CLOW

Clow Corporation
Engineered Products Division

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# PURGE AND VENT VALVE OPERABILITY

# QUALIFICATION ANALYSIS

Report No. 6-18-85
Rev. A
PREPARED FOR

ILLINOIS POWER COMPANY
CLINTON POWER STATION UNIT 1

by

Steven M. Nondahl

Robert Sansone

Work performed under Baldwin Purchase Order Number - C43005

Clow Job Number: 83-2462(N)

This report covers Equipment Designation:

IVR006A
IVR006B
IVR007A
IVR007B

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| | | | | | |

# CERTIFICATION

This is to certify that all valves (Equipment Nos. IVR006A, IVR006B, IVR007A, IVR007B) have been evaluated for operability under the installed conditions indicated in supplied drawings (M06-1111) and purchasing specifications (BA-K-2882-29).
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 M. Nondahl 6/20/85

Steven M. Nondahl

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# 1. INTRODUCTION

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

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

to determine flow and torque characteristics and application of this test data to the installed condition of the subject valves are presented. Information as to the structural integrity of the valve and operator assembly under seismic and other inplant loadings are also presented. This information, in combination with the supporting detailed technical reports (see 8.0 references), represents a demonstrated qualification of the subject valves to the best of our knowledge for the required service application.

## 1.1 Testing Performed

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

*Instrument Society of America

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

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

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

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

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

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

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

1.2 Qualification Method

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

A. Environmental

All portions of the Clow Tricentric are 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 (Styrene-Butadiene Rubber) binder which may degrade under radiation. Further, the asbestos laminations are shielded by the SST laminations and disc components. Although the asbestos may become embrittled on the periphery, the valve will still perform its sealing function (see Radiation Sensitivity Analysis Report Wyle 17629-01). A

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. (See Bettis Nuclear Qualification Test Report 37274 Rev. A Patel Report PEI-TR-852201-02) (A)

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 the code required Design Report (Clow DR-83-2462(N)), and the Structural Analysis Reports (PEI-TR-852400-1 and PEI-TR-833600-1 Rev. A). 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 static end load due to fluid pressure)

5. Actuator mounting structure

   a. Adaptor flange

   b. Bolting

   Actuators and instruments mounted on the actuators are qualified separately by the manufacturer by general test results.

C. Operability Under Flow

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

The following method is used to show operability:

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

3. Determine a torque modification factor based on the installed (from buyer prints) or a worst case upstream piping condition using the mitered elbow or two valves in series test data.
4. Determine predicted torque values for all disc angles based on 2 and 3 above.
5. Provide tabulation or plot of actuator output torque for all actuator angles.
6. Show that actuator output provides sufficient margin to overcome aerodynamic and other torques (bearing, packing, disc wt.) to close the valve.
7. From the above data, actuator type, and Vought full size test valve data, project a closing rate under the conditions analyzed above.

In the above calculations, the following assumptions are employed:

a. Containment pressure is at a maximum value and full flow is developed before valve starts to close.
b. The pressure downstream of the valve is atmospheric. In the elbow experiment it was noted that downstream

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

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

d. Torque coefficients used in the CVAP program are worst 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. (I.E. most conservative data samples) A

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

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

# 2.0 DESIGN OF VALVE AND ACTUATOR ASSEMBLY

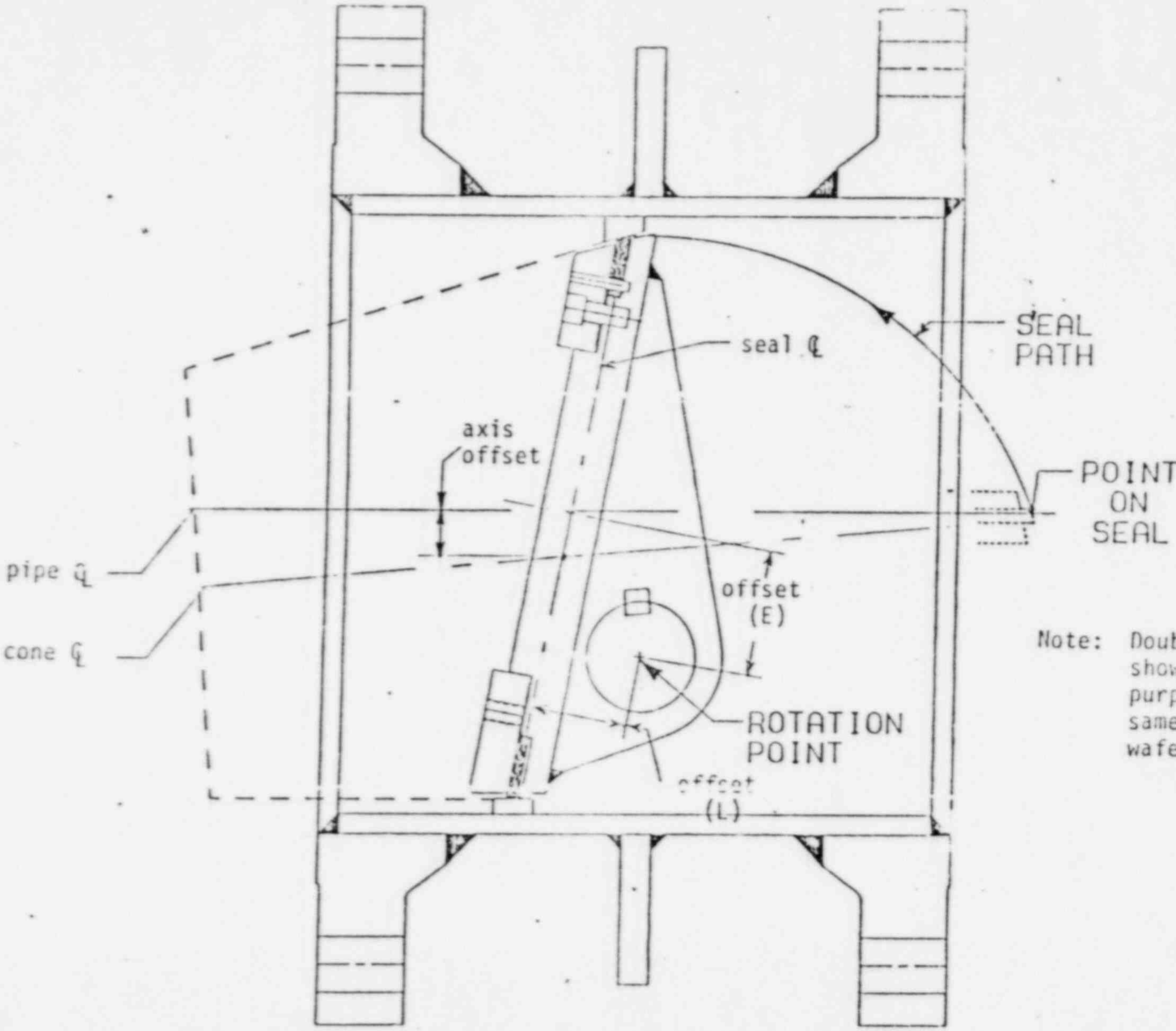
## 2.1 Valve Design

### 2.1.1 Geometry

The Tricentric valve uses a geometry that is unique not only to purge valves but to butterfly valves in general. This feature gives the Tricentric functional characteristics which are desirable in purge valve applications. Thru use of a conical sealing surface, with the cone axis offset from the pipe axis and a rotation point selected so that it is offset from both the pipe axis and the seal plane, a metal to metal seal can be obtained (Figure 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 - TRICENTRIC VALVE OFFSETS

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

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

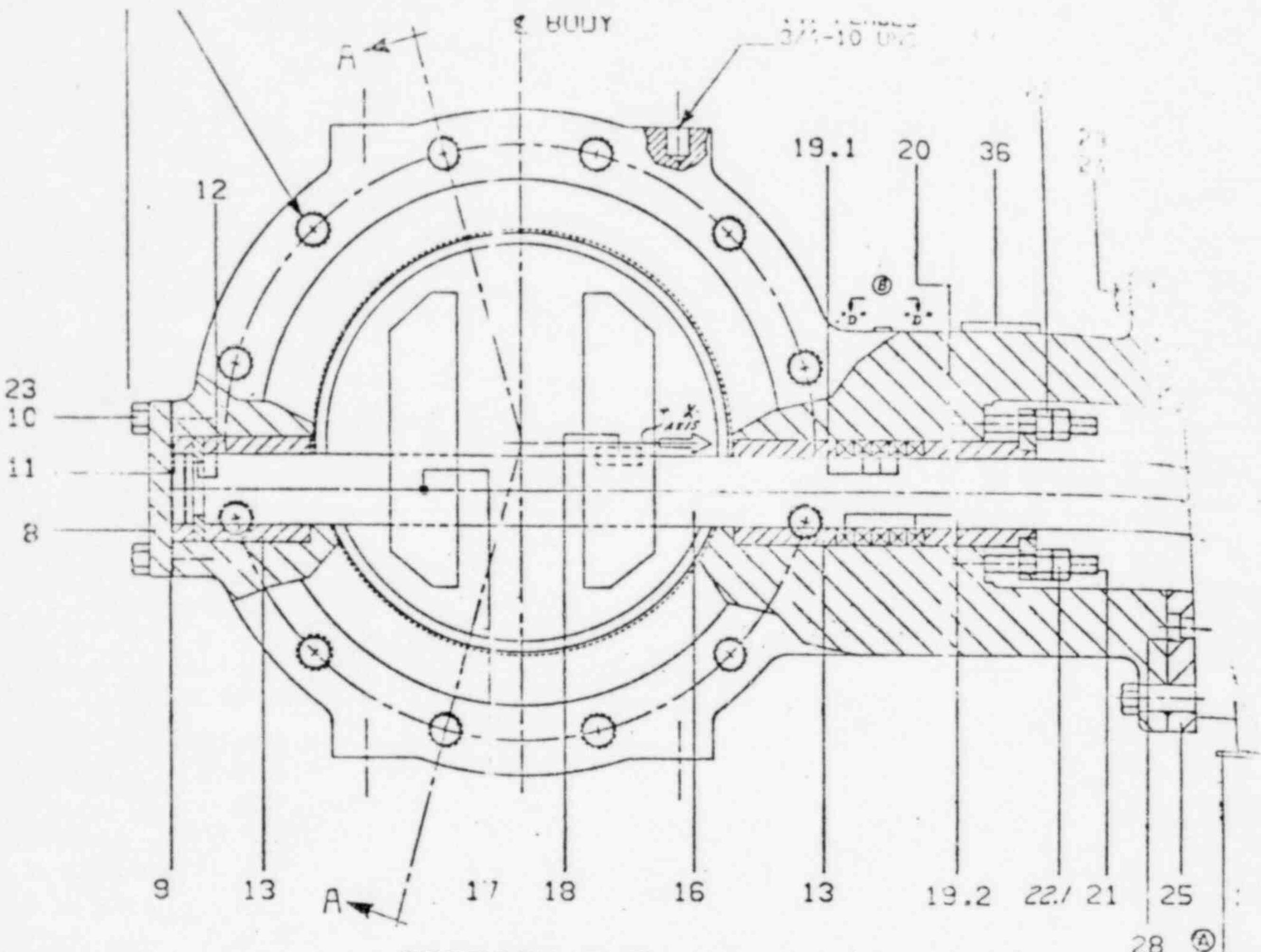
2 offsets is required to provide a valve that has no binding or interference problems as the seal is rotated out of the seat. This relationship is determined analytically to provide the best performance without overdesigning the valve components. All of these features have been incorporated into the lugged wafer body that results in a very rugged and sturdy valve design capable of meeting or exceeding all the requirements set forth in most specifications.

2.1.2 Materials

A complete list of valve component materials used on Purchase Order Number C43005 may be found on the General Arrangement Drawing (D-0809) which follows this section.

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

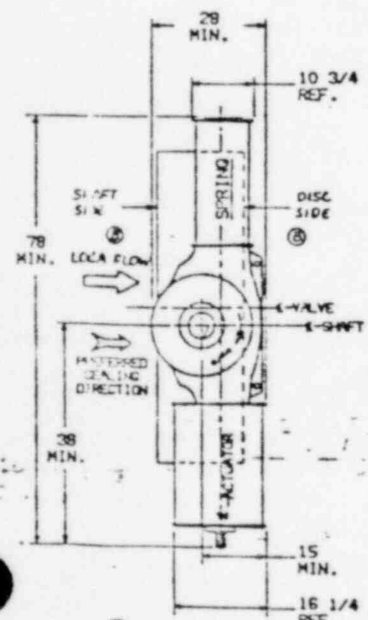
further specified because the seat is welded to the body (SA516 GR 70) and the L grade has a lower carbon content that will reduce the carbide precipitation in the heat affected zone of the seat. The seal is a laminate of 316 SST and asbestos. Both laminants are 1/16 inch thick. The 316 SST was chosen in the "straight" grade since no welding is done on the seal. The asbestos used is made of John Manville style 60 or equal material. The laminated type seal was selected for its ability to seal with less torque than would be required for a solid seal. The laminate allows each SST member to act independently and to conform to the contour of the machined seat as seating torque is applied. The asbestos member not only allows each SST member to act independently but also reduces the seal area in contact with the seat and therefore, results in application of higher normal stresses to the seal for any given seating torque.




SECTION B-B

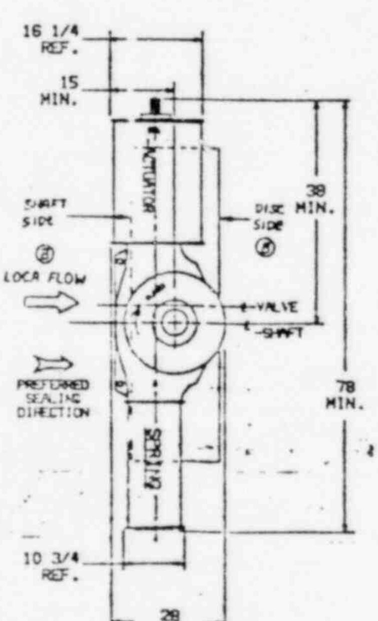
CODE REFERENCE: CLASS 2, SECTION ... ASM
VESSEL CODE, 1980 EDITION INCLUDI... NDA
FLANGE BOLTING DIMENSIONS PER ANS..., C
RAISED F...

