.70	17500	5807	N/A	0.33
Cover Plate Bolts	SA 193 GR.B7	25000	$12276\sigma_n$ 172τ	N/A	0.20*

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

** Per ASME, Section III, Subsection NC, Article NC3520

*** Evaluated Against $.9\sigma_y$

Table 12

Summary of Allowable Stresses
8" Valve (Loads Per Generic Report)

LOCATION	MATERIAL	ALLOWABLE STRESS (psi) (PER ASME SECTION III, TABLES I-7.1 THROUGH I-7.3)	STRESS VALUE (psi)	ELEMENT	STRESS RATIO
Valve Body	SA 516 GR.70	17500	7088	221	0.41
Disc	SA 516 GR.70	17500	6767	286	0.39
Drive Shaft	SA 564 Type 630 H-1100	34550	27610	301	0.80
Operator Adapter Plate	SA 516 GR.70	1 (ASME "S") = 17500	2718 σ_m	344	0.16
		1.5 (ASME "S") = 26250	25313 σ_{m+b}	344	0.96
Adapter Plate Bolts (7g)	SA 193 GR.B7	25000	55374 σ_N 10602 τ^N	N/A	0.95*
Adapter Plate Bolts (11g)	SA 193 GR.B7	94500	83937 σ_N 15333 τ^N	N/A	0.92**
Cover Plate	SA 516 GR.70	17500	6234	N/A	0.35
Cover Plate Bolts	SA 193 GR.B7	25000	4195 σ_N 30 τ^N	N/A	0.07*

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

**Principal Stress Evaluated Against 90% Fy.

4.2 Bettis Actuator Resonant Frequency Test

A Low-Level Seismic Vibration Test was performed on a NT312-SR5 and NT820-SR4 actuator to determine resonant frequencies. The test was performed at NTS*, Saugus, Ca. The test program consisted of uniaxial sine sweep testing in each of the three orthogonal axis. The actuator was instrumented with accelerometers to measure input and response accelerations. The test identified the units structural resonances within the frequency range of 1 to 100 Hz. This information is supplied by a report under separate cover (see References 7.0 Report PEI-TR-83-29 Rev. A).

4.3 Asco Solenoid Valve Resonant Frequency Test

A valve actuator solenoid valve, Asco model 831664, was subjected to both a sine sweep test and sine beat test in each of three orthogonal test orientations for a previous Clow contract. In addition, the specimen was tested for leakage prior to and after each test segment (a segment being a test is one of the three orientations). Also, during the test, pressure was applied and measured, and functional operability was monitored.

The test demonstrated no major resonances between one and 130 Hz. One orientation showed a system resonance between 130 and 140 Hz which was outside of the required operability range. The sine beat test which consisted of 6260 beats per orientation at 5 to 100 Hz and accelerations of 2.0 to 11.0 g (within test table acceleration limits) showed the solenoid valve to be operable before, during, and after the test. No detectable leakage occurred during any phase of the tests.

* National Technical System

4.3 Asco Solenoid Valve Resonant Frequency Test (Con't)

Although these test were not performed for the subject contract (no report is submitted for review or approval), the report is available for review at Clow's Westmont, Illinois facility.

4.4 Static Load Test During Simulated LOCA Flow

As part of the operability test performed at Vought (see Reference 7.0) an 11.0 g load was applied in each of two orthagonal directions through the approximate center of gravity of the actuator. With the load applied and flow through the valve greater than expected in service, the valve operated within the required time period. This aspect of the test demonstrated that the actuator to valve connection was sufficiently rigid to remain fully operable under this load. Further details are included in the subject report.

4.5 Bettis Actuator Seismic/Hydrodynamic Operability Test

A Dynamic Test was performed on a NT312-SR5 and NT820-SR4 actuator to demonstrate operability under anticipated loadings which may be encountered in service. The test was performed at NTS, Saugus, Ca., in accord with NTS Test Procedure 528-0951. The test demonstrated the units would operate as required before, during, and after the test. This information is supplied by a report under separate cover (see Reference 7.0 Report No. PEI-TR-83-29 Rev. A).

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 13) 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(1) 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 = .528) 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

(1) Instrument Standards Association

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 for produced 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 13 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 14 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 9% deviation) correlation was obtained for torque critical items for the 8" valves. For the 26" valves correlation is good (less than 10% deviation) for all critical dimensions other than disc thickness and offset E, From the model test data, greater disc thickness would reduce the potential for torques tending to resist valve closure. The offset E used for calculation is 30% larger than the production size which would cause the calculated torques to be higher than actual in service torques, thus calculation made by this method would be considered highly conservative.

TABLE 13

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 ℄ to Seal ℄, 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 +	1.25	.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 *	3.38	.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 14

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

ELEMENTS	8" SIZE	8" RATIO	26" SIZE	26" RATIO
*I.D.	7.981	1.05	25.00	.98
*A_2	7.244	1.09	24.15	.99
*K_2	7.069	1.07	23.57	.96
Shaft Dia.	1.50	1.05	3.25	1.07
Shaft C_L to Seal C_L, L	1.50	.93	2.63	1.10
*Disc Thickness	1.25	.95	1.75	1.16
*Shaft Offset E	1.375	1.05	.66	1.30
Shaft Offset LC	1.410	NA	.725	NA
Ear Width	2.00	NA	4.00	NA
Ear Height	2.25	NA	4.00	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 axiz of elliptical seal

E = Offset between shaft axis and disc center (See Figure 4)

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 a 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 (see photo from water table study Figure 23).[1] Further, the mitered elbow produces flow patterns more severe than expected for tee flow (see Figures26a and26b). The testing performed has given added evidence in support of this assumption. (See report reference 7.0 C-3) 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 22) and aid closure for geometry 2.

Based on these considerations, models of a 12", 24", and 48" valve (per Table 13) 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

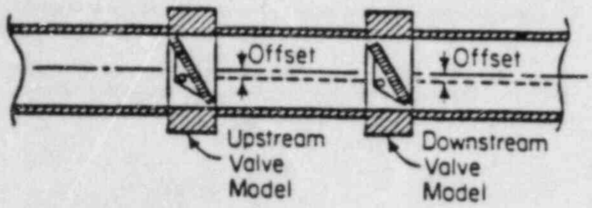
[1] See reference 7.0 E-3 .

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.

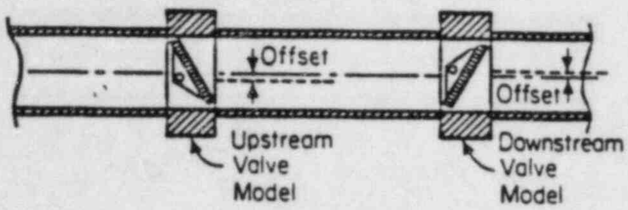
For the subject job some valves are installed closer than 2 diameters from an elbow. Since the mitered elbow used in the model tests is a worst case condition and radius type elbows are typically used for in plant installation, use of the test data for 2 diameters downstream for determining installed operability is considered reasonable. If torque operating margins are adequate, this judgement is further justified.

5.1.3 Tests With Two Valves In Series

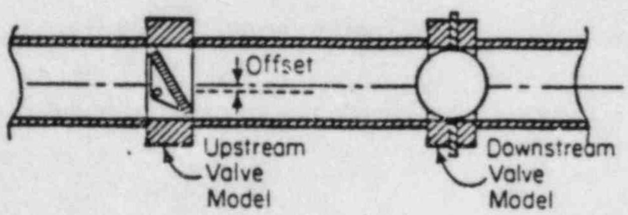
When two valves are installed in series in a pipe run at a relatively close distance (less than 8 diameters) some level of interaction will occur. Several different orientations of the two valves relative to one another are possible as shown in Figure 22. Model tests performed to determine aerodynamic torque characteristics, indicate that orientation 2 with the upstream valve failed (stuck) at 60° open and the downstream valve at full open would represent a worst case condition (highest torques resisting downstream valve closure). These model tests are more fully described in a separate report indicated in the references (Section 7.0).



