REFERENCE 6.2-23

CLOW

Clow Corporation
Engineered Products Division

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Westmont, IL 60559

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PURGE AND VENT VALVE OPERABILITY

QUALIFICATION ANALYSIS

Report No. 6-06-83

PREPARED FOR

PHILADELPHIA ELECTRIC CO.
LIMERICK GENERATING STATION
UNIT 1

by

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June 1983

Work performed under Bechtel Purchase Order Number 8031-P-144 AC, Rev. 1 items 1.1, 1.3, 1.5, 1.7, 1.9, 1.11, 1.13

Clow Job Numbers: 82-2053-01, 02, 03, 04, 05, 06, and 07(N)

This report covers Valve Mark Nos: MO-57-109, 112, 115, 135, 147, 161, 162, 163, 164, and AO-57-104, 114, 121, 123, 124, 131.

CERTIFICATION

This is to certify that all valves (Tag Nos. MO-57-109, MO-57-112, MO-57-115, MO-57-135, MO-57-147, MO-57-161, MO-57-162, MO-57-163, MO-57-164, AO-57-104, AO-57-114, AO-57-121, AO-57-123, AO-57-124, AO-57-131) have been evaluated for operability under the installed conditions indicated in Bechtel Material Requisition 8031-P-144, Rev. 1 and accompanying specifications as amended by Clow exceptions and processed Bechtel SDDR's. The information contained in this report is the result of complete and carefully conducted analyses and to the best of our knowledge is true and correct in all respects. The information presented in combination with the supporting documents referenced, represents a demonstrated qualification of the subject valves to the best of our knowledge for the required service application.

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1. INTRODUCTION

The Nuclear Regulatory Commission has, since 1979, been highly concerned about the operability of purge and vent valves during certain postulated occurrences. Their study in this area has shown that many valves were designed only to operate under normal flow requirements. For a postulated loss of coolant accident, such valves may fail to close in the time required to prevent discharge of radioactive gases to the outside environment. Such a failure could exceed 10 CFR guidelines and present a significant hazard to the health of persons in the area.
NRC Branch Technical Position CSB 6-4 gives some backgroud 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 Clow's Tricentric butterfly valve, the Bettis pneumatic actuator, and Limitorque electric actuator used to operate the valve. In addition descriptions of various tests performed to determine flow and torque characteristics

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

1.1 Testing Performed

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

In the Spring of 1981, Clow personnel met with representatives of the NRC to review the test program to that point and to obtain recommendations for additional testing. As a result, Clow and it's fluid dynamic consultant set up two additional programs to determine how the aerodynamic torque characteristics of the Tricentric valve varied with installed piping conditions. For such 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 elbow (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 practive). Due to the fact that each experiment required an increasing amount of test combinations, the experiments were done in a phased approach.

The upstream elbow tests were performed first for a scale model of a 12" valve in 3 orientations relative to the elbow and and a 3 spacings (2, 4, & 8 diameters) from the elbow. From the results a worst case was determined to occur at 2 diameters.

Thus the scale models of the 24" and 48" valves were tested only at 2 diameters. Upstream elbow effects diminished significantly at 4 diameters and were barely detectable at 8 diameters.

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

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

1.2 Qualification Method

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

covered in this and supplemental reports.

A. Environmental

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

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

B. Structural (For Seismic and Other Loadings)

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

Qualification Report, the code required Design Report, and the Structural Analysis Report. The elements considered by these reports include:

1. Valve body
2. Valve disc
3. Valve disc shaft
4. Valve disc shaft connection
    a. Disc ear
    b. Drive keys
    c. Dowel pin (retains shaft from hydrostatic end load only)
5. Actuator mounting structure
    a. Adaptor flange
    b. Bolting

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

C. Operability Under Flow

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

worst case conditions (minimum voltage, air supply, or maximum backpressure for pneumatic actuators if applicable).

The following method is used to show operability:

1. Determine no flow worst case operating time from bench tests.
2. Using Clow program CVAP calculate aerodynamic torques for straight pipe conditions.
3. Determine a torque modification factor based on the installed (from buyer prints) or a worst case upstream piping condition using the mitered elbow or two valves in series test data.
4. Determine predicted torque values for all disc angles based on 2 and 3 above.
5. Provide tabulation or plot of actuator output torque for all actuator angles
6. Show that actuator output provides sufficient margin to overcome aerodynamic and other torques (bearing, packing, disc wt.) to close the valve.
7. From the above data, actuator type, and Vought full size test valve data, project a closing rate under the conditions analyzed above.