DESIGN CONDITIONS:
PRESSURE:
BODY: 285 PSIG, RATING CLA... 1B
DISC: 30 PSID
TEMPERATURE: 185° F
OPERATNG TEMPERATURE: 122° F
PRESSURE TESTS:
SHELL TEST: 450 PSIG (HYDROSTA...
SEAT TEST : SHAFT SIDE W/ 15 P.S...
LEAKAGE PAST SEAT...
Ⓐ VALVE WEIGHT: 510 LBS. (APPROX. ...
ACTUATOR WEIGHT: 638 LBS. (APPRO...
MINIMUM THICKNESSES:
BODY WALL: .44
BEARNG NECK: .17 (THE MIN. ...
BOLT HOLE AND BODY ...
(ABOVE DIM.S' INCLUDE .08 MIN...




Ⓐ VIEW C-C

FOR TAG NOS.:
IVR006B



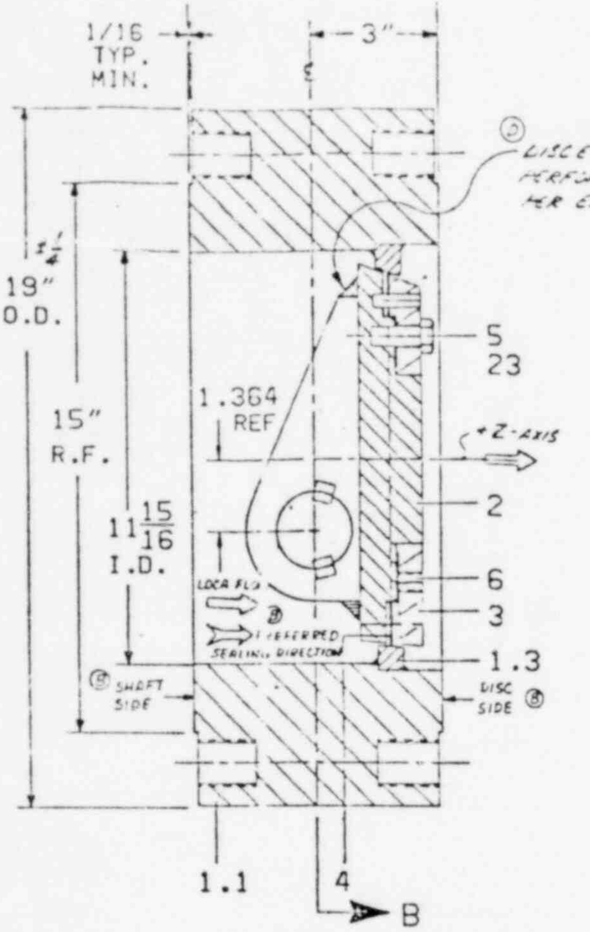

VIEW C-C Ⓑ

FOR TAG NOS.:
IVR006A
IVR007B
Ⓒ IVR007A

| | DESCRIPTION | ITEM | SI... |
|---|---|---|---|
| Ⓒ | DISC BOLTS | 5 | 1/2-1... |
| Ⓒ | COVER PLATE BOLTS | 10 | 1/2-1... |
| Ⓔ | GLAND BOLTS | 22.1 | 1/2-1... |
| Ⓒ | ADAPTOR PLATE MTG BOLTS | 24 | 3/4-... |
| | ACTUATOR MOUNTING BOLTS | 28 | 7/8... |

NOTE: ACTUATOR DIMENSIONS GIVEN AS "MIN.", ARE MINIMUM CLEARANCE DIM.S FOR ACTUATOR/ACCESSORY ADJUSTMENT, PIPING & MAINTENANCE.

AND MATERIAL




SECTION A-A

? END PRESSURE
SUMMER, 1982.
WITH 1/16"

(HYDROSTATIC)
PER MINUTE
(PNEUMATIC)

NGE

## JOB INFORMATION

CUST.: ILLINOIS POWER COMPANY
CLINTON POWER STATION
UNIT 1
BALDWIN ASSOCIATES P. O. NO.:
C43005

CLOW JOB NO.: 83-2462-01(N)

| CLOW SERIAL NUMBERS | VALVE TAG NO.S | C.G. OF VALVE W/ACTUATOR (B) X | Y | Z | |
|---|---|---|---|---|---|
| 83-2462-01(N)-01 | 1VR006A | 15.67 | -2.19 | -3.02 | (A) |
| 83-2462-01(N)-02 | 1VR006B | " | 2.14 | 1.31 | (A) |
| (B) 83-2462-01(N)-03 | 1VR007A | " | -2.19 | -3.02 | (A) |
| 83-2462-01(N)-04 | 1VR007B | " | -2.19 | -3.02 | (A) |

(A) C.G. OF VALVE W/O ACTUATOR:
X=3.81, Y=-.42, Z=.16

■ DENOTES SAFETY RELATED NON-PRESSURE BOUNDARY PARTS.
△ DENOTES PRESSURE RETAINING PARTS.
✳ DENOTES RECOMMENDED SPARE PARTS.

| | ITEM | DESCRIPTION | QTY | MATERIAL | REMARKS |
|---|---|---|---|---|---|
| | 22 / | JAM NUT | 4 | COMM. | |
| | 36 | NAME PLATE | 1 | SST | FASTENED WITH SST DRIVE SCREWS |
| | 33 | PARALLEL KEY | 1 | C-1045 COLD DRAWN | 90,000 PSI MIN. TENSILE |
| (A) | 28 | SOC.HD CAP SCRW | 4 | A 193 GR. B7 | |
| ■ | 26 | ACTUATOR | 1 | COMM. | BETTIS/PAUL-MUNROE T-316-SR2-HD |
| | 25 | ADAPTOR PLATE | 1 | A 516 GR. 70 | |
| (C) | 24 | SOC. HD.CAP SCRW | 6 | A 193 GR. B7 | TO BE LOCKWIRED TOGETHER |
| | 23 | LOCKWIRE | A/R | SST | |
| △ | 22 | NUT | 4 | SA 194 GR. 2H | |
| △ | 21 | STUD | 4 | SA 193 GR. B7 | |
| △ | 20 | GLAND | 1 | SA516 GR.70 W/A 515 | ALT. / A513 |
| ✳ | 19.2 | PACKING | 2 | ASBESTOS | J.C. 187-I |
| ✳ | 19.1 | PACKING | 3 | GRAFOIL | J.C. STYLE 241 |
| | 18 | PARALLEL KEY | 2 | C-1045 COLD DRAWN | 90,000 PSI MIN. TENSILE |
| | 17 | DOWEL PIN | 1 | COMM. ALLOY STEEL | 110,000 PSI MIN. TENSILE, 24-30 RC |
| ■ | 16 | DRIVE SHAFT | 1 | SA 564 T 630 | COND. H-1100 |
| ✳ | 13 | BEARING | 2 | HETCAR H-10 | |
| ✳ | 12 | ANNULAR KEY | 1 | B 505 ALLOY 932 | |
| | 11 | SPACER | 1 | C-1010 THRU C-1030 | ALT. A515 OR A516 GR.70 |
| △ | 10 | HEX HD.CAP SCRW | 4 | SA 193 GR. B7 | TO BE LOCKWIRED TOGETHER |
| ✳ | 9 | GASKET | 1 | GRAFOIL | |
| △ | 8 | COVER PLATE . | 1 | SA 516 GR. 70 | |
| | 6 | SOC. SET SCRW | 2 | COMM. | |
| ■ | 5 | HEX HD.CAP SCRW | 16 | SA 193 GR.B7 | TO BE LOCKWIRED TOGETHER |
| ✳ | 4 | LAMINATED SEAL | 1 | A240 T316 W/ J.M. STYLE 80 | |
| ■ | 3 | CLAMP RING | 1 | SA 516 GR. 70 | |
| (D)△ | 2 | DISC W/ DISC EARS | 1 | SA 516 GR. 70 | |
| ■ | 1.3 | VALVE SEAT | 1 | SA 479 T 316L | ALT. SA 240 T316L |
| △ | 1.1 | VALVE BODY | 1 | SA 515 GR. 70 | |

| | | | |
|---|---|---|---|
| MATERIAL N/A | DO NOT SCALE PRINT | THIS DRAWING IS THE EXCLUSIVE PROPERTY OF CLOW CORPORATION. NO USE WHATSOEVER OF THIS DRAWING FOR ANY PURPOSE IS PERMITTED. NOR REPRODUCTION IN WHOLE OR IN PART SHALL BE MADE WITHOUT THE EXPRESS WRITTEN PERMISSION OF CLOW CORPORATION. | CLOW ENGINEERED PRODUCTS DIVISION |
| HEAT TREAT N/A | UNLESS OTHERWISE SPECIFIED ALL DIMENSIONS IN INCHES | DRAWN [illegible] DATE 11-07-84 | TITLE 12" WAFER STOP-VALVE ASSEMBLY |
| QUANTITY 4 / NO. PER N/A | TOLERANCES DECIMAL ± .010 FRACTIONAL ± 1/8 ANGULAR ± 3° | CHECKED [illegible] DATE 11-14-84 | DRAWING NO. D-0809 △ |
| SCALE NTS | SURFACE FINISH √125 REMOVE BURRS & SHARP EDGES | APPROVED [illegible] DATE 12/[illegible]/84 | JOB NO. 83-2462-01(N) |

2.1.3 Operation

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

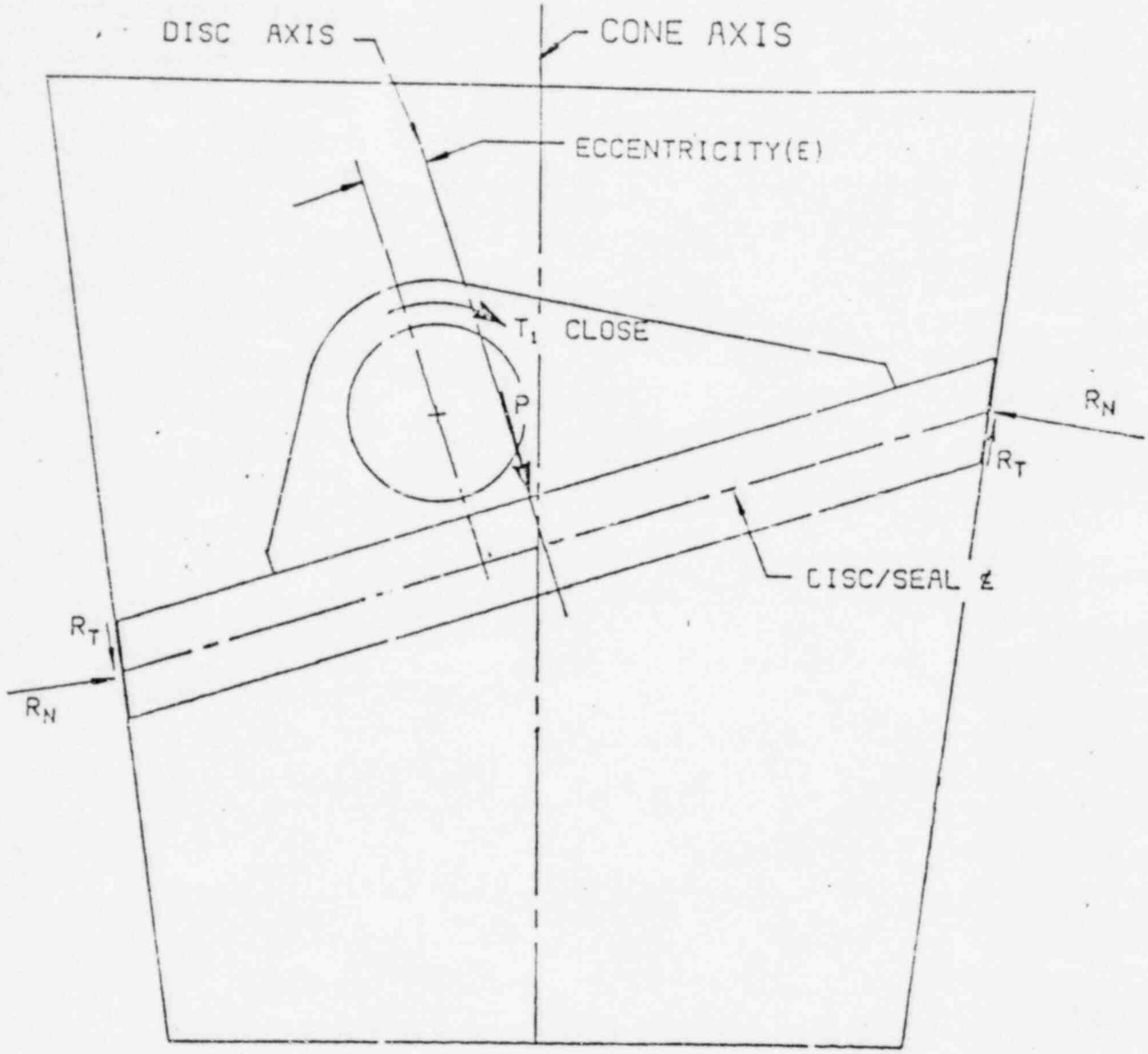
1. Bearing friction torque is the result of the flow or pressure forces acting on the disc which are transmitted to the bearing through the shaft which supports the disc. The bearing friction torque is proportional to these forces acting on the disc and the coefficient of friction between the shaft and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

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

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

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

4. Seating torque is the amount of torque required to develop the normal forces between the seat and seal to effect a tight closure. Seating torque is dependent on the sealing materials, seal thickness, valve geometry, valve size, differential pressure and leakage requirements. As seen in Fig. 3, as the valve is seated by applying a closing moment $T_1$, the normal forces $R_N$ will increase. Since the seal angle varies around the seal circumference, $R_N$ also varies, thus the point where $R_N$ is a minimum must be loaded sufficiently to effect a seal. Sealing characteristics will be further discussed in the section under Valve Sealing Characteristics (Section 7.0).