Orientation 1



Orientation 2



Orientation 3

FIGURE 22 POSSIBLE ORIENTATION OF TWO CLOW VALVES INSTALLED IN SERIES

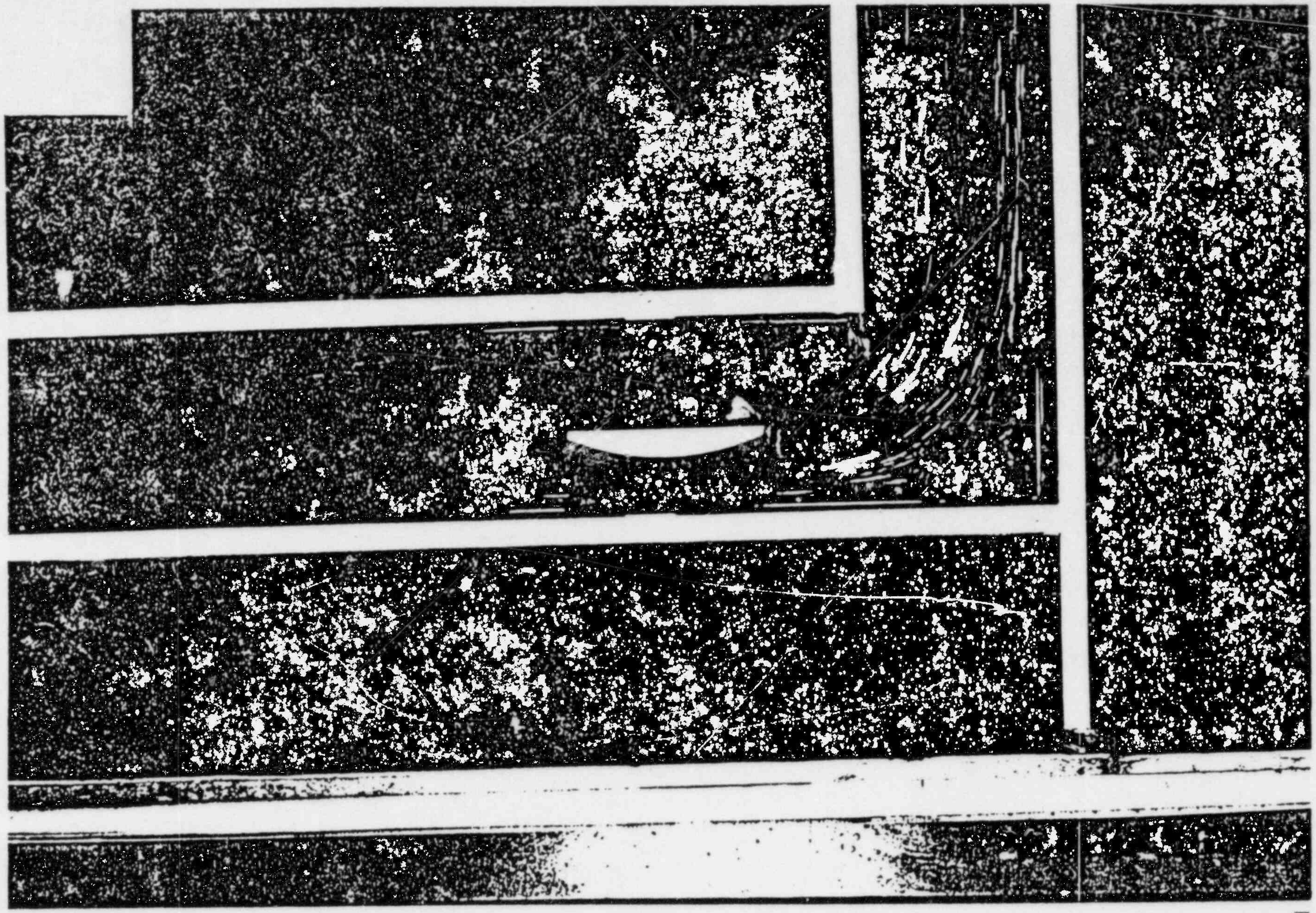


FIGURE 23 - Water Table Study of Choked Flow Pattern With Disc Full Open (90°)

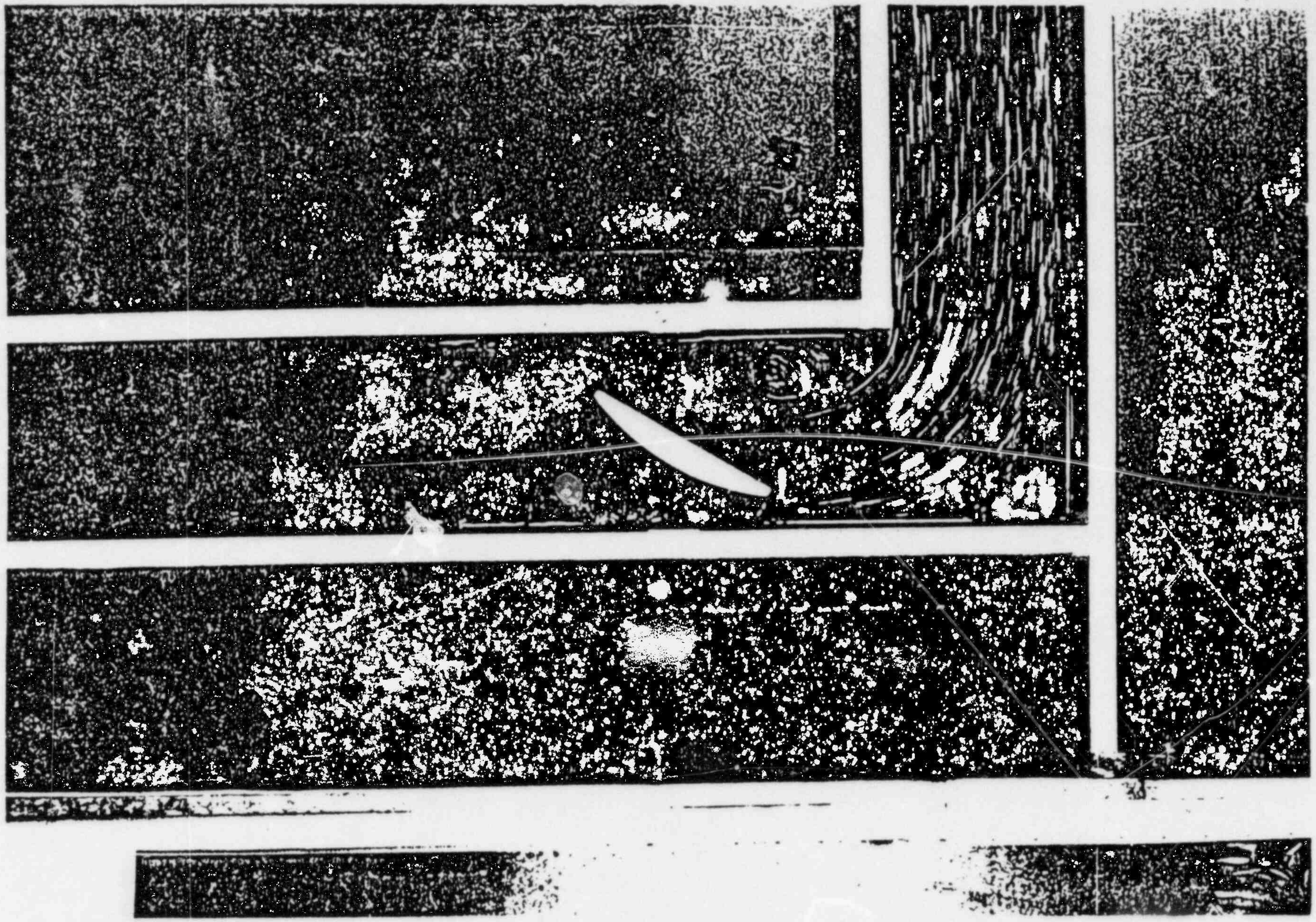


FIGURE 24 - Water Table Study of Choked Flow Pattern With Disc Partially Open (60^{o})