$T_1$ = Closing torque applied by actuator

P = Force equivalent to disc pressure loading

$R_N$ = Normal seat reaction force due to torque application

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

DISC WITH CLOSING FORCES APPLIED

FIGURE 3

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

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

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

* Center of Gravity

As seen in the Vought Corp. Test Report (Reference 8.0 B-1) the running torque was approximately 1000 in.lbs. as shown in Fig. 8 Run 1 and Fig. 15 Run 8, with no flow through the valve. This running torque is a combination of bearing, packing, and weight offset torque values. The unseating torque may also be seen, which was approximately 1500 in.-lbs. when a seating torque of approximately 18,000 in.-lbs. was used to close the valve with a 80 psig air supply to the actuator.

2.2 Actuator Design

2.2.1 Geometry

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

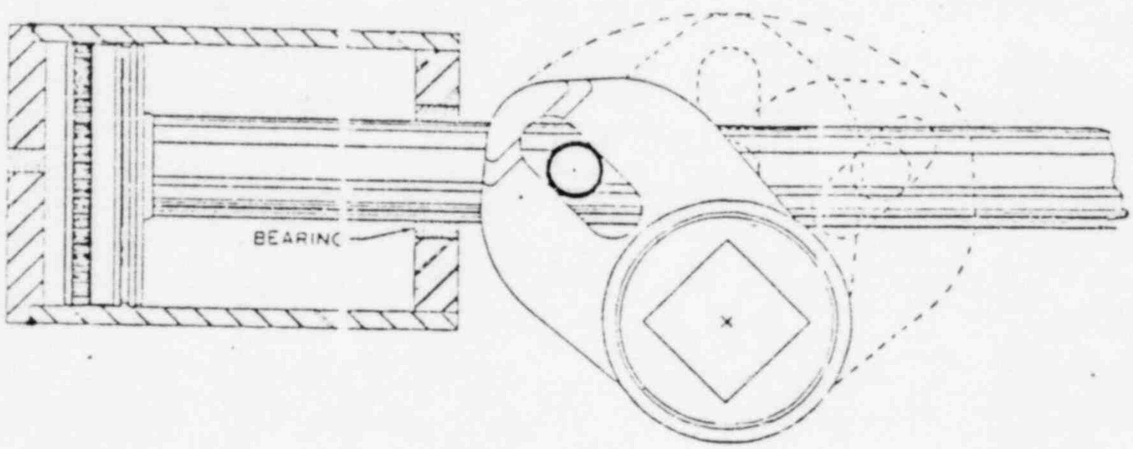



FIGURE 4 - ACTUATOR SCOTCH YOKE DESIGN

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

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

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

TORQUE AT CENTER OF STROKE

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

Where:

T = Torque in in-lb

P = Operating pressure in p.s.i.

MA = Moment arm in inches at center

A = Area of the piston in square inches

TORQUE AT BEGINNING AND END OF STROKE

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

Where:

T = Torque in in-lb

F = Resultant total force in pounds = $\frac{P \times A}{\text{Cos.}\ 45^\circ}$ (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 5.

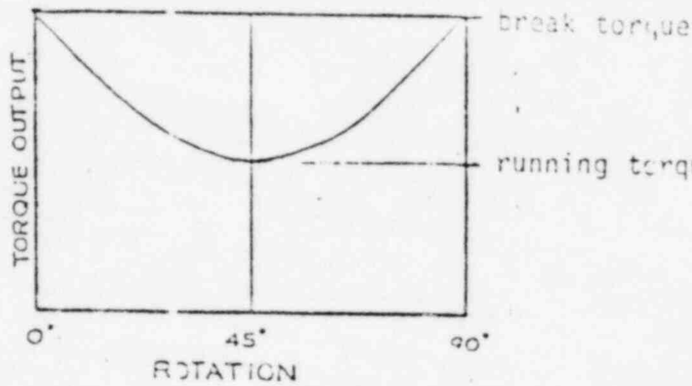



FIGURE 5 - Typical torque output for double acting scotch yoke actuator.

Since thrust is converted to rotary motion, a spring is used opposing the air cylinder to provide a "Fail Safe" actuator. The "Fail Safe" actuator is capable of performing its safety related function in the event of a loss of either the air supply or the control signal to the solenoid valve which controls the air supply to the actuator. The basic construction of the "Fail Safe" actuator is seen here (Figure 6).

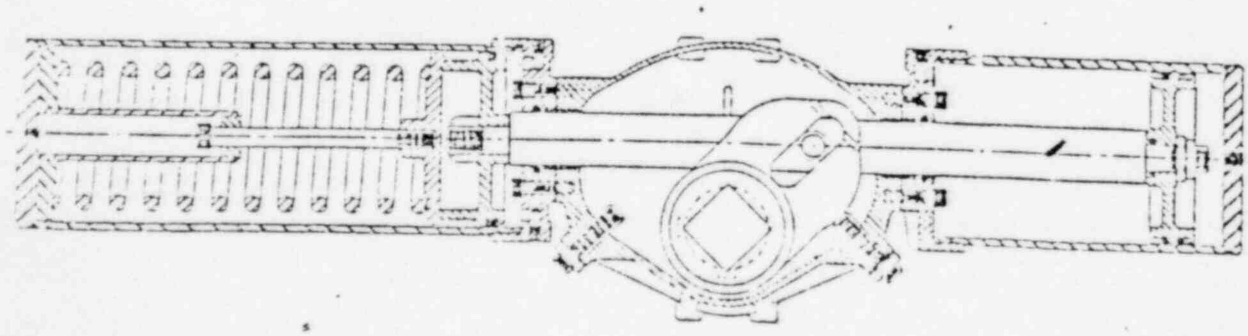


FIGURE 6 - Fail safe, spring return actuator design

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.

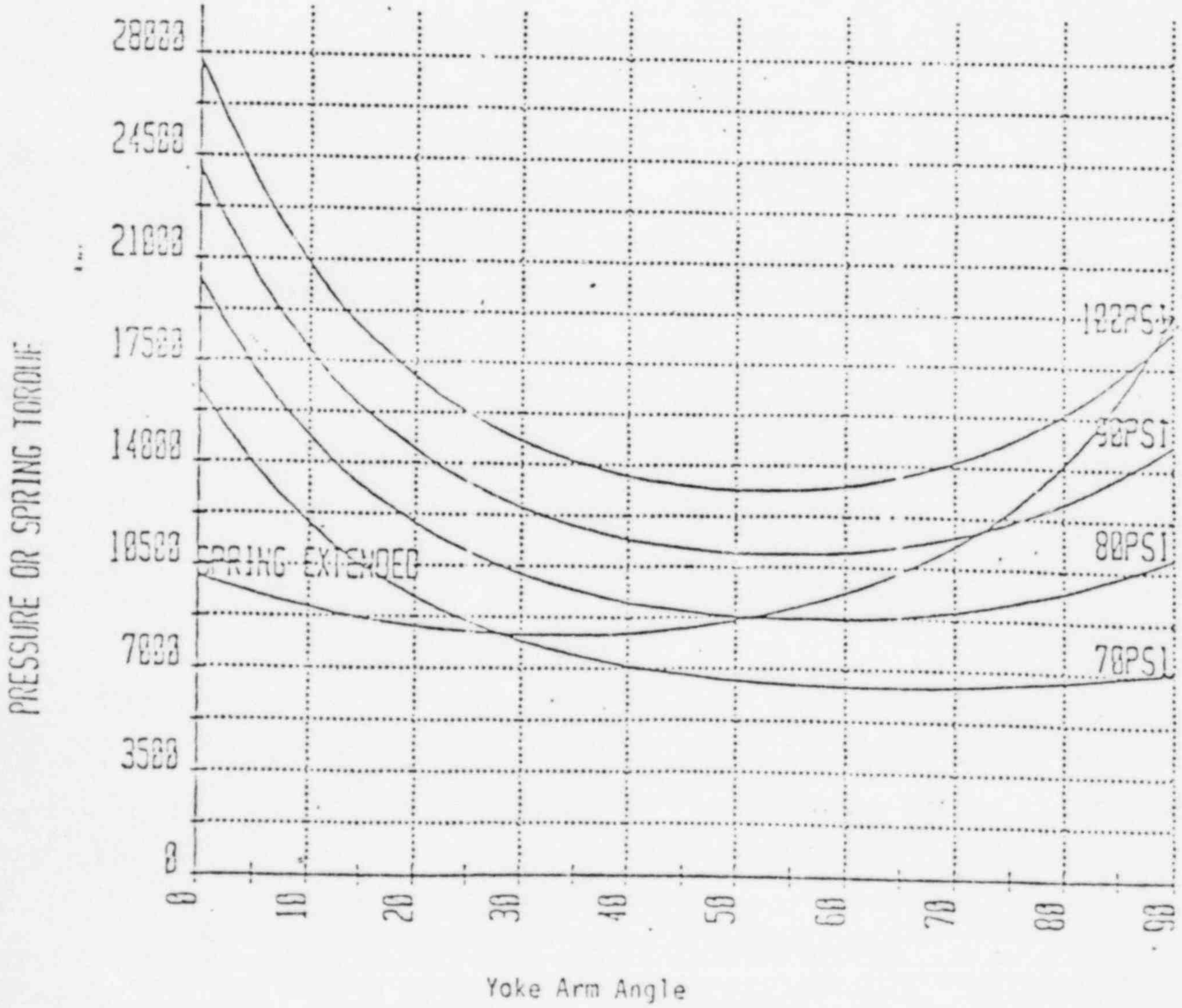



FIGURE 7 - Typical torque output curves for a spring return actuator

2.2.2 Actuator Design Materials

The Bettis actuators used for this job are NT316-SR2-M3 series actuators. These were further specified to be the N version for nuclear service and qualified per IEEE 323, IEEE 344, and IEEE 382. Also, upgraded seismic qualifications are provided based on Patel Report PEI-TR-83-29 with Addendum I and II. These actuators incorporate use of special materials for nuclear service as listed below.

Special Material:

Grease - Dow Corning Molykote 44

Seals - Ethylene Propylene

Internal cylinder coating - Molybdenum disulfide

Yoke pin and rollers - Ryton coated

2.2.3 Actuator and Valve Operation

2.2.3.1 Actuator and Accessories Supplied

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

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

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

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

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

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

*Items which are not required to provide the valve safety function are not IEEE qualified and do not need to be.

TABLE 1

ACTUATOR ACCESSORIES FOR EACH UNIT

| Valve Size (in.) | Equipment Nos. | Clow Job No. | Bettis Actuator Model No. | Fail-safe Rotation (viewed act. end of unit) | Fail-safe Valve Position | Asco Solenoid Valve Model No. | Namco limit switches and lever arm Model Nos. (1 closed position switch) (1 open position switch) | Other Items (each unit) |
|---|---|---|---|---|---|---|---|---|
| I. RECOMMENDED EQUIPMENT | | | | | | | | |
| 12" | IVR006A | 83-2462(N) | NT316-SR2-M3 | CW | Close | NPX831664E | | |
| | IVR006B | " | " | " | " | " | EA180-31302 L.S. | Fisher type 95H regulator |
| | IVR007A | " | " | " | " | " | | |
| | IVR007B | " | " | " | " | " | EA180-32302 L.S. | Rosedale Y6-1/2-40-CI filter |
| | | | | | | | EL010-53337 L.A. | Misc fittings and electrical accessories |
| II. SUPPLIED EQUIPMENT | | | | | | | | |
| 12" | IVR006A | 83-2462(N) | NT316-SR2-M3 | CW | Close | NPL831664E | | |
| | IVR006B | " | " | " | " | " | EA180-31302 L.S. | Fisher type 95H regulator |
| | IVR007A | " | " | " | " | " | | |
| | IVR007B | " | " | " | " | " | EA180-32302 L.S. | Rosedale Y6-1/2-40-CI filter |
| | | | | | | | EL010-53337 L.A. | Misc fittings and electrical accessories |

The torque plots provided in this section represent the calculated output torque of the actuators for the spring and various supply pressure shown. The graphs, which follow, show how the torque output varies for the pressure stroke as a function of supply pressure. It can also be seen that the spring output torque is not a function of supply pressure. The graphs also demonstrate that the output torque (pressure on spring stroke) is a function of yoke position. The graphs provided are based on the numerical data provided in Figure 8.

FIGURE 8



# T316 SR2

CALCULATED TORQUE DATA

## DATA INPUT

| | |
|---|---|
| CYLINDER DIAMETER (in)= | 15.58 |
| CENTER OR TIE BAR DIAMETER (in)= | 0.875 |
| PISTON ROD DIAMETER (in)= | 1.375 |
| NUMBER OF PISTONS = | 1 |
| MOMENT ARM (in)= | 2.812 |
| SPRING LOAD A (lbs)= | 4597 |
| SPRING LOAD B (lbs)= | 7436 |
| BREAK EFFICIENCY (%)= | 70 |
| RUNNING EFFICIENCY (%) = | 85 |
| ENDING EFFICIENCY (%) = | 74 |
| PRESSURES (psi) = | 50 60 70 80 |
| ACTUATOR TYPE,CB=1,HD=2,T,TR=3, = | 3 |

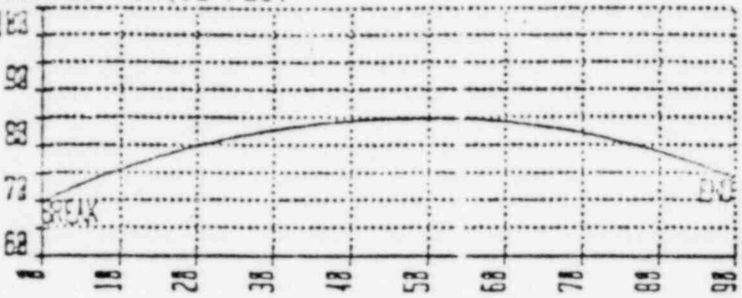
| YOKE ARM ANGLE (degrees) | SPRING TORQUE (in lb) | PRESSURE TORQUE ( 50)psi | PRESSURE TORQUE ( 60)psi | PRESSURE TORQUE ( 70)psi | PRESSURE TORQUE ( 80)psi | EFFICIENCY SPR. % | EFFICIENCY PRES. % |
|---|---|---|---|---|---|---|---|
| 0 | 21045 | 17090 | 24490 | 31889 | 39289 | 74 | 70 |
| 5 | 19047 | 14765 | 21357 | 27949 | 34542 | 77 | 73 |
| 10 | 17513 | 13033 | 19013 | 24994 | 30974 | 79 | 76 |
| 15 | 16338 | 11721 | 17235 | 22749 | 28263 | 81 | 78 |
| 20 | 15450 | 10718 | 15878 | 21037 | 26197 | 82 | 80 |
| 25 | 14798 | 9950 | 14844 | 19739 | 24633 | 83 | 82 |
| 30 | 14348 | 9365 | 14068 | 18772 | 23475 | 84 | 83 |
| 35 | 14077 | 8929 | 13505 | 18081 | 22657 | 85 | 84 |
| 40 | 13973 | 8618 | 13124 | 17629 | 22135 | 85 | 85 |
| 45 | 14031 | 8415 | 12905 | 17394 | 21884 | 85 | 85 |
| 50 | 14254 | 8312 | 12838 | 17364 | 21889 | 85 | 85 |
| 55 | 14653 | 8301 | 12918 | 17536 | 22153 | 84 | 85 |
| 60 | 15249 | 8383 | 13151 | 17919 | 22688 | 83 | 84 |
| 65 | 16072 | 8559 | 13546 | 18533 | 23520 | 82 | 83 |
| 70 | 17173 | 8837 | 14123 | 19409 | 24695 | 80 | 82 |
| 75 | 18619 | 9227 | 14911 | 20596 | 26280 | 78 | 81 |
| 80 | 20514 | 9746 | 15955 | 22163 | 28372 | 76 | 79 |
| 85 | 23011 | 10415 | 17315 | 24216 | 31116 | 73 | 77 |
| 90 | 26347 | 11259 | 19082 | 26904 | 34726 | 70 | 74 |

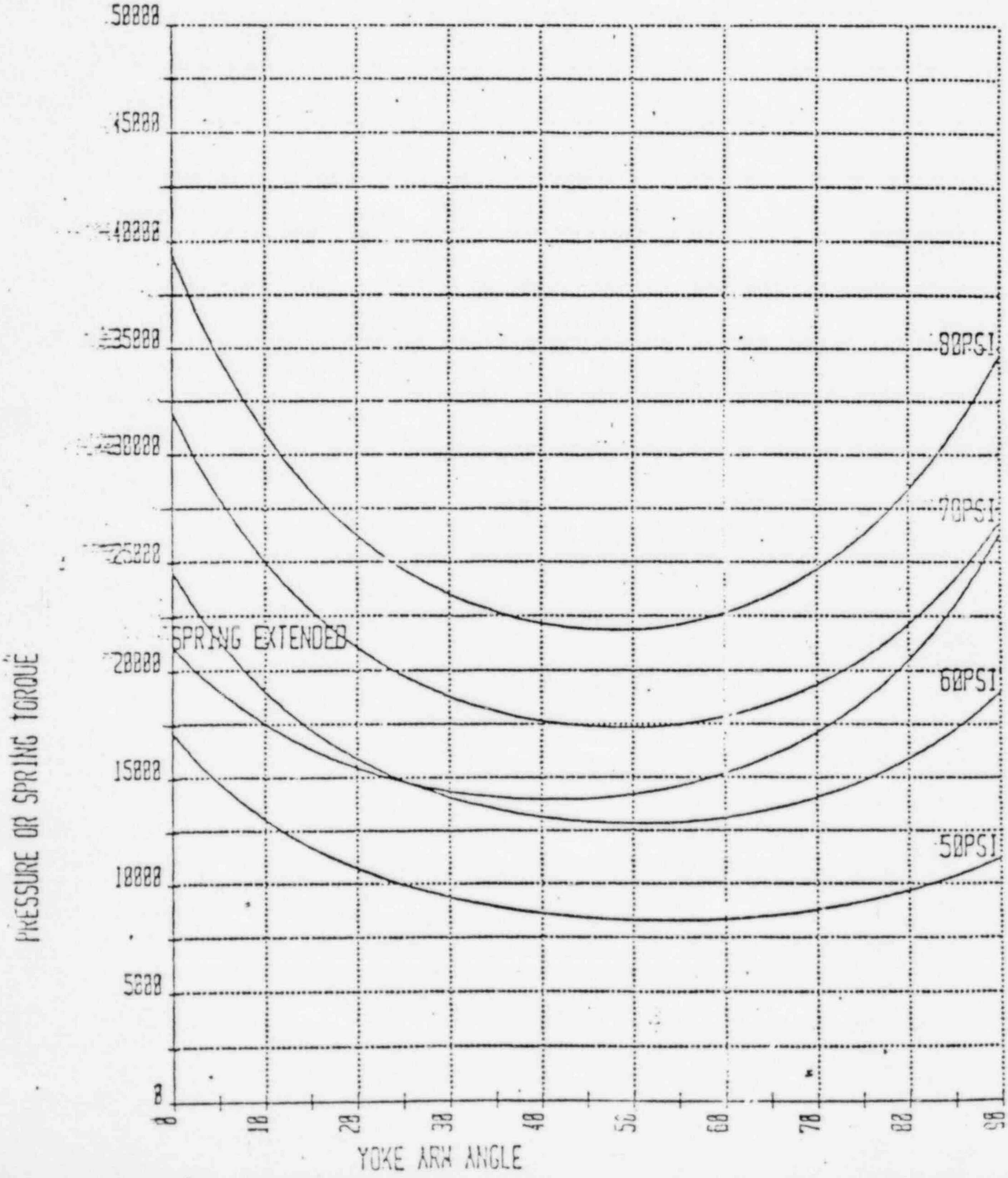
FIGURE 9

CALCULATED TORQUE PLOT

EFFICIENCY PLOT

EFFICIENCY vs ANGLE



2.2.3.2 Operating Time

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

TABLE 2
VALVE BENCH TEST OPERATING TIMES

| Equipment No. of Valve | Valve Size (inch) | Bettis Actuator Model No. | Opening Time Sec. | Closing Time Sec. |
|---|---|---|---|---|
| IVR006A | 12 | NT316-SR2-M3 | 5.0 | 4.2 |
| IVR006B | 12 | " | 5.1 | 4.3 |
| IVR007A | 12 | " | 4.9 | 4.2 |
| IVR007B | 12 | " | 4.9 | 3.9 |

A

## 3.0 VALVE OPERATING AND INSTALLATION REQUIREMENTS

### 3.1 Valve Operating Conditions

The valves were designed to fail close (on loss of power or signal to the solenoid valve) and to allow closure and sealing against a 15 PSI differential applied to the shaft side of the disc during post LOCA flow.

Seismic and other loading conditions for operation are as indicated in the specification. Actuator qualifications for environmental and seismic are covered by previous actuator qualifications supplied by the manufacturer (Bettis) and added seismic tests performed in accord with NTS (National Technical System) Test Plan No. 528-0951 (included in PEI-TR-83-29 Rev. A) for Clow Job No. 82-2053(N) (see References Sect. 8.0 B.2). The Bettis units have been tested as described in PEI-TR-83-29 Rev. A and have demonstrated their ability to function as required.

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

| | | |
|---|---|---|
| Operating conditions - | Normal operating pressure | = 1.0 PSIG |
| | Normal operating temperature | = 122°F |
| | Normal operating flow | = 8000 SCFM |

| | | |
|---|---|---|
| Design conditions- | Max. operating pressure body only | = 285 PSIG @ 100°F |
| | Max. pressure differential disc | = 15 PSID |
| | Max. temperature | = 185°F |
| | Required Torque to seat | = 18,200 in-lb |
| | Failure mode | = Fail close |
| | Allowed leakage | = .4 cc/min air @15 PSID |

### 3.2 Valve Installation Configurations

In addition to the pressure and flow conditions specified in 3.0, the valve performance is affected by the as installed orientation. Upstream and downstream, tees, elbows, reducers, and other valves can affect the aerodynamic torque characteristics of butterfly valves. These effects are discussed in Section 5.0. The installed configurations for the subject valves, as derived from Sargent & Lundy Drawings M06-1111, Rev E are summarized in Figures 10 and 11.

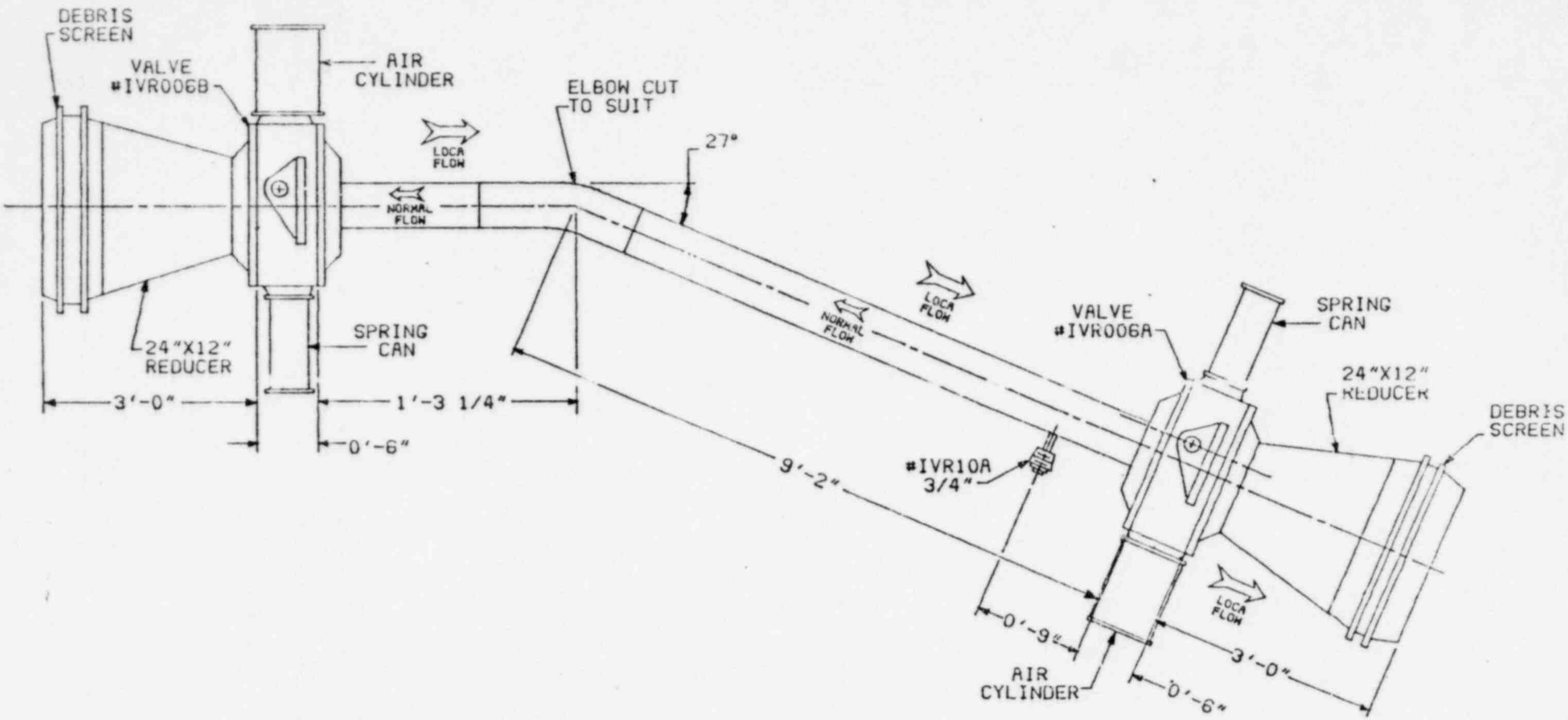



FIGURE 10 - PLAN "R"

CONTINUOUS PURGE HVAC SUCTION LINE

REF. SARGENT&LUNDY DWG. #M06-1111
REF. CLOW JOB # 83-2462-01(N)