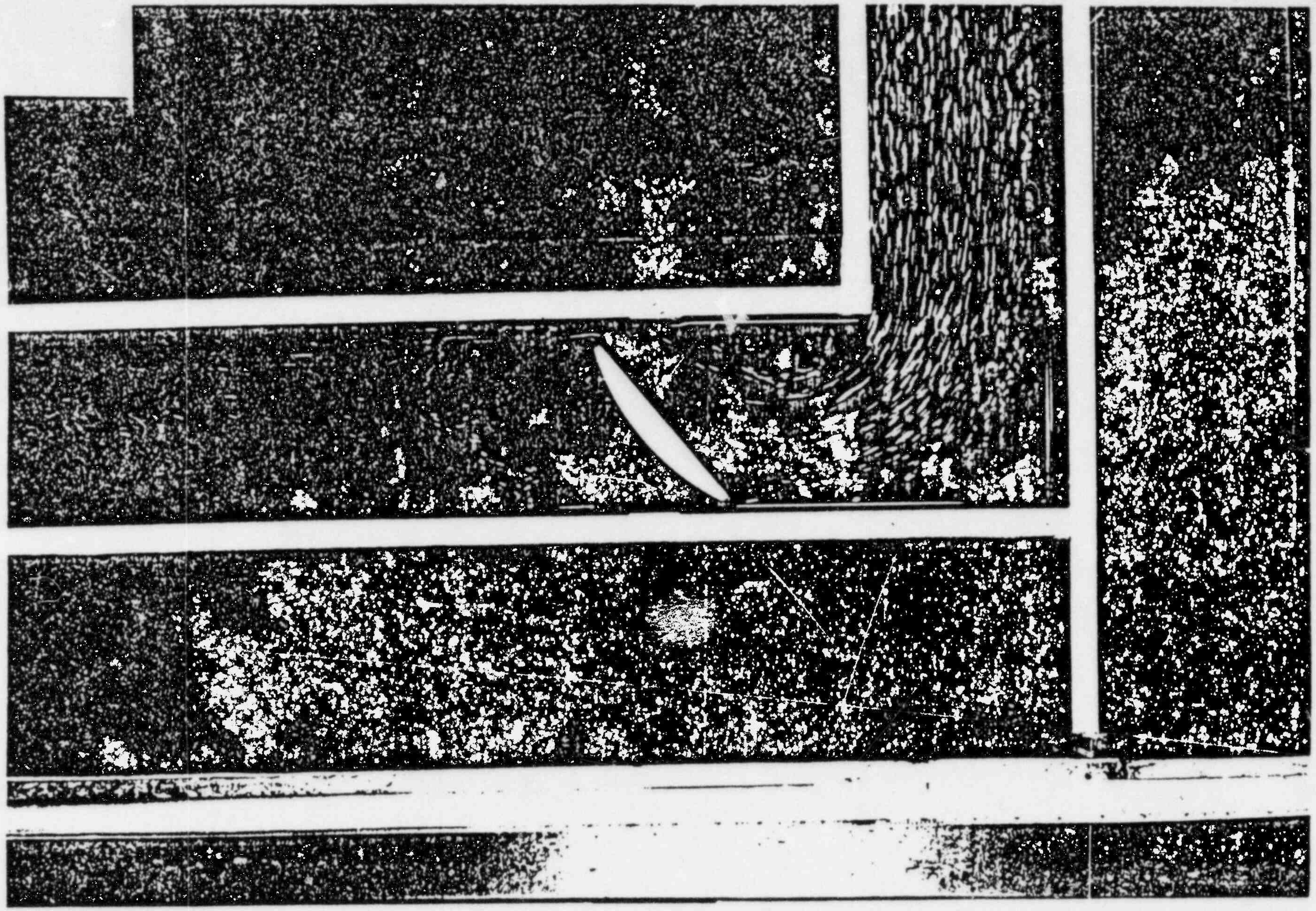


FIGURE 25 - Water Table Study of Choked Flow Pattern With Disc Partially Open (40^{o})