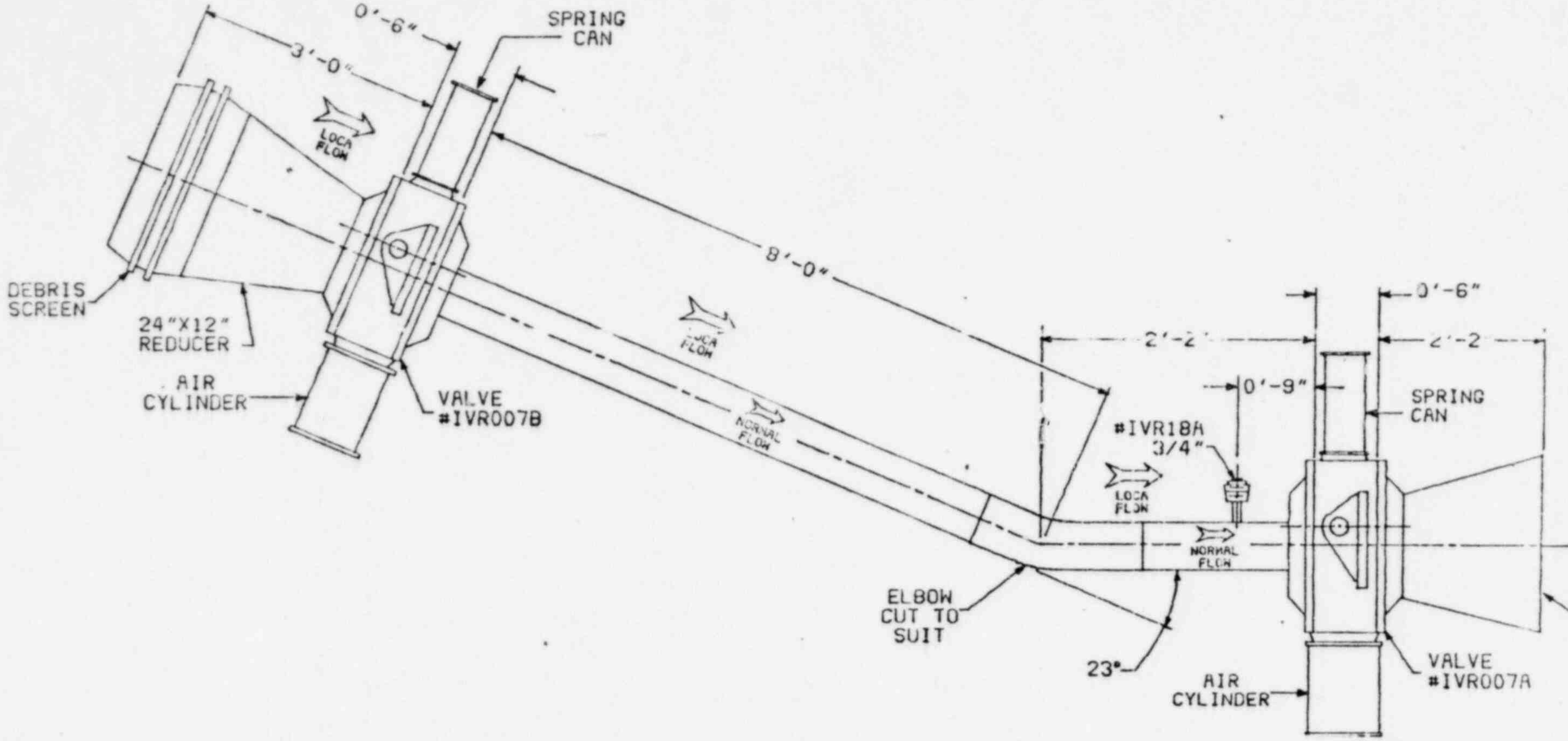



FIGURE 11 - PLAN "S"

CONTINUOUS PURGE HVAC EXHAUST LINE

REF. SARGENT&LUNDY DWG. #M06-1111
REF. CLOW JOB # 83-2462-01(N)

## 4.0 VALVE STRUCTURAL INTEGRITY UNDER SEISMIC AND OPERATIONAL LOADINGS

Operability of the subject valves has been demonstrated by a combination of testing and analysis in accord with the design specification. Separate reports have been prepared and provided to demonstrate suitability of valve components and the assembly. A listing is provided in the References 8.0 at the end of the report. This section summarizes the results of such tests and analyses in meeting the conditions as presented in Section 3.0.

### 4.1 VALVE STRESS ANALYSIS

Valve stress analysis was performed by Patel Engineers, Huntsville, Alabama for the subject valves. The analysis was made using the ANSYS finite element computer program developed by Swanson Analysis System, Inc., Houston, PA. This public domain program has had a sufficient history of use to justify its applicability and validity. The analysis performed compares the nuclear specific requirements of the Clinton Valves, Report PEI-TR-852400-1 to an already performed worst case generic qualification report for a Clow 12" Wafer Stop Valve, PEI-TR-833600-1 Rev.A. The comparison of key elements in these reports is shown in Tables 3 and 4.

TABLE 3

COMPARISON OF CLINTON NUCLEAR SPECIFIC REQUIREMENTS

TO

GENERIC NUCLEAR QUALIFICATION DATA (REFERENCE 8A)

| LOADINGS | GENERIC | CLINTON |
|---|---|---|
| Pressure | | |
| Shell (psig) | 285 | 285 |
| Seat (psid) | 175 | 15 |
| Torque (in-lb) | 39,275 | 18,200 |
| Seismic Acceleration | | |
| NS (g) | 10.9 | 4.5 |
| EW (g) | 10.9 | 4.5 |
| Vertical (g) | 10.9 | 3.0 |
| OPERATOR | | |
| Weight (lb) | 700 | 638 |
| Center of Gravity | | |
| X (in) | 15 | 4.5 |
| Y (in) | 15 | 5.5 |
| Z (in) | 10 | 5.5 |
| FREQUENCY $f_o$ (Hz) | 48.8 Hz | $f_o \geq$ 50 Hz |
| EVALUATION AGAINST | ASME Section III Design and Level A | ASME Section III Design and Level A |

Table 4

Summary of Allowable Stresses

| LOCATION | MATERIAL | ALLOWABLE STRESS (psi) (PER ASME SECTION III, TABLES I-7.1 THROUGH I-7.3) | STRESS VALUE (psi) | REPORT IN WHICH ITEM IS ANALYZED** /SEISMIC LOAD LEVEL | STRESS RATIO |
|---|---|---|---|---|---|
| Valve Body | SA 516 GR.70 | 17500 | 8575 | Generic 10.9 g | .49 |
| Disc | SA 516 GR.70 | 17500 | 9275 | Generic 10.9 g | .53 |
| Drive Shaft | SA 564 Type 630 H-1100 | 34550 | 29395 | Generic 10.9 g | .85 |
| Operator Adapter Plate | A 516 GR.70 | 17500 | $\sigma_\eta$ = 1893 | Clinton 4.5 g | .11 |
| Adapter Plate Bolts | A 193 GR.B7 | 25000 (Axial = 62500 Shear = 25833) | $\sigma_\eta$ = 13958 $\tau$ = 9231 | Clinton 4.5 g | .36 * |
| Operator/Adapter Bolts | A 193 GR.B7 | 25000 (Axial = 62500 Shear = 25833) | $\sigma_\eta$ = 13471 $\tau$ = 10593 | Clinton 4.5 g | .41* |
| Cover Plate | SA 516 GR. 70 | 17500 | 9356 | Generic 10.9 g | .54 |
| Cover Plate Bolts | SA 193 GR.B7 | 25000 (Axial = 62500 Shear = 25833) | $\sigma_\eta$ = 6370 $\tau$ = 60 | Generic 10.9 g | .10* |

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

** Generic Report is Patel PEI-TR-833600-1. Clinton Report is Patel PEI-TR-852400-1 (A)

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

The conclusion that can be drawn is that the structural integrity of the subject valve assemblies fully meets the requirements of Baldwin Design Specification BA-K-2882-29, Issue 2 and ASME Section III, 1980 Edition thru Summer 82.

4.2 Actuator Tests

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

| | | |
|---|---|---|
| Unit 1 | NT-820-SR4-S | Spring ending torque = 33,098 in lb<br>Pressure torque+ = 178,160 in lb |
| Unit 2 | NT-312-SR5 | Spring ending torque = 5,810 in lb<br>Pressure torque+ = 31,253 in lb |

+ at 80 PSIG pressure to air cylinder

The units were both spring return fail closed units and were representative of the line of actuators which would be used on containment purge and vent valves. The units were tested for baseline performance, subjected to OBE and SSE leve s in accord with the procedure and specification, and then were operationally tested. The units proved to be operable both during and after the required tests and upon inspection showed no signs of noticeable wear. Successful operation of these units in combination with previous generic qualification for environmental conditions generically qualifies the NT316-SR2-M3 units used on this job. (Note a report is provided justifying similarity, see PEI-TR-8522201-02 and Section 8.0)

## 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 4 ) were designed and built of 150 lb class Tricentric valves of standard design. Scale models of a 12, 24, 48, and 96 inch valve were constructed and tested using University of Illinois facilities under the direction of A.L. Addy, Ph. D. (Engineering Consultant in Fluid Dynamics and Engineering and Associate Head, Department of Mechanical and Industrial Engineering, U. of I. at Urbana, Champaign, Ill.). The tests were made with air in accord with ISA standards for a straight pipe run flow test. The tests were run at various pressure ratios (upstream to downstream pressure) in both the choked and non-choked pressure regimes. Very low pressure ratios were also applied to allow correlation to incompressible (liquid) flow in accord with ISA standards. Tests were made with flow in the normal direction for Tricentrics (shaft upstream) and for reverse flow (shaft downstream). Further, several pressure ratios near the choked flow point were applied to determine the point of choking. This test pointed out that the standard rule of thumb (downstream pressure/upstream pressure = .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

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 $80^{0}$ 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 $90^{0}$ (¼ turn) valve design. Therefore, at small opening angles ($0^{0}$ to $20^{0}$) 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 of the subject valves.

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

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

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

where

$T$ = predicted aerodynamic torque (in lb)

$C_T$ = torque coefficient developed in model tests

$\Delta P$ = 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 4 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 5 shows a comparison between the provided size valves and sizes interpolated from test valves.

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

TABLE 4

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 5

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

| VALVE | | |
|---|---|---|
| ELEMENTS | 12" | |
| | Size | Ratio |
| *I.D. | 11.938 | 1.00 |
| *$A_2$ | 11.145 | 1.02 |
| *$K_2$ | 10.876 | .99 |
| Shaft Dia. | 2.000 | 1.13 |
| Shaft $C_L$ to Seal $C_L$,L | 1.875 | 1.07 |
| *Disc Thickness | 1.50 | 1.00 |
| *Shaft Offset E | 1.312 | .95 |
| Shaft Offset LC | 1.364 | NA |
| Ear Width | 2.00 | NA |
| Ear Height | 8.58 | NA |

*Elements considered important to torque characteristics

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

$A_2$ = Major axis of elliptical seal

$K_2$ = Minor axis of elliptical seal

E = Offset between shaft axis and disc center (see Figure 2)

LC = Offset between shaft axis and pipe run centerline

All dimensions in inches

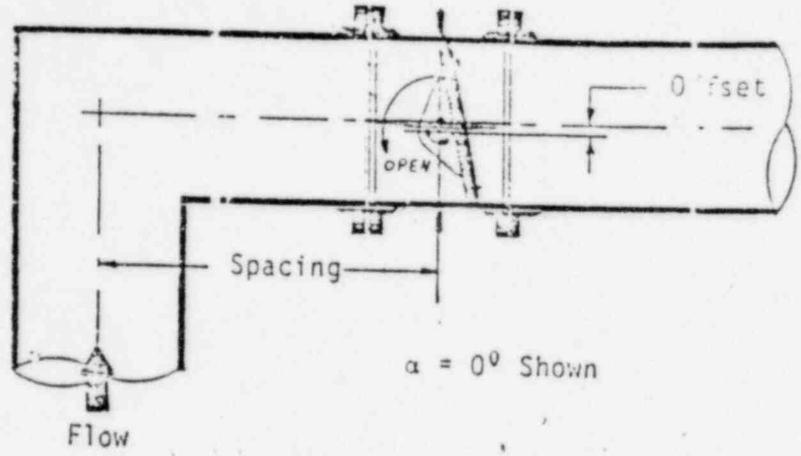
### 5.1.2 Tests With An Upstream Elbow

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

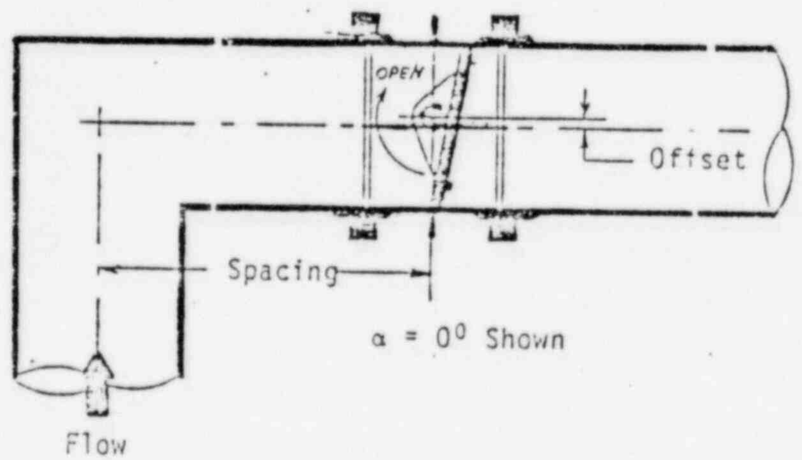
The mitered elbow produces the greatest separated flow region at the inside of the turn and biases the flow to the outside corner to a maximum. Flow around the corner produces a lower local pressure around the inside of the turn and higher local pressure to the outside. This will oppose closure for geometry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

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

Geometry 1




Geometry 2




Geometry 3

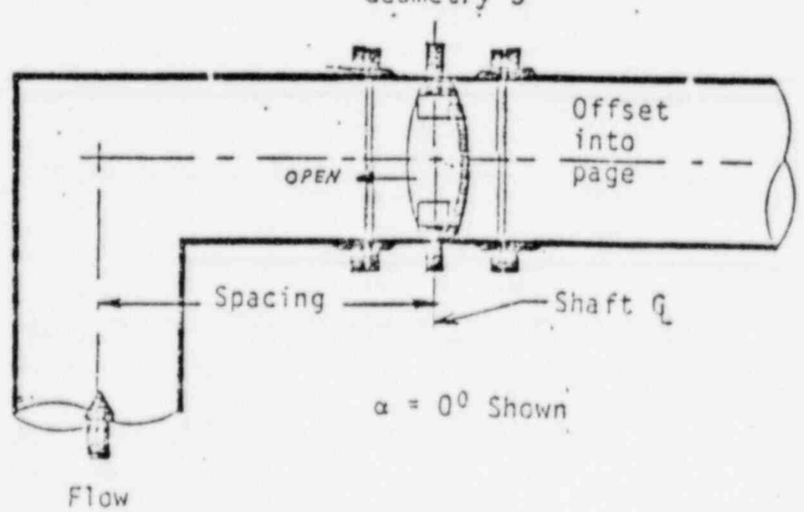



FIGURE 12- Valve Orientations Relative to Upstream Elbow

### 5.1.3 Downstream Piping Effects

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

5.2 Model Data Verification

A test of a full size 12" valve was run at Vought's High Speed Wind Tunnel in Dallas, Texas (see reference 8.0B1) to demonstrate operability and substantiate model test data. The tests demonstrated the valve would operate in the required 5 second period. It further showed that torque values were less than predicted from model data. The valve used for the test incorporated a one piece thru shaft design while the model had a two piece shaft. To verify the torque effect due to this change, another test was made (data not put into a formal report form) in which a 2 piece shaft was installed in place of the thru shaft. The test was made with the disc held in a stationary position by a manual worm gear type actuator. The result was that the peak torque was the same for both the one and two piece shaft design. The only difference was that the two piece shaft design showed a peak torque closer (by 5 to 10 degrees) to the full open position. A test was also run with the one piece shaft design with the disc held in a stationary position. This was done to provide direct correlation with the model tests which were done in this manner. It also allowed a comparison to the torques measured during the dynamic test with the shaft connected to the pneumatic actuator. A summary of the operability test is included in Appendix B.

5.3 Application of Model Aerodynamic Test To Full Size Valve Operability

5.3.1 Valve Operating Times Expected In Service

All valves are designed to close within 5 seconds for flow conditions produced by the maximum differential pressure of 15 PSIG when 100 PSIG is released from the actuator air cylinder. The valves will close under these installed conditions due to the fact that the operator output torque (spring torque) and the valve aerodynamic torque are tending to close the valve at all disc angles for LOCA conditions. (See Table 6 and Tables 9,10,11.) While not required for LOCA, to open the valve under the above conditions, 3104 in-lb of torque is required to crack the disc off the seat, and 4606 in-lb max is required to hold the valve disc open. (See Table 6 ) The air torque of the actuator (valve open direction) is rated at 20,200 in-lb @ 80 PSIG, and therefore is more than adequate for the required worst case operating conditions.

In the Vought Test (Reference 8.0 B1 ) closing times were shown to improve slightly with flow through the valve. The conduct of the test would suggest that opening times in actual service might be retarded about .3 to .5 seconds and closing times might be improved by the same amount under maximum differential pressure conditions relative to the Clow bench test data.

5.3.2 AERODYNAMIC TORQUES 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 6 thru 11 present torque and other data for the subject valves at various operating conditions. The item of concern for valve operability is TQA (for normal operating conditions, open cycle) and TQA (for maximum operating conditions, closing cycle). The positive torque values tend to close the valve. The meanings of the other listings can be found in 8.0 References.

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 the subject job there are four identical valves of which one is installed more than 8 diameters downstream of an elbow. Another valve is installed approximately two pipe diameters downstream from an elbow, and the remaining two valves are installed with debris screening and 24 x 12 concentric reducers on their inlet sides. (See Figures 10 and 11). All of the valves are installed with the shaft side facing LOCA flow, thus LOCA flow will act to close the valves.

The valve which is more than 8 pipe diameters downstream from an elbow can obviously be modeled using test data for straight pipeline flow since torque deviation from the straight flow situation is negligible when the as installed valves are greater than 8 diameters from an upstream elbow. For the valve that is installed approximately 2 pipe diameters from an elbow, use of the test data for mitered elbow installed 2 diameters upstream for determining installed operability is considered

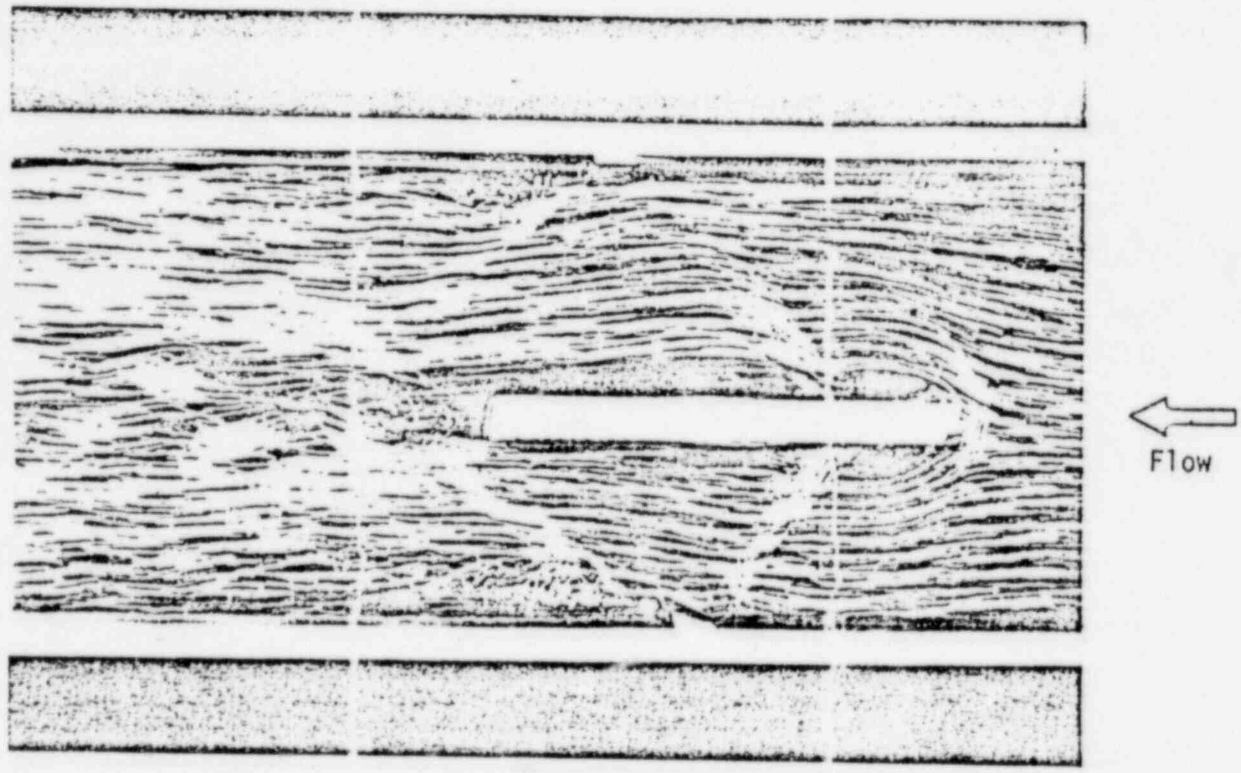
conservative. The 90° mitered elbow used in the model tests is a worst case condition compared to radius type elbows which are typically used for in-plant installation. The elbow angle in this case is 23°, further making a 90° mitered elbow assumption conservative. If torque operating margins are adequate, these judgements are justified

For the remaining two valves with debris screens and 24 x 12 reducers on their inlet sides, using trends from straight pipe flow test data would be considered conservative. During a LOCA the flow influxes into the screen, and as the flow passes through the screen and towards the shaft side of the valve disc, it will be turbulent and not fully developed as the flow would be in the case of a straight pipeline. The reducer would tend to converge the turbulent flow somewhat, but it will still not be fully developed due to the short length of the reducer. Thus, the aerodynamic torques will be less than predicted because the flow turbulance would create a more evenly distributed pressure over the area of the valve disc. Further, the screen would act as a flow restriction and something less than full LOCA pressure would actually be seen by the valve disc. The net result is that with the screen and reducer installed, the tendency for the flow to close the valve will be diminished as compared to the straight pipeline flow case. Experiments in which a restriction, namely another valve, placed upstream at a close distance from the first valve has shown the torque coefficients of the downstream valve to be less than would be expected for single valve in straight pipeline flow given the same pressure ratio (Reference C 2).

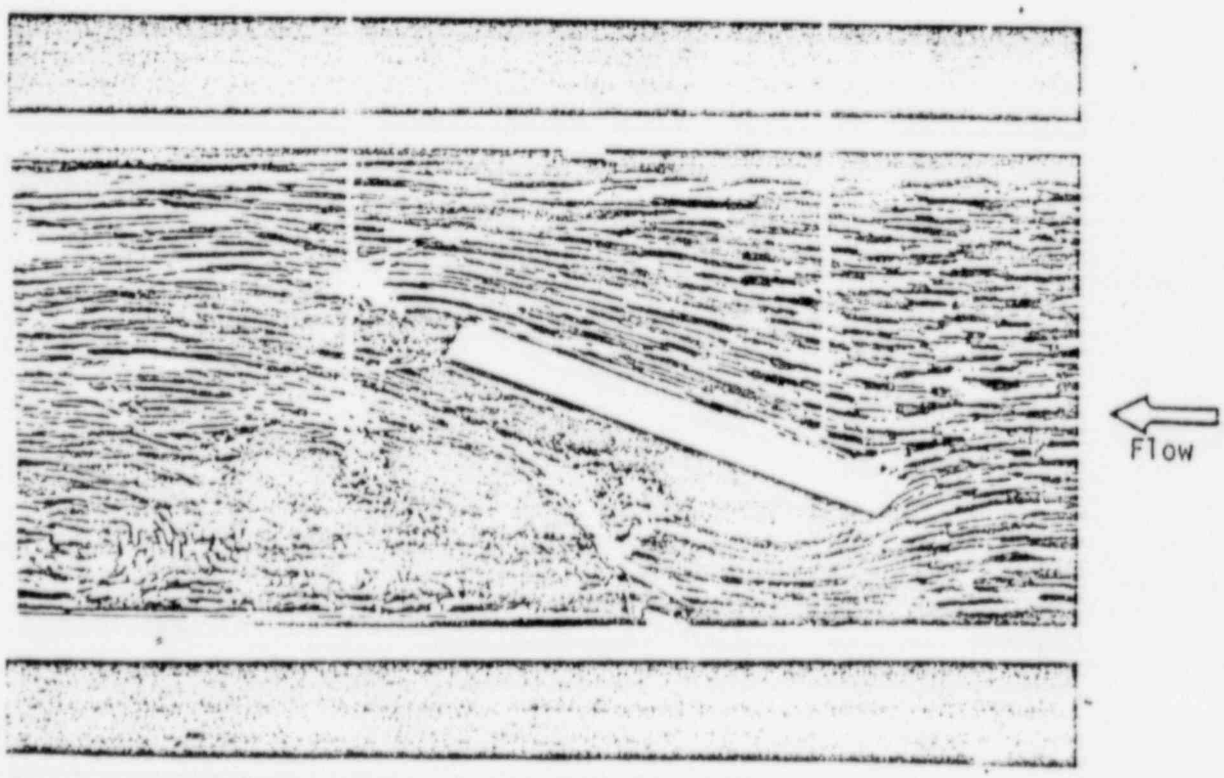
Therefore, the two valves with screens and reducers modeled as single valves in straight pipeline flow will yield the largest aerodynamic torque coefficients for analysis purposes, and would be considered conservative.

On two of the valves there is a 3/4" pipe pressure tap approximately one foot from the valve inlet. This tap is normally closed and should be of no consequence to either the valve flow or the torque during a LOCA.

The torque values for all four valves are listed in Tables 9 thru 11.