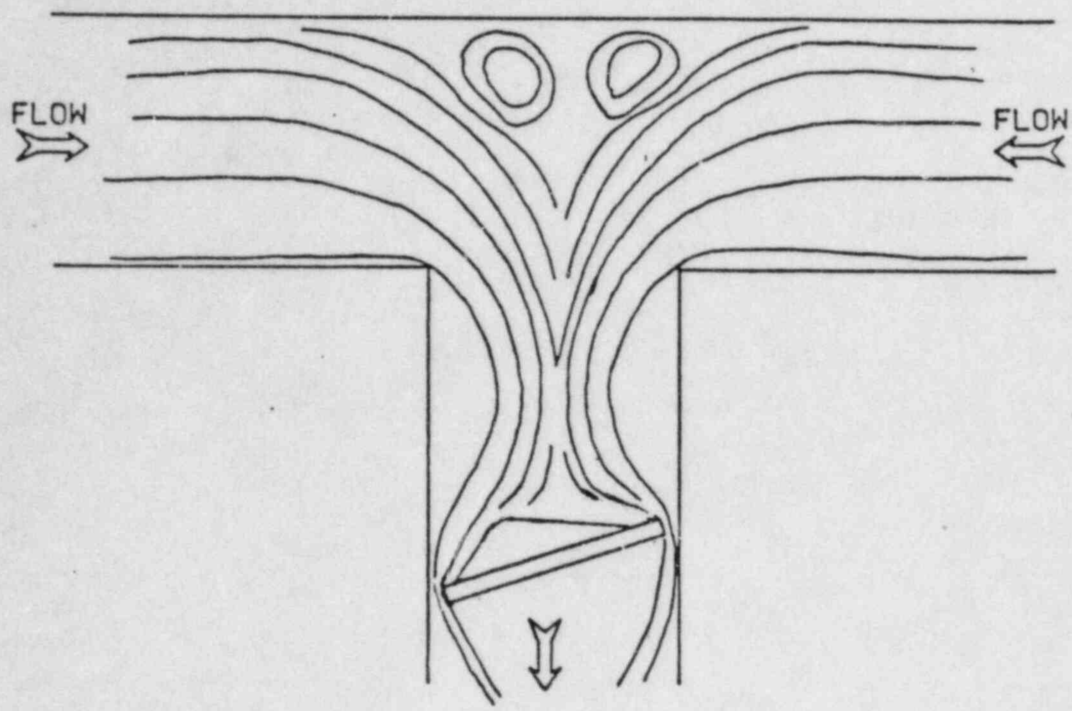


FIGURE 26a
TEE WITH FLOW FROM TWO SIDES

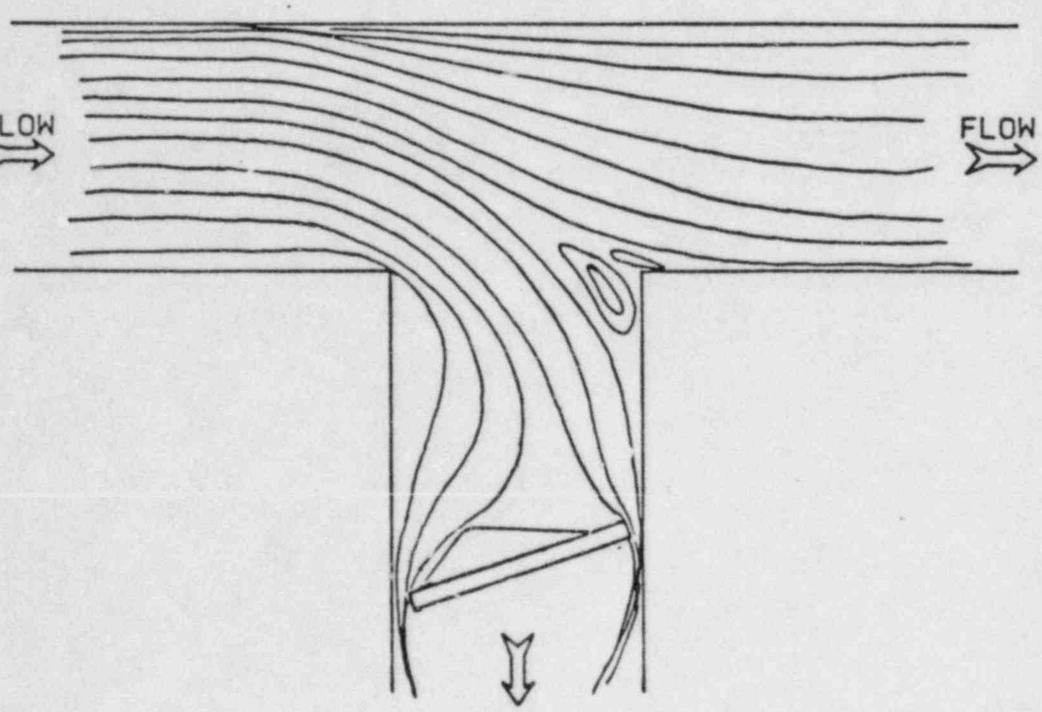
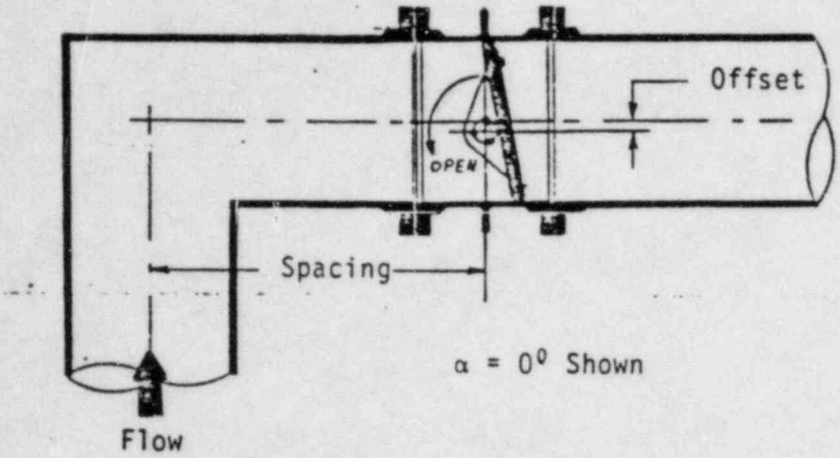
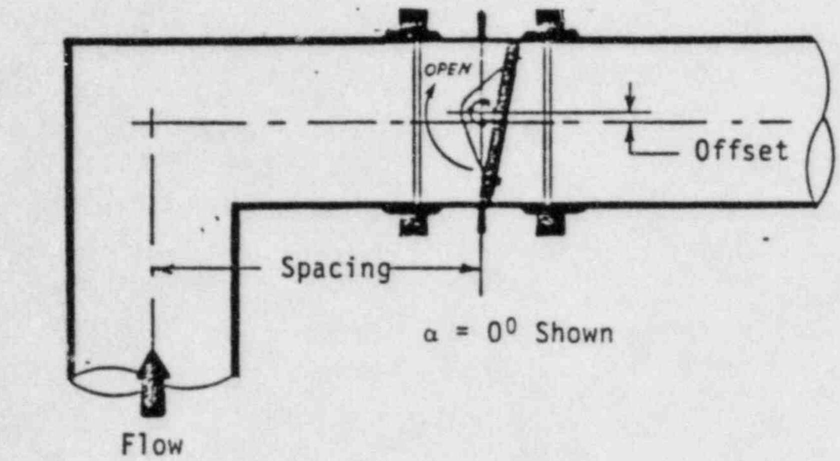


FIGURE 26b
TEE WITH FLOW FROM ONE SIDE

Geometry 1



Geometry 2



Geometry 3

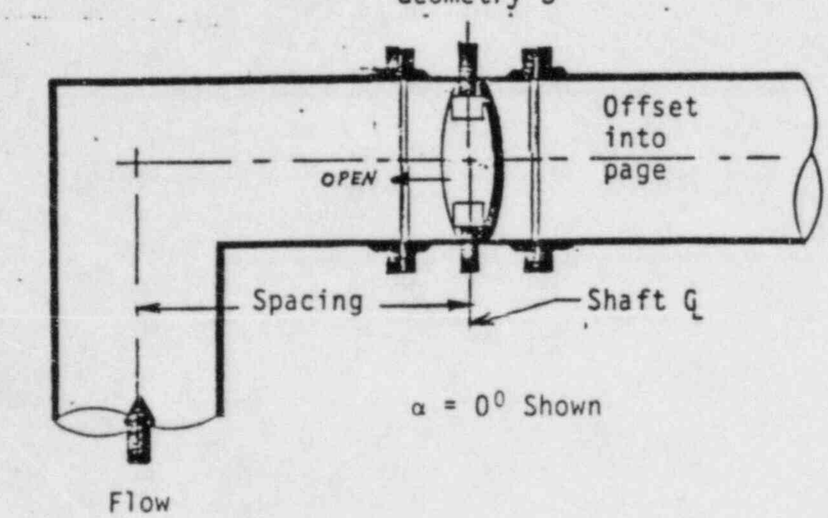


FIGURE 27 - Valve Orientations Relative to Upstream Elbow

5.1.4 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 (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-7) 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 were designed to close within 5 seconds for flow conditions produced by maximum differential pressure (see 3.0, Table 5). These are the maximum conditions expected in the event of a LOCA. The valves were designed to fully open within 5 seconds for conditions of normal flow, though most are capable of opening fully within this time for maximum pressure differential. All air actuated valves will fail closed through use of a return spring in the actuator. They will open within 5 seconds if the air supply to the actuator is adequate.

In the Vought Test, which used a pneumatic/spring return actuated valve, (Reference 7.0B-7) closing times were shown to improve slightly with flow through the valve. Opening times were retarded on the order of 1/2 to 2 seconds depending on flow conditions. These changes are of a conservative nature since it was necessary to restrict both the valve opening and closing air supplies to prevent pressure upstream of the valve from increasing to an unreasonable level during the test. The conduct of the test would suggest that opening times in actual service for similar valve/actuator assemblies might be retarded about .3 to .5 (since normal flows are much lower than tested flows) and closing times might be improved by the same amount under maximum differential pressure conditions relative to the Clow benct test data.

TABLE A-1

DEFINITION OF TERMS USED IN TABLES B-1 THRU C-2

PATM	- Atmospheric pressure
PSU	- Upstream static pressure
GAS=A	- Gas analyzed assumed to have properties similar to air
UF	- Analysis if for unchoked flow (sub sonic gas velocity)
W80	- Flow rate with disc in full open position
DV	- Nominal valve size
TSU	- Static upstream temperature in degrees Rankine
Gamma	- Specific heat ratio for gas selected
Option 1	- Program parameter selection made (important only to person running program)
ES	- English system of units used
MW	- Molecular weight of gas selected
DP80	- Pressure drop (ΔP) across valve in full open position for given flow conditions
Alpha	- Angle of valve disc off of seat for double flange style valve with seat at 10° angle relative to valve flange face
CF	- Mass flow coefficient from experimental data
WR	- Portion of full open flow for selected disc angle which will pass thru valve for given flow conditions
DPS	- Downstream static pressure (PSIA)
POU	- Upstream stagnation pressure
PSC	- Downstream static pressure for onset of valve choking
POD	- Downstream stagnation pressure
TQRI	- Torque coefficient based on experiments
W	- Mass of gas flowing thru valve
TQ	- Torque induced on valve disc and stem due to aerodynamic flow for conditions specified in a straight piperun at onset of choked flow or less than choked flow. See definition of TQA below.
TQA	- Torque induced on valve disc and stem due to aerodynamic flow for choked conditions specified in a straight piperun.