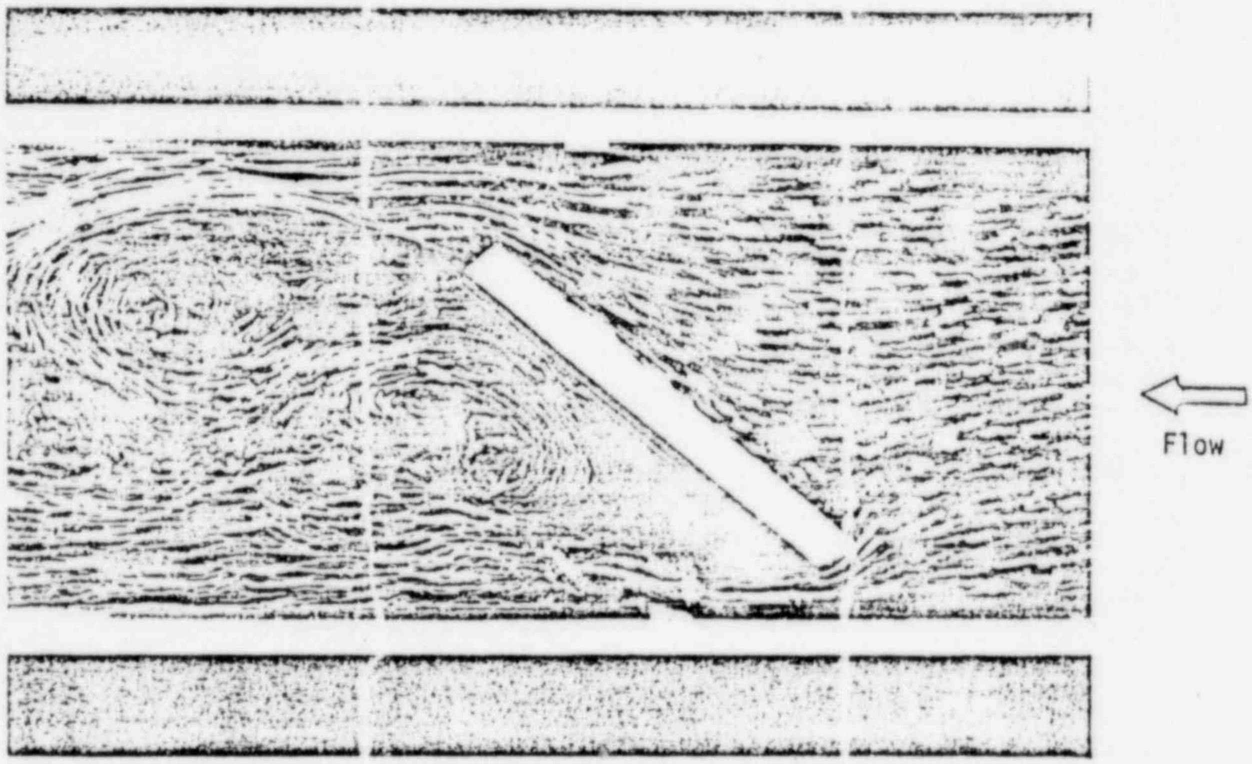



α = 80°

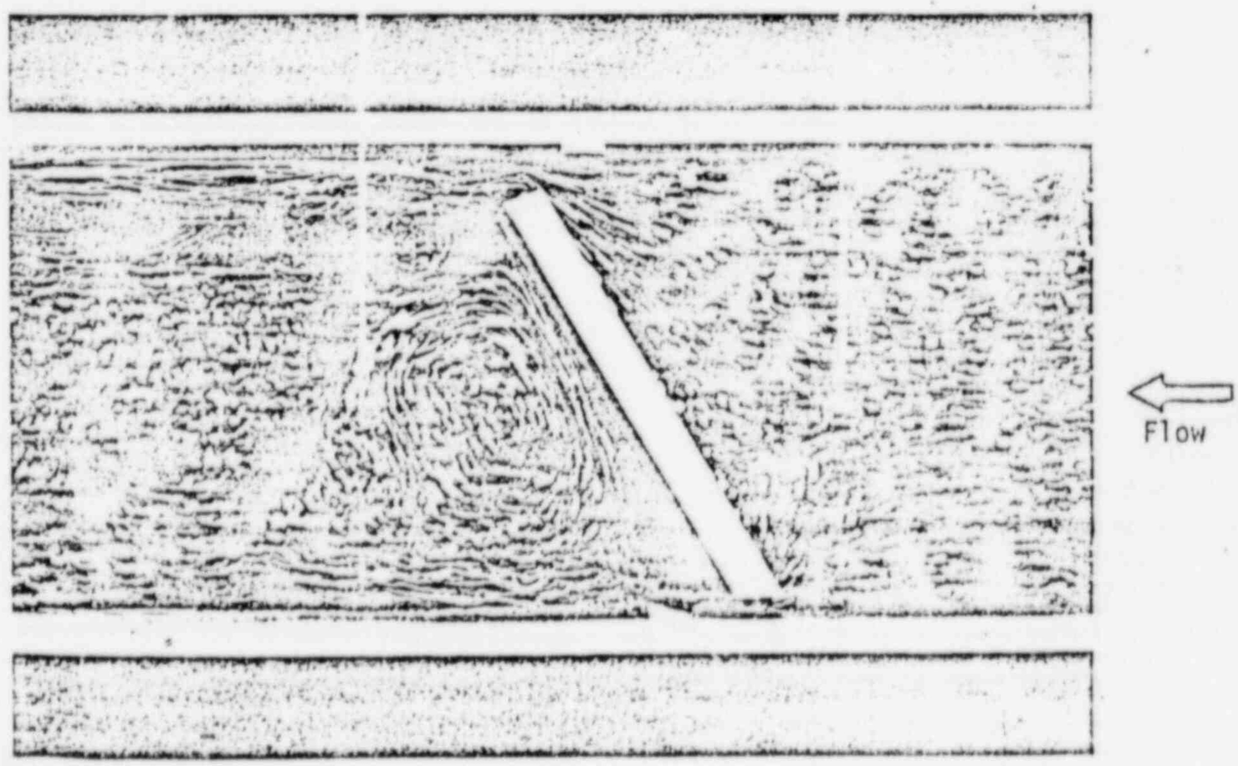



α = 60°

Figure 13 Close-up view of single test valve experiments with a back pressure ratio of 0.45.




α = 40°




α = 20°

Figure 14 Close-up view of single test valve experiments with a back pressure ratio of 0.45.

# DEFINITION OF TERMS USED IN TABLES 6, 7, & 8

| | |
|---|---|
| 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 ( $\Delta$P) 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. |
| YCV | - Flow coefficient |

TABLE 6

EMERGENCY FLOW, MAX CONTAINMENT PRESSURE
STRAIGHT PIPE RUN

CASE: BALDWIN/ILLPBOWER
DATE: 03-04-85
PATM: 14.70(PSIA)
PSU = 29.70(PSIA) TSU = 644.67(R)
MEDIUM: GAS = A GAMMA = 1.40
FLOW = CF OPTION = 2
DV = 12.000(IN)

UNITS SYSTEM: ES
SHAFT: US

MW = 29.0

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OUTPUT DATA
-----------

CHOKING PRESSURE RATIOS: PSC/POU = .752 DPS/PSU = .186
SOLUTION: W80 = 41.01(LBM/S)

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

PSC/POU = .7523

(A)

| ALPHA | CF | WR | DPS/PSU | PSU/POU | PSC/POU | POD/POU | TQR1 |
|---|---|---|---|---|---|---|---|
| 80.0 | .5447 | 1.0000 | .1861 | .9243 | .7523 | .8701 | .0676 |
| 75.0 | .5345 | .9814 | .1888 | .9274 | .7509 | .8592 | .1115 |
| 70.0 | .5144 | .9444 | .1939 | .9332 | .7477 | .8473 | .1415 |
| 65.0 | .4858 | .8918 | .2005 | .9409 | .7427 | .8347 | .1600 |
| 60.0 | .4501 | .8264 | .2080 | .9498 | .7351 | .8219 | .1693 |
| 55.0 | .4090 | .7510 | .2155 | .9590 | .7248 | .8094 | .1713 |
| 50.0 | .3639 | .6682 | .2227 | .9678 | .7117 | .7976 | .1679 |
| 45.0 | .3164 | .5808 | .2291 | .9759 | .6957 | .7870 | .1608 |
| 40.0 | .2678 | .4917 | .2346 | .9829 | .6772 | .7776 | .1515 |
| 35.0 | .2198 | .4036 | .2390 | .9885 | .6566 | .7698 | .1412 |
| 30.0 | .1739 | .3193 | .2423 | .9929 | .6347 | .7634 | .1311 |
| 25.0 | .1315 | .2414 | .2446 | .9959 | .6124 | .7587 | .1221 |
| 20.0 | .0942 | .1729 | .2461 | .9979 | .5909 | .7554 | .1148 |
| 15.0 | .0634 | .1164 | .2470 | .9991 | .5719 | .7536 | .1099 |
| 10.0 | .0407 | .0747 | .2474 | .9996 | .5570 | .7529 | .1078 |
| 5.0 | .0275 | .0505 | .2476 | .9998 | .5482 | .7532 | .1087 |

| ALPHA (DEG) | YCV (...) | W (LBM/HR) | TQ (IN-LBF) | TQA (IN-LBF) |
|---|---|---|---|---|
| 80.0 | 2847.28 | 147645.37 | 645.64 | 1752.31 |
| 75.0 | 2765.31 | 144898.00 | 1083.62 | 2899.32 |
| 70.0 | 2609.77 | 139439.00 | 1420.95 | 3702.61 |
| 65.0 | 2403.30 | 131676.87 | 1675.82 | 4221.93 |
| 60.0 | 2166.39 | 122019.69 | 1855.89 | 4508.02 |
| 55.0 | 1915.11 | 110875.47 | 1965.31 | 4605.94 |
| 50.0 | 1660.92 | 98652.69 | 2009.44 | 4557.39 |
| 45.0 | 1411.67 | 85759.59 | 1996.84 | 4401.76 |
| 40.0 | 1172.79 | 72604.19 | 1939.62 | 4176.22 |
| 35.0 | 948.32 | 59594.81 | 1852.44 | 3915.37 |
| 30.0 | 741.72 | 47139.62 | 1751.08 | 3650.46 |
| 25.0 | 556.48 | 35646.77 | 1650.89 | 3408.72 |
| 20.0 | 396.45 | 25524.37 | 1565.54 | 3212.70 |
| 15.0 | 266.08 | 17180.45 | 1506.05 | 3079.87 |
| 10.0 | 170.48 | 11023.04 | 1480.48 | 3022.45 |
| 5.0 | 115.31 | 7459.96 | 1493.64 | 3047.38 |

See Reference 8.0 C-1 and Page 59A definition of terms

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

(A)

TABLE 7

NORMAL FLOW CHARACTERISTICS
STRAIGHT PIPE RUN (SHAFT UPSTREAM)

CASE: BALDWIN/ILL POWER
DATE: 03-04-85
PATM: 14.70(PSIA)
PSU = 15.70(PSIA)
MEDIUM: GAS = A
FLOW = CF
DV = 12.000(IN)

TSU = 581.67(R)
GAMMA = 1.40
OPTION = 2

UNITS SYSTEM: ES
SHAFT: US

MW = 29.0

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OUTPUT DATA
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CHOKING PRESSURE RATIOS: PSC/POU = .752 DPS/PSU = .186
SOLUTION: W80 = 22.82(LBM/S)

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

PSC/POU = .7523

A

| ALPHA | CF | WR | DPS/PSU | PSU/POU | PSC/POU | POD/POU | TQR1 |
|---|---|---|---|---|---|---|---|
| 80.0 | .5447 | 1.0000 | .1861 | .9243 | .7523 | .8701 | .0676 |
| 75.0 | .5345 | .9814 | .1888 | .9274 | .7509 | .8592 | .1115 |
| 70.0 | .5144 | .9444 | .1939 | .9332 | .7477 | .8473 | .1415 |
| 65.0 | .4858 | .8918 | .2005 | .9409 | .7427 | .8347 | .1600 |
| 60.0 | .4501 | .8264 | .2080 | .9498 | .7351 | .8219 | .1693 |
| 55.0 | .4090 | .7510 | .2155 | .9590 | .7248 | .8094 | .1713 |
| 50.0 | .3639 | .6682 | .2227 | .9678 | 7117 | .7976 | .1679 |
| 45.0 | .3164 | .5808 | .2291 | .9759 | .6957 | .7870 | .1608 |
| 40.0 | .2678 | .4917 | .2346 | .9829 | .6772 | .7776 | .1515 |
| 35.0 | .2198 | .4036 | .2390 | .9885 | .6566 | .7698 | .1412 |
| 30.0 | .1739 | .3193 | .2423 | .9929 | .6347 | .7634 | .1311 |
| 25.0 | .1315 | .2414 | .2446 | .9959 | .6124 | .7587 | .1221 |
| 20.0 | .0942 | .1729 | .2461 | .9979 | .5909 | .7554 | .1148 |
| 15.0 | .0634 | .1164 | .2470 | .9991 | .5719 | .7536 | .1099 |
| 10.0 | .0407 | .0747 | .2474 | .9996 | .5570 | .7529 | .1078 |
| 5.0 | .0275 | .0505 | .2476 | .9998 | .5482 | .7532 | .1087 |

| ALPHA (DEG) | YCV (...) | W (LBM/HR) | TQ (IN-LBF) | TQA (IN-LBF) |
|---|---|---|---|---|
| 80.0 | 2847.28 | 82156.12 | 341.26 | 116.82 |
| 75.0 | 2765.31 | 80627.34 | 572.75 | 193.29 |
| 70.0 | 2609.77 | 77589.78 | 751.05 | 246.84 |
| 65.0 | 2403.30 | 73270.56 | 885.76 | 281.46 |
| 60.0 | 2166.39 | 67896.91 | 980.94 | 300.53 |
| 55.0 | 1915.11 | 61695.83 | 1038.77 | 307.06 |
| 50.0 | 1660.91 | 54894.52 | 1062.10 | 303.83 |
| 45.0 | 1411.67 | 47720.25 | 1055.44 | 293.45 |
| 40.0 | 1172.79 | 40400.02 | 1025.19 | 278.41 |
| 35.0 | 948.32 | 33161.06 | 979.11 | 261.02 |
| 30.0 | 741.72 | 26230.48 | 925.54 | 243.36 |
| 25.0 | 556.48 | 19835.36 | 872.59 | 227.25 |
| 20.0 | 396.45 | 14202.83 | 827.47 | 214.18 |
| 15.0 | 266.08 | 9559.93 | 796.03 | 205.32 |
| 10.0 | 170.48 | 6133.69 | 782.51 | 201.50 |
| 5.0 | 115.31 | 4151.03 | 789.47 | 203.16 |

See Reference 8.0 C-1 and Page 59A for definition of terms

A

--------COMPUTATIONS COMPLETE

TABLE 8

NORMAL FLOW CHARACTERISTICS,
STRAIGHT PIPE RUN (SHAFT DOWN STREAM)

CASE: BALDWIN/ILL POWER
DATE: 03-04-85
PATM: 14.70(PSIA)
PSU = 15.70(PSIA) TSU = 581.67(R)
MEDIUM: GAS = A GAMMA = 1.40
FLOW = CF OPTION = 2
DV = 12.000(IN)

UNITS SYSTEM: ES
SHAFT: DS

MW = 29.0

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OUTPUT DATA
-----------

CHOKING PRESSURE RATIOS: PSC/POU = .756 DPS/PSU = .175
SOLUTION: W80 = 24.20(LBM/S)

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

PSC/POU = .7556 (A)

| ALPHA | CF | WR | DPS/PSU | PSU/POU | PSC/POU | POD/POU | TQR1 |
|---|---|---|---|---|---|---|---|
| 80.0 | .5728 | 1.0000 | .1746 | .9154 | .7556 | .8904 | -.0240 |
| 75.0 | .5695 | .9942 | .1755 | .9165 | .7553 | .8779 | -.0360 |
| 70.0 | .5531 | .9655 | .1802 | .9217 | .7534 | .8641 | -.0457 |
| 65.0 | .5255 | .9174 | .1875 | .9300 | .7495 | .8493 | -.0533 |
| 60.0 | .4887 | .8531 | .1963 | .9402 | .7432 | .8342 | -.0590 |
| 55.0 | .4447 | .7763 | .2055 | .9511 | .7338 | .8191 | -.0630 |
| 50.0 | .3954 | .6903 | .2144 | .9618 | .7210 | .8045 | -.0657 |
| 45.0 | .3429 | .5987 | .2223 | .9716 | .7049 | .7906 | -.0671 |
| 40.0 | .2891 | .5047 | .2289 | .9800 | .6856 | .7776 | -.0675 |
| 35.0 | .2359 | .4119 | .2342 | .9868 | .6638 | .7656 | -.0672 |
| 30.0 | .1853 | .3235 | .2382 | .9919 | .6403 | .7546 | -.0664 |
| 25.0 | .1392 | .2430 | .2409 | .9954 | .6165 | .7446 | -.0654 |
| 20.0 | .0993 | .1734 | .2426 | .9977 | .5940 | .7355 | -.0643 |
| 15.0 | .0675 | .1178 | .2436 | .9989 | .5745 | .7270 | -.0635 |
| 10.0 | .0454 | .0793 | .2440 | .9995 | .5602 | .7192 | -.0631 |
| 5.0 | .0347 | .0605 | .2442 | .9997 | .5530 | .7119 | -.0634 |

| ALPHA (DEG) | YCV (...) | W (LBM/HR) | TQ (IN-LBF) | TQA (IN-LBF) |
|---|---|---|---|---|
| 80.0 | 3121.81 | 87117.19 | -113.49 | -41.42 |
| 75.0 | 3091.53 | 86614.03 | -171.50 | -62.25 |
| 70.0 | 2946.29 | 84115.25 | -224.76 | -79.46 |
| 65.0 | 2719.69 | 79918.16 | -275.24 | -93.51 |
| 60.0 | 2445.37 | 74323.00 | -322.55 | -104.68 |
| 55.0 | 2149.81 | 67630.19 | -365.10 | -113.18 |
| 50.0 | 1851.10 | 60140.55 | -401.08 | -119.21 |
| 45.0 | 1560.56 | 52154.14 | -429.14 | -123.01 |
| 40.0 | 1285.21 | 43969.47 | -448.75 | -124.88 |
| 35.0 | 1029.76 | 35882.47 | -460.29 | -125.19 |
| 30.0 | 798.00 | 28185.46 | -464.96 | -124.37 |
| 25.0 | 593.73 | 21165.70 | -464.60 | -122.86 |
| 20.0 | 421.21 | 15102.93 | -461.42 | -121.17 |
| 15.0 | 285.41 | 10266.47 | -457.84 | -119.76 |
| 10.0 | 191.84 | 6910.94 | -456.17 | -119.11 |
| 5.0 | 146.35 | 5274.89 | -458.56 | -119.65 |

See Reference 8.0 C-1 and Page 59A for definition of terms (A)

--------COMPUTATIONS COMPLETE

Table 9



TORQUE FOR AS-INSTALLED CONDITIONS FOR
Valve Nos. IVR007A

Model Data For Aerodynamic Torque Modification: Mitered elbow 2 diameter upstream Geometry 1
All Torques in In-lbs.
(Positive torques tend to close valve)

| 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 | 116 | 1752 | .1 | 12 | 175 |
| 70 | 80 | 246 | 3702 | .9 | 221 | 3332 |
| 60 | 70 | 300 | 4508 | 1 | 300 | 4508 |
| 50 | 60 | 303 | 4557 | 1 | 303 | 4557 |
| 40 | 50 | 278 | 4176 | 1 | 278 | 4176 |
| 30 | 40 | 243 | 3650 | 1 | 243 | 3650 |
| 20 | 30 | 214 | 3212 | 1 | 214 | 3212 |
| 10 | 20 | 201 | 3022 | 1 | 201 | 3022 |

*For 1.0 PSID **15 PSID

#Torque modification factor represents effect of installed condition as compared to flow in a straight pipe run. Based on Test Data.

Table 10

TORQUE FOR AS-INSTALLED CONDITIONS FOR
Valve Nos. IVR006A, IVR006B

Model Data For Aerodynamic Torque Modification: Valves under straight line flow conditions.
All torques in In-lbs.
(Positive torques tend to close valve)

| 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 | -42 | 1752 | 1 | -42 | 1752 |
| 70 | 80 | -80 | 3702 | 1 | -80 | 3702 |
| 60 | 70 | -105 | 4508 | 1 | -105 | 4508 |
| 50 | 60 | -120 | 4557 | 1 | -120 | 4557 |
| 40 | 50 | -125 | 4176 | 1 | -125 | 4176 |
| 30 | 40 | -125 | 3650 | 1 | -125 | 3650 |
| 20 | 30 | -121 | 3212 | 1 | -121 | 3212 |
| 10 | 20 | -119 | 3022 | 1 | -119 | 3022 |

*For 1.0 PSID **15 PSID

#Torque modification factor represents effect of installed condition as compared to flow in a straight pipe run. Based on Test Data.

Table 11
TORQUE FOR AS-INSTALLED CONDITIONS FOR
Valve Nos. IVR007B

Model Data For Aerodynamic Torque Modification:
All torques in In-lbs.
(Positive torques tend to close valve)

Valves under straight line flow conditions.

| 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 | 116 | 1752 | 1 | 116 | 1752 |
| 70 | 80 | 246 | 3702 | 1 | 246 | 3702 |
| 60 | 70 | 300 | 4508 | 1 | 300 | 4508 |
| 50 | 60 | 303 | 4557 | 1 | 303 | 4557 |
| 40 | 50 | 278 | 4176 | 1 | 278 | 4176 |
| 30 | 40 | 243 | 3650 | 1 | 243 | 3650 |
| 20 | 30 | 214 | 3212 | 1 | 214 | 3212 |
| 10 | 20 | 201 | 3022 | 1 | 201 | 3022 |

*For 1.0 PSID **15 PSID

#Torque modification factor represents effect of installed condition as compared to flow in a straight pipe run. Based on Test Data.

5.3.3 CONCLUSIONS CONCERNING VALVE OPERABILITY

For a LOCA condition it can be seen in Tables 9, 10, & 11 that torques for the subject valves are positive (closing) torques for all disc positions. For these valves, any flow condition from none to maximum, in combination with the timed bench tests show the valve will close within 5 seconds or less. As shown in Section 5.3.1, the valves will operate in both the open and closed directions under maximum LOCA conditions.

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

6.0 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 $\Delta P$ across the valve is determined from the customer's spec and/or data sheet. The containment pressure and temperature condition(s) is based on design basis considerations (i.e. pressure = 15 psig and temperature = 185°F) and are higher

than analytically determined DBA-LOCA conditions. This is much more conservative than torques calculated from a containment pressure time-history analysis at incremental valve angle positions. 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 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, 5.2, 5.3)
3. Installation effects were accounted for in all cases, but downstream piping back pressure was not. A higher than atmospheric downstream pressure would result in a smaller $\Delta$P across the valve, and thus smaller calculated torques. Not accounting for this produces a more conservative calculation. (Section 5.1.3) A
4. Clow does not consider a containment pressure response profile. Clow assumes the isolation signal may be delayed until full containment pressure is reached, then the valve will be called upon to close. Actual time lag for equipment response is not considered by Clow since the approach taken is more conservative. Clow does, however, record test lag time as part of unit bench testing. (Section 1.2.C, 5.3.1, 5.2.3)

5. Valve angle and predicted $\Delta P$ for choking across the valve is presented. The maximum $\Delta P$ is conservatively stated as equal to the maximum service $\Delta P$ for all angles. (Section 5.3.2)
6. Codes used, allowed stresses, and predicted stresses are presented in the Code Design Report and/or Seismic Analysis Report(s). Load combinations are described in these reports . The valve is analyzed by finite element techniques. (Section 4.0)
9. The vent/purge valves located inside containment are not effected by backpressure because both sides of the actuator piston will be influenced by the containment air pressure. Thus we do not believe it is necessary to consider backpressure. (A)

10. Clow to date has not used accumulators for valves used in containment isolation system service.
11. NA to Clow design.
12. Units are not modified to limit the travel angle, except by actuator stops. The purpose of these stops is to limit travel of the disc to being parallel with the pipe centerline. Clow's report shows, from the test data base and bench tests of each unit, that sufficient torque is available to close and seat the valve against all flow induced loads. Since Clow's seat/seal design is conical, no special considerations for low temperature is required. (See Section 5.3.1, 5.3.2)
13. Clow selects operators for each unit with maximum operating torques much larger than that produced by flow interaction with the disc. (See Section 5.3.1)

14. Not applicable to air operators.

15. Not applicable to air operators.

16. Not applicable to air operators. A manual jack screw is provided and the unit is tagged indicating the full disengagement length of the screw. No <u>automatic</u> features are provided to insure disengagement. Proper operation is assumed by administrative controls and procedural checks. (A)

17. The valve, being of all metal construction except for packings, seal laminations, and gaskets, will not degrade under the required environmental conditions. Metal components are generally accepted in the industry as suitable for the required environmental conditions. Tests at both high and low temperatures have been performed by Gebruder Adams of Bokum, West Germany for the subject seal/seat design. Seismic considerations are covered by both analysis and previous static load tests. (See Section 1.2, Section 7.0, Section 8.0 B1).

18. All operators and solenoid valves installed by Clow are qualified to appropriate IEEE requirements by testing. (See Section 2.2.2, 2.2.3).

19. All tests are summarized in the supplied qualification report and are documented by separate test reports. (See Section 8.0 F1)

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 VALVE SEALING CHARACTERISTICS

## 7.1 Normal Sealing

Table 13 shows the sealing ability of the Clinton valves as they were shop tested for record. The tests were performed with pressure on the indicated side of the disc and the opposite side open to atmosphere. The normal recommended flow direction for these valves is with pressure on the shaft side. During this test, the air under water method was used to indicate leakage.

Table 12

VALVE SEALING CHARACTERISTICS

| VALVE MARK NO. | VALVE SIZE (IN.) | TEST PRESSURE PSIG | PRESSURIZED SIDE SHAFT SIDE | PRESSURIZED SIDE CLAMP RING SIDE | LEAKAGE (BUBBLES/MIN) | |
|---|---|---|---|---|---|---|
| 1VR006A | 12" | 15 | X | NA | 0 | A |
| 1VR006B | 12" | 15 | X | NA | 0 | |
| 1VR007A | 12" | 15 | X | NA | 0 | |
| 1VR007B | 12" | 15 | X | NA | 0 | |

7.2 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 7.1). 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 it's life, this match tends to improve and a visual seating pattern appears. This results in improved sealing as the valve ages.

This has been verified by experience and is documented in the Shell International Cycling Test (Reference 8.0 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.

7.3 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 8.0 D2.)

As can be seen from Figures 10 and 11, debris screens are provided to prevent large debris from entering the valves. A

7.4 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.

# 8.0 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-833600-1, Rev.A Seismic Qualification Analysis of Clow 12 Inch Wafer Stop Valve.

2. Technical Report PEI-TR-852400-1, Addendum to PEI Technical Report PEI-TR-833600-1 covering 12" valves IVR006A, IVR006B, IVR007A, IVR007B.

B. Seismic Qualification Test Reports

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

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

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

## 8.0 REFERENCES (con't)

C. Air Flow Tests

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

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

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

D. Other Reports and Information

1. Operating Instructions for Clow Tricentric Wafer Stop Valve covers installation, maintenance, and operating instructions for 83-2462(N) valves.

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

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. Baldwin Associates Design Specification BA-K-2882-29, Issue 2

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

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

## 8.0 REFERENCES (con't)

F. Environmental and Seismic Qualification Reports

1. PEI-TR-852201-02 "Seismic Qualification Status Report on Clow Wafer Valves, G.H. Bettis Valve Actuators, and Associated Control Components", dated 4/15/85

2. G.H. Bettis Nuclear Qualification Test Report 37274, Revision 9, dated November 28, 1984.

3. "Qualification of EA180 Series Limit Switches for use in Nuclear Power Plants" Namco Controls Report No. QTR105, Revision 1, dated August 28, 1980. A

4. "Report on Qualification of Automatic Switch Company (ASCO) Catalog NP-1 Solenoid Valves for Safety-Related Applications in Nuclear Power Generating Stations," ASCO Report No. AQR-67368/Revision 1, updated.

APPENDIX A

NUCLEAR REGULATORY

PURGE VALVE

OPERABILITY

GUIDE LINES

# BRANCH TECHNICAL POSITION CSB 6-4 *

# CONTAINMENT PURGING DURING NORMAL PLANT OPERATIONS

## A. BACKGROUND

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

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

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

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

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

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

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

The design and use of the purge and vent lines should be based on the premise of achieving acceptable calculated offsite radiological consequences and assuring that emergency core 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 margins to withstand loads imposed while valve closes during a design basis accident. Torsional shear, shear, bending, tension and compression loads/stresses should be considered. Seismic loadings should be addressed.

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

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

Bench Testing

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

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

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

3. Whether the following load and environmental factors were considered

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

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

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

In-Situ Testing

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

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

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

End CSB 6-4

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

1. The $\Delta P$ across the valve is in part predicated on the containment pressure and gas density conditions. What were the containment conditions used to determine the $\Delta P$'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 $\Delta P$ (across valve) | Maximum $\Delta P$ (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 of the 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 DBA 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 rationale 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 containment.

APPENDIX B

DESCRIPTION OF OPERATIONAL TESTS

OF A 12 INCH CLOW TRICENTRIC VALVE

FOR

NUCLEAR PURGE SYSTEM SERVICE

BY

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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 and for normal operation 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. A