TABLE B-1
NORMAL FLOW CALCULATIONS 8" INTAKE VALVES
TAG NOS. 1VQ042, 1VQ043, 2VQ042, 2VQ043
(STRAIGHT PIPE UPSTREAM)

CASE: COM-ED/LASALLE UNITS 1 & 2
DATE: 07-03-85
PATM: 14.70(PSIA)
PSU = 16.70(PSIA) TSU = 594.67(R)
MEDIUM: GAS = A GAMMA = 1.40
FLOW = UF OPTION = 1
W80 = 1100.00(SCFM)
DV = 8.000(IN)

UNITS SYSTEM: ES
SHAFT: DS

MW = 29.0

OUTPUT DATA

SOLUTION: DP80 = .05(PSIG)

PSD/POU = .9957

ALPHA	CF	WR	DPS/PSU	PSU/POU	PSC/POU	POD/POU	TQR1
80.0	.55[illegible]9	1.0000	.0030	.9987	.7532	.9417	-.0404
75.0	.5487	.9942	.0030	.9987	.7528	.9290	-.0598
70.0	.5329	.9655	.0031	.9988	.7507	.9166	-.0735
65.0	.5063	.9174	.0032	.9989	.7464	.9045	-.0827
60.0	.4709	.8531	.0034	.9990	.7396	.8928	-.0882
55.0	.4285	.7763	.0035	.9992	.7298	.8813	-.0910
50.0	.3810	.6903	.0037	.9994	.7169	.8703	-.0918
45.0	.3304	.5987	.0038	.9995	.7007	.8595	-.0911
40.0	.2786	.5047	.0040	.9997	.6815	.8490	-.0896
35.0	.2273	.4119	.0041	.9998	.6600	.8389	-.0875
30.0	.1786	.3235	.0042	.9999	.6370	.8292	-.0851
25.0	.1341	.2430	.0042	.9999	.6138	.8197	-.0826
20.0	.0957	.1734	.0043	1.0000	.5918	.8106	-.0800
15.0	.0650	.1178	.0043	1.0000	.5730	.8019	-.0772
10.0	.0438	.0793	.0043	1.0000	.5591	.7934	-.0739
5.0	.0334	.0605	.0043	1.0000	.5523	.7853	-.0698

ALPHA (DEG)	YCV (...)	W (LBM/HR)	TQ (IN-LBF)
80.0	1263.45	4930.95	-1.04
75.0	1253.12	4902.47	-1.55
70.0	1202.20	4761.04	-1.95
65.0	1120.68	4523.48	-2.27
60.0	1018.83	4206.78	-2.54
55.0	905.19	3827.97	-2.75
50.0	786.64	3404.04	-2.90
45.0	668.22	2952.00	-3.00
40.0	553.69	2488.73	-3.05
35.0	445.60	2031.00	-3.07
30.0	346.46	1595.34	-3.04
25.0	258.31	1198.01	-3.00
20.0	183.49	854.85	-2.93
15.0	124.44	581.10	-2.84
10.0	83.67	391.17	-2.72
5.0	63.83	298.57	-2.58

TABLE B-2
NORMAL FLOW CALCULATIONS 26" INTAKE VALVES
TAG NOS. 1VQ026, 1VQ027, 1VQ029, 1VQ030, 2VQ026,
2VQ027, 2VQ029, 2VQ030
(STRAIGHT PIPE UPSTREAM)

CASE: COM-ED/LASALLE UNITS 1 & 2
DATE: 07-03-85
PATM: 14.70(PSIA)
PSU = 16.70(PSIA) TSU = 594.67(R)
MEDIUM: GAS = A GAMMA = 1.40
FLOW = UF OPTION = 1
W80 = 11000.00(SCFM)
DV = 26.000(IN)

UNITS SYSTEM: ES
SHAFT: DS

MW = 29.0

OUTPUT DATA

SOLUTION: DP80 = .03(PSIG)

PSD/POU = .9969

ALPHA	CF	WR	DPS/PSU	PSU/POU	PSC/POU	POD/POU	TQR1
80.0	.6148	1.0000	.0019	.9988	.7589	.9419	.0622
75.0	.6113	.9942	.0020	.9988	.7587	.9292	.0758
70.0	.5936	.9655	.0020	.9989	.7575	.9168	.0812
65.0	.5640	.9174	.0021	.9990	.7547	.9047	.0810
60.0	.5245	.8531	.0023	.9991	.7494	.8929	.0771
55.0	.4773	.7763	.0024	.9993	.7410	.8815	.0714
50.0	.4244	.6903	.0026	.9994	.7288	.8704	.0650
45.0	.3681	.5987	.0027	.9996	.7130	.8596	.0591
40.0	.3103	.5047	.0028	.9997	.6935	.8492	.0541
35.0	.2532	.4119	.0029	.9998	.6712	.8391	.0502
30.0	.1989	.3235	.0030	.9999	.6469	.8293	.0474
25.0	.1494	.2430	.0030	.9999	.6220	.8199	.0449
20.0	.1066	.1734	.0031	1.0000	.5982	.8108	.0419
15.0	.0725	.1178	.0031	1.0000	.5776	.8020	.0371
10.0	.0488	.0793	.0031	1.0000	.5624	.7935	.0287
5.0	.0372	.0605	.0031	1.0000	.5547	.7854	.0148

ALPHA (DEG)	YCV (...)	W (LBM/HR)	TQ (IN-LBF)
80.0	15726.02	49309.55	35.46
75.0	15581.55	49024.73	43.49
70.0	14883.81	47610.39	48.17
65.0	13779.03	45234.83	50.57
60.0	12430.50	42067.87	51.18
55.0	10959.67	38279.62	50.44
50.0	9456.81	34040.42	48.82
45.0	7988.19	29519.96	46.74
40.0	6586.53	24887.33	44.73
35.0	5281.94	20309.99	43.02
30.0	4095.19	15953.37	41.63
25.0	3048.09	11980.08	40.16
20.0	2163.12	8548.47	37.89
15.0	1465.87	5810.96	33.74
10.0	985.20	3911.69	26.23
5.0	751.63	2985.67	13.54

TABLE B-3
NORMAL FLOW CALCULATIONS 26" EXHAUST VALVES
TAG NOS. 1VQ031, 1VQ034, 1VQ036, 1VQ040,
2VQ031, 2VQ034, 2VQ036, 2VQ040
(STRAIGHT PIPE UPSTREAM)

CASE: COM-ED/LASALLE UNITS 1 & 2
DATE: 07-03-85 UNITS SYSTEM: ES
PATM: 14.70(PSIA) SHAFT: US
PSU = 16.70(PSIA) TSU = 594.67(R)
MEDIUM: GAS = A GAMMA = 1.40 MW = 29.0
FLOW = UF OPTION = 1
WBO = 11000.00(SCFM)
DV = 26.000(IN)

OUTPUT DATA

SOLUTION: DP80 = .03(PSIG)

PSD/POU = .9968

ALPHA	CF	WR	DPS/PSU	PSU/POU	PSC/POU	POD/POU	TQR1
80.0	.6033	1.0000	.0021	.9988	.7582	.9967	.0250
75.0	.5921	.9814	.0021	.9989	.7574	.9925	.0839
70.0	.5698	.9444	.0022	.9990	.7553	.9890	.1194
65.0	.5381	.8918	.0023	.9991	.7514	.9862	.1371
60.0	.4986	.8264	.0024	.9992	.7450	.9842	.1421
55.0	.4531	.7510	.0026	.9993	.7357	.9831	.1385
50.0	.4031	.6682	.0027	.9995	.7232	.9826	.1300
45.0	.3504	.5808	.0028	.9996	.7074	.9830	.1194
40.0	.2967	.4917	.0030	.9997	.6885	.9842	.1089
35.0	.2435	.4036	.0030	.9998	.6671	.9862	.0997
30.0	.1926	.3193	.0031	.9999	.6439	.9890	.0926
25.0	.1457	.2414	.0032	.9999	.6200	.9926	.0875
20.0	.1043	.1729	.0032	1.0000	.5969	.9970	.0837
15.0	.0702	.1164	.0032	1.0000	.5762	1.0000	.0797
10.0	.0450	.0747	.0032	1.0000	.5599	1.0000	.0733
5.0	.0305	.0505	.0032	1.0000	.5501	1.0000	.0615

ALPHA (DEG)	YCV (...)	W (LBM/HR)	TQ (IN-LBF)
80.0	15256.98	49309.55	15.13
75.0	14817.58	48391.98	51.90
70.0	13987.26	46568.86	76.72
65.0	12875.29	43976.50	92.71
60.0	11603.09	40751.25	101.55
55.0	10254.77	37029.44	104.65
50.0	8886.25	32947.31	103.56
45.0	7547.25	28641.37	99.64
40.0	6267.97	24247.82	94.35
35.0	5064.43	19903.05	89.15
30.0	3960.05	15743.36	84.72
25.0	2970.49	11905.05	81.38
20.0	2115.74	8524.45	78.68
15.0	1419.76	5737.80	75.37
10.0	909.64	3681.40	69.49
5.0	615.24	2491.42	58.41

TABLE C-1
EMERGENCY FLOW CALCULATIONS ALL 8" VALVES
(STRAIGHT PIPE UPSTREAM)

CASE: COM-ED/LASALLE UNITS 1 & 2
DATE: 07-03-85 UNITS SYSTEM: ES
PATM: 14.70(PSIA) SHAFT: US
PSU = 59.70(PSIA) TSU = 799.67(R)
MEDIUM: GAS = A 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 = 30.94(LBM/S)

NOTE: TQ 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	.9784	.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	.2518	.9998	.5470	.7487	.1295

ALPHA (DEG)	YCV (...)	W (LBM/HR)	TQ (IN-LBF)	TQA (IN-LBF)
80.0	1152.62	111393.94	522.60	1990.35
75.0	1121.44	109321.09	774.60	2916.24
70.0	1061.79	105202.53	976.20	3596.82
65.0	981.71	99346.16	1134.14	4064.40
60.0	888.66	92060.06	1250.44	4347.27
55.0	788.71	83652.22	1325.59	4471.86
50.0	686.43	74430.44	1360.77	4464.43
45.0	585.15	64703.00	1359.27	4352.11
40.0	487.29	54777.64	1326.94	4163.26
35.0	394.75	44962.44	1272.01	3927.31
30.0	309.16	35565.42	1204.49	3674.41
25.0	232.16	26894.39	1135.46	3434.88
20.0	165.49	19257.37	1076.37	3238.68
15.0	111.11	12962.13	1038.46	3115.06
10.0	71.20	8316.55	1032.39	3092.27
5.0	48.16	5628.31	1068.14	3197.56

TABLE C-2
EMERGENCY FLOW CALCULATIONS ALL 26" VALVES
(STRAIGHT PIPE UPSTREAM)

CASE: CON-ED/LASALLE UNITS 1 & 2

DATE: 07-03-85 UNITS SYSTEM: ES
PATM: 14.70(PSIA) SHAFT: US
PSU = 59.70(PSIA) TSU = 799.67(R)
MEDIUM: GAS = A GAMMA = 1.40 MW = 29.0
FLOW = CF OPTION = 2
DV = 26.000(IN)

OUTPUT DATA

CHOKING PRESSURE RATIOS: PSC/POU = .758 DPS/PSU = .162
SOLUTION: W80 = 391.95(LBM/S)

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

PSD/POU = .7582

ALPHA	CF	WR	DPS/PSU	PSU/POU	PSC/POU	POD/POU	TQR1
80.0	.6033	1.0000	.1622	.9050	.7582	.8864	.0119
75.0	.5921	.9814	.1658	.9089	.7574	.8754	.0706
70.0	.5698	.9444	.1726	.9164	.7553	.8628	.1055
65.0	.5381	.8918	.1815	.9263	.7514	.8491	.1225
60.0	.4986	.8264	.1913	.9375	.7450	.8349	.1267
55.0	.4531	.7510	.2011	.9491	.7357	.8210	.1223
50.0	.4031	.6682	.2104	.9602	.7232	.8079	.1131
45.0	.3504	.5808	.2185	.9703	.7074	.7961	.1018
40.0	.2967	.4917	.2254	.9789	.6885	.7857	.0907
35.0	.2435	.4036	.2309	.9859	.6671	.7771	.0811
30.0	.1926	.3193	.2351	.9912	.6439	.7702	.0736
25.0	.1457	.2414	.2380	.9950	.6200	.7651	.0683
20.0	.1043	.1729	.2398	.9974	.5969	.7617	.0644
15.0	.0702	.1164	.2409	.9988	.5762	.7597	.0603
10.0	.0450	.0747	.2414	.9995	.5599	.7591	.0538
5.0	.0305	.0505	.2416	.9998	.5501	.7596	.0421

ALPHA (DEG)	YCV (...)	W (LBM/HR)	TQ (IN-LBF)	TQA (IN-LBF)
80.0	16198.48	1411022.50	2030.81	9438.94
75.0	15654.47	1384765.00	12332.82	56069.08
70.0	14644.37	1332596.00	19349.07	84499.62
65.0	13343.54	1258414.50	23873.80	99175.75
60.0	11899.62	1166121.50	26332.77	103783.87
55.0	10416.20	1059619.50	27070.24	101463.97
50.0	8957.32	942807.50	26483.50	94905.25
45.0	7560.31	819590.50	25036.18	86358.94
40.0	6246.52	693866.00	23205.26	77598.37
35.0	5029.92	569537.75	21401.16	69862.00
30.0	3922.28	450505.62	19894.31	63799.75
25.0	2936.64	340670.37	18762.04	59433.70
20.0	2089.41	243932.19	17861.29	56139.91
15.0	1401.30	164190.81	16823.91	52645.25
10.0	897.50	105345.37	15063.22	47034.77
5.0	606.99	71293.59	11781.93	36759.22

5.3.2 Aerodynamic Torques 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 A-1 thru C-2 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 condition, closing cycle). All torque values shown as positive, tend to close the valves. All torque values shown as negative tend to open the valve or resist closure. The meanings of the other listings can be found in Table A-1 and Section 7.0, References, 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.

For Figures 11,12,14 and 18 (Sect.3.2), the configurations for 1VQ027, 2VQ027, 1VQ030, 2VQ030, 1VQ036 and 2VQ036 indicate upstream piping is at a sufficient distance so all the valves will respond as if under fully developed straight run pipe flow. Thus, for all these valves the torque modification factor comparing straight line flow to actual flow is 1.0 as indicated in Tables 15,18 & 21.

For Figures 11 and 12, the configurations for 1VQ026 and 2VQ026 are best represented by the case of two valves in series at a seperation of two valve diameters (actual seperation is approximately 2½ dia.) with an orientation similar to 3 per Figure 27 . It was assumed that the upstream valve was frozen in a position which produced a worst case torque on the down stream valve. The corresponding torque modification factors are listed in Table 16 .

For Figure 13 an elbow with less than 90° turn is the upstream element for valves 1VQ029 and 2VQ029. Since the elbow is not a full 90° turn elbow and the spacing to the valve is .62 diameters, a mitered 90° elbow at 2 diameters in configuration 2 (see Fig.27) was selected for calculation basis. Since the elbow is not a full 90° turn elbow actual flow is probably closer to straightline flow. This geometry was selected as a worst case comparison. Torque modification factors (9.4 at full open) are presented in Table 17 for this orientation.

For Figure 15 the configuration of valves 1VQ031 and 2VQ031 best represented by a mitered elbow at 2 diameters in geometry 1 (see Figure 27). It should be noted that for approximately the first 3° to 5° the aerodynamic flow tends to hold the valve open, yet the actuator torque is sufficiently large to overcome this resistance.

For figures 16 and 17, the configuration of valves 1VQ034 and 2VQ034 are best represented by comparison to data for a mitered elbow 2 diameters upstream per geometry 2. Corresponding torques are presented in Table 20 .

For Figures 19 and 20 the configuration of valves 1VQ040, 2VQ040, 1VQ042 and 2VQ042 are best represented by comparison to a mitered elbow 4 diameters upstream per geometry 2. The torque modification factors in Tables 22 and 23 show that the effects of the elbow at 4 diameters or greater is of little concern.

For Figure 21 the configuration of valves 1VQ043 and 2VQ043 are best represented by comparison to data for a mitered elbow 2 diameters upstream per geometry 3.

The tables show model test valve angle and actual valve angle

for the supplied units. There is a 10^{0} difference between these due to the seat angle design differences explained in previous sections. It is reasonable to expect all angles over 20^{0} to be a proper representation of the magnitude and direction of torques. At 20^{0} or below, the magnitude may differ but the direction is correctly indicated. Since peak torques occur in the 60 to 80^{0} range, these low end torques are of no consequence.

Table 15

Valve No. 1VQ027 & 2VQ027 (26")

Model Data For Torque Modification: Valves under straight line flow conditions
All torques in In-lbs.

Model Test Valve Angle	Actual Valve Angle	Torque for Straight Flow Normal	Torque for Straight Flow Maximum	Torque Modification Factor	Torque for Installed Condition Normal	Torque for Installed Condition Maximum
80	90	-36	9,439	1.0	-36	9,439
70	80	-48	84,500	1.0	-48	84,500
60	70	-51	103,784	1.0	-51	103,784
50	60	-49	94,905	1.0	-49	94,905
40	50	-45	77,598	1.0	-45	77,598
30	40	-42	63,800	1.0	-42	63,800
20	30	-37	56,140	1.0	-37	56,140
10	20	-26	47,035	1.0	-26	47,035

Table 16

Valve No. 1VQ026 & 2VQ026 (26")

Model Data For Torque Modification: 2 valves in series at 2 dia. separation orientation 3
All torques in In-lbs.

Model Test Valve Angle	Actual Valve Angle	Torque for Straight Flow Normal	Torque for Straight Flow Maximum	Torque Modification Factor	Torque for Installed Condition Normal	Torque for Installed Condition Maximum
80	90	-36	9,439	2.0	-72	18,878
70	80	-48	84,500	1.28	-61	108,160
60	70	-51	103,784	1.31	-67	135,957
50	60	-49	94,905	1.38	-67	130,969
40	50	-45	77,598	1.43	-64	110,965
30	40	-42	63,800	1.20	-50	76,560
20	30	-37	56,140	1.16	-43	65,122
10	20	-26	47,035	1.14	-30	53,620

Table 17

Valve No. 1VQ029 & 2VQ029 (26")

Model Data For Torque Modification: All torques in In-lbs.

Mitered elbow 2 diameter upstream Geometry 2

Model Test Valve Angle	Actual Valve Angle	Torque for Straight Flow Normal	Maximum	Torque Modification Factor	Torque for Installed Condition Normal	Maximum
80	90	-36	9,439	9.4	-338	88,727
70	80	-48	84,500	1.54	-74	130,130
60	70	-51	103,784	1.21	-62	125,579
50	60	-49	94,905	1.17	-57	111,039
40	50	-45	77,598	1.03	-46	79,926
30	40	-42	63,800	1.00	-42	63,800
20	30	-37	56,140	.96	-36	53,894
10	20	-26	47,035	.89	-23	41,861

Table 18

Valve No. 1VQ030 &* 2VQ030 (26")

Model Data For Torque Modification: All torques in In-lbs.

Valves under straight line flow conditions

Model Test Valve Angle	Actual Valve Angle	Torque for Straight Flow Normal	Maximum	Torque Modification Factor	Torque for Installed Condition Normal	Maximum
80	90	-36	9,439	1.0	-36	9,439
70	80	-48	84,500	1.0	-48	84,500
60	70	-51	103,784	1.0	-51	103,784
50	60	-49	94,905	1.0	-49	94,905
40	50	-45	77,598	1.0	-45	77,598
30	40	-42	63,800	1.0	-42	63,800
20	30	-37	56,140	1.0	-37	56,140
10	20	-26	47,035	1.0	-26	47,035

Table 19

Valve No. 1VQ031 & 2VQ031 (26")

Model Data For Torque Modification: Mitered elbow 2 diameters upstream Geometry 1
All torques in In-lbs.

Model Test Valve Angle	Actual Valve Angle	Torque for Straight Flow Normal	Maximum	Torque Modification Factor	Torque for Installed Condition Normal	Maximum
80	90	15	9,439	-4.46	-67	- 42,098
70	80	77	84,500	+1.56	+120	131,820
60	70	102	103,784	1.16	118	120,389
50	60	104	94,905	1.21	125	114,835
40	50	94	77,598	1.20	113	93,118
30	40	85	63,800	1.12	95	71,456
20	30	79	56,140	1.0	79	56,140
10	20	69	47,035	.89	61	41,861

Table 20

Valve No. 1VQ034 & 2VQ034 (26")

Model Data For Torque Modification: Mitered elbow 2 diameter upstream Geometry 2
All torques in In-lbs.

Model Test Valve Angle	Actual Valve Angle	Torque for Straight Flow Normal	Maximum	Torque Modification Factor	Torque for Installed Condition Normal	Maximum
80	90	15	9,439	9.4	-338	88,727
70	80	77	84,500	1.54	-74	130,130
60	70	102	103,784	1.21	-62	125,579
50	60	104	94,905	1.17	-57	111,039
40	50	94	77,598	1.03	-46	79,926
30	40	85	63,800	1.00	-42	63,800
20	30	79	56,140	.96	-36	53,894
10	20	69	47,035	.89	-23	41,861

Table 21

Valve No. 1VQ036 & 2VQ036 (26")

Model Data For Torque Modification: All torques in In-lbs.

Valves under straight line flow conditions

Model Test Valve Angle	Actual Valve Angle	Torque for Straight Flow Normal	Maximum	Torque Modification Factor	Torque for Installed Condition Normal	Maximum
80	90	15	9,439	1.0	15	9,439
70	80	77	84,500	1.0	77	84,500
60	70	102	103,784	1.0	102	103,784
50	60	104	94,905	1.0	104	94,905
40	50	94	77,598	1.0	94	77,598
30	40	85	63,800	1.0	85	63,800
20	30	79	56,140	1.0	79	56,140
10	20	69	47,035	1.0	69	47,035

Table 22

Valve No. 1VQ040 & 2VQ040 (26")

Model Data For Torque Modification: All torques in In-lbs.

Mitered elbow 4 diameters upstream Geometry 2

Model Test Valve Angle	Actual Valve Angle	Torque for Straight Flow Normal	Maximum	Torque Modification Factor	Torque for Installed Condition Normal	Maximum
80	90	15	9,439	2.0	30	18,878
70	80	77	84,500	1.36	105	114,920
60	70	102	103,784	1.15	117	119,352
50	60	104	94,905	1.0	104	94,905
40	50	94	77,598	1.0	94	77,598
30	40	85	63,800	1.0	85	63,800
20	30	79	56,140	1.0	79	56,140
10	20	69	47,035	1.0	69	47,035

Table 23

Valve No. 1VQ042 & 2VQ042 (8")

Model Data For Torque Modification: All torques in In-lbs.

Mitered elbow 4 diameters upstream Geometry 2

Model Test Valve Angle	Actual Valve Angle	Torque for Straight Flow Normal	Maximum	Torque Modification Factor	Torque for Installed Condition Normal	Maximum
80	90	-1	1990	2.0	-2	3980
70	80	-2	3597	1.59	-3	5719
60	70	-3	4347	1.20	-4	5216
50	60	-3	4464	1.04	-3	4643
40	50	-3	4163	1.03	-3	4288
30	40	-3	3674	1.0	-3	3674
20	30	-3	3239	1.0	-3	3239
10	20	-3	3092	1.0	-3	3092

Table 24

Valve No. 1VQ043 & 2VQ043 (8")

Model Data For Torque Modification: All torques in in-lbs.

Mitered elbow 2 diameter upstream Geometry 3

Model Test Valve Angle	Actual Valve Angle	Torque for Straight Flow Normal	Maximum	Torque Modification Factor	Torque for Installed Condition Normal	Maximum
80	90	-1	1990	1.12	-1	2229
70	80	-2	3597	1.26	-3	4532
60	70	-3	4347	1.12	-3	4869
50	60	-3	4464	1.03	-3	4598
40	50	-3	4163	1.03	-3	4163
30	40	-3	3674	1.0	-3	3674
20	30	-3	3239	1.0	-3	3239
10	20	-3	3092	1.0	-3	3092

5.3.3 Conclusions Concerning Valve Operability

To determine whether a given valve actuator assembly will operate under the required flow conditions, two sets of criteria must be applied; one for pneumatic actuated valves and one for electric actuated valves. The following criteria apply for pneumatic and electric actuated valves:

1. Actuator torque output must overcome with sufficient margin the worst case torque resisting valve closure.
2. Peak aerodynamic induced closing torques must not exceed actuator or valve design torques.

For LOCA flow conditions it can be seen in Tables 15 thru 24 that all aerodynamic torques for all valves except 1VQ031 and 2VQ031 tend to aid valve closure for all disc angles. For valves 1VQ031 and 2VQ031 aerodynamic torques for the first 3 to 5 degrees from full open resist closure. Pertinent torques for air operated valves are listed in Table 25.

TABLE 25

Pneumatic Actuated Valve Torques
Torques (in-lb)

Valve Size	Valve No.	Valve Design Torque	1	2	3	4	5
26"	1VQ027 1VQ030 1VQ036 2VQ027 2VQ030 2VQ036	145,000	none	none req'd	103,784	401,650	145,000
26"	1VQ026 2VQ026	145,000	none	none req'd	135,957	401,650	145,000
26"	1VQ029 2VQ029	145,000	none	none req'd	130,130	401,650	145,000
26"	1VQ031 2VQ031	145,000	42,098	129,000	131,820	401,650	145,000
26"	1VQ034 2VQ034	145,000	none	none req'd	130,130	401,650	145,000
26"	1VQ040 2VQ040	145,000	none	none req'd	119,352	401,650	145,000
8"	1VQ042 2VQ042	16,100	none	none req'd	5,719	44,919	16,100
8"	1VQ043 2VQ043	16,100	none	none req'd	4,869	44,919	16,100

1 Worst case closure resisting aerodynamic torque

2 Actuator torque used to overcome aerodynamic torque

3 Maximum aerodynamic torque

4 Torque to yield actuator key

5 Actuator safe structural torque (min)

From the preceding data it can be seen that the minimum actuator torque margin over that required to overcome worst case aerodynamic torque is better than 3.06. The safety factor is obtained even after full containment pressure has been developed!

From 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. Further, no credit has been taken for activation of the first valve under back pressure conditions produced by closure of the second valve or the effect of pressure drop across the first valve or closure of the second valve.

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 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 derection. The test performed was an air test in which the smallest detectable leakage was .006 SCFM.

TABLE 26

VALVE SEALING CHARACTERISTICS

VALVE MARK NO.	VALVE SIZE (IN.)	TEST PRESSURE PSIG	LEAKAGE SCFM Pressurized Side	
			Shaft Side	Clamp Ring Side
1VQ027	26	2,50	0/0	0/0
1VQ030	26	2,50	0/0	0/0
1VQ036	26	2,50	0/0	0.0035/0
2VQ027	26	2,50	0/0	0/0
2VQ030	26	2,50	0/0	0/0.012
2VQ036	26	2,50	0/0	0/0.007
1VQ026	26	2,50	0/0	0/0.007
2VQ026	26	2,50	0/0	0/0
1VQ029	26	2,50	0/0	0/0
2VQ029	26	2,50	0/0	0/0
1VQ031	26	2,50	0/0	0.0008/0
2VQ031	26	2,50	0/0	0/0.005
1VQ034	26	2,50	0/0	0/0
2VQ034	26	2,50	0/0	0/0
1VQ040	26	2,50	0/0	0/0
2VQ040	26	2,50	0/0	0/0
1VQ042	8	2,50	0/0	0/0.0027
2VQ042	8	2,50	0/0	0/0
1VQ043	8	2,50	0/0	0/0
2VQ043	8	2,50	0/0	0/0

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 Internation Cycling Test (reference 7.0 D3). 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.

6.5 RESPONSE TO 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 valves disc shapes are identical or only slightly different. All differences, although small, are fully documented. (Section 5.1)
3. Installation effects were accounted for in all cases, but downstream piping back pressure was not, since this produces a more conservative calculation. (Section 5.3.1)
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

flow induced loads. Since Clow's seat/seal design is conical, no special considerations for low seat temperature is required. (Section 5.3.2, 2.2.1)

13. Clow selects operators for each unit with maximum operating torque much larger than that produced by flow interaction with the disc. (Section 5.3.2)
14. Since Clow's seating torque is higher than required running torques, the unit torque switch settings are compatible. (Section 5.3.2)
15. Such conditions are presented to the actuator manufacturer by supplying a copy of the customer's spec. The reduced voltage does have a small (less than a few tenths of a second) effect on operating times. Emergency mode power source is the buyer's responsibility. (Section 2.3.3.3)
16. Yes, handwheels automatically disengage upon electric activation.
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. (Section 1.2, 6.0, 8.0)

18. All operators and solenoid valves installed by Clow are qualified to appropriate IEEE requirements by testing. (Section 2.3.2)
19. All tests are summarized in the supplied qualification report and are documented by separate test reports. (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).

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-852200-2 (April 8, 1985)
   "Seismic Qualification Analysis of Clow 26 inch Wafer Stop Valve"
   Generic Report for Clow 26" Wafer Design , Clow P.O. No.30-14110

2. Technical Report PEI-TR-83-24 Rev. A (July 18, 1983)
   "Seismic Qualification Analysis of Clow 8 inch Wafer Stop Valve"
   Generic Report for Clow 8" Wafer Design

3. Technical Proposal PEI-TP-85-22 (Jan. 2, 1985)
   "Seismic Qualification of Valves and Actuators for use in LaSalle County Station Units 1 & 2"
   Site specific Qualification Plan

4. Technical Report PEI-TR-852202-3 (June 18, 1985)
   "Qualification Report of Clow 8-Inch and 26-Inch Wafer Valves Assemblies , Clow Job Number 84-2842-(N). CECO Valve Tag Numbers
   8": 1VQ042, -43, 2VQ042, -43
   26": 1VQ026, -27, -29, -30, -31. -34, -36, -40
   2VQ026, -27, -29, -30, -31, -34, -36, -40"
   Site specific report showing qualification of 8" and 26" valves and actuators to CECO Specs.

B Seismic Qualification Test Reports

Prepared by: Patel Engineers
Huntsville, Alabama

1. Technical Report PEI-TR-83-29 Rev. A (Aug. 10,1983)
   "Seismic Qualification of Clow Wafer Stop Valve Assemblies Job Numbers 82-2053-01(N), -02(N), -03(N), -05(N), -07(N)"
   Seismic vibration tests of Bettis Actuators and static load test of Actuator and valve assemblies

2. Clow Corp. Addendum I to Patel Technical Report PEI-TR-83-29(Aug.16,1983)
   "Static Load Test and Seismic Qualification of Clow Wafer Stop Valve Assemblies, 24" HBB-BF-MO-57-115, -135, -147
   18" HBB-BF-MO-57-112"
   Static load test of 24" and 18" valves with large electric actuator

REFERENCES (con't)

prepared by: Wyle Laboratories
Huntsville, Alabama

3. Test Procedure 541/0465/WB (May 83)
"Static Load Test Procedure For An 18-Inch Valve Assembly"

4. Test Procedure 46823-1 (June 83)
"Static Load Test Program on an 18" Butterfly Valve Assembly With Limitorque Operator"

5. Report No. 45832-1 "Low Level Seismic Vibration Test Program on a 12" Butterfly Valve Assembly" (Nov. 23, 1981).
Low level biaxial sine sweep resonant search.

6. Report No. 45828-1 "Seismic Simulation Test Program on a Valve Actuator Solenoid Valve" (Nov. 22, 1981). Low level sine sweep resonant search and sine beat test (to 11.0 g max.) for Asco solenoid valve.

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

7. 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.

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.

REFERENCES (con't)

2. Report on "Aerodynamic Torque and Mass Flowrate For Compressible Flow Through Three Geometrically Similar Scale-Model Clow Valves Located Downstream of a 90° Mitered Elbow"

3. Report on "Aerodynamic Torque and Mass Flowrate For Compressible Flow Through Geometrically Similar Scale-Model Clow Valves in Series" (October 1982)

4. Report on "Water Table Investigation of a Two-Dimensional Scale-Model of a 24-Inch Clow Tricentric Butterfly Valve" (November 1982)

D. Other Reports and Information

1. Operating Instructions for Clow Tricentric Wafer Stop Valve covers installation, maintenance, and operating instructions for 84-2842(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. Specification T-3750 "Replacement High Performance Butterfly Valves (Section III), LaSalle County Station-Units 1 and 2 Commonwealth Edison Company, Project No. 6854-30".

2. "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. 14,1983.

3. "A Parametric Study of a Butterfly Valve Utilizing the Hydraulic Analogy" by Bruce A. Coers; Thesis for fulfillment of Master of Science in Mechanical Engineering requirements, Graduate School of Bradley University, Peoria, ILL.,1983.

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 non-routine 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. <u>BRANCH TECHNICAL POSITION</u>

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 five seconds.

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 margin 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 upstream 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^{o} increments if practical).

Valve Position (in degrees - 90^{o} = 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.

APPENDIX B

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

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.