In the above calculations, the following assumptions are employed:

a. Containment pressure is at a maximum value and full flow is developed before the 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 back-pressure and lower torques.

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

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

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

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

2.0 DESIGN OF VALVE AND ACTUATOR ASSEMBLY

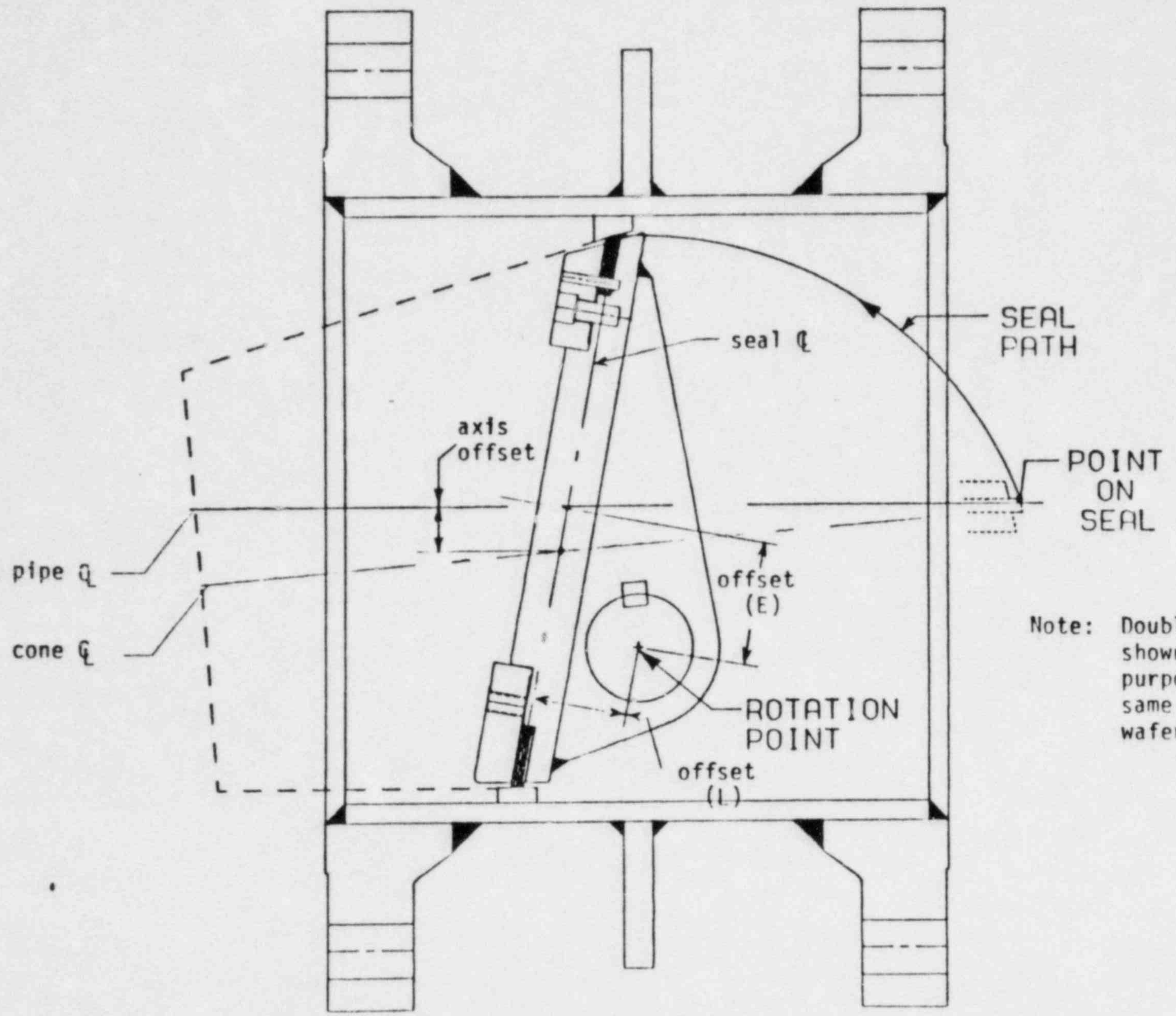
2.1 Valve Design

2.1.1 Geometry

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

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

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



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

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

2.1.2 Materials

A complete list of valve component materials used on Bechtel Purchase Order 8031-P-144-AC may be found on the General Arrangement Drawings (D-0699 thru D-0705) which follow this section.

Since purge and vent valves must perform safety related functions not only during normal conditions but also during and after upset, emergency and faulted conditions, the material selections were based on a worst case event. Because the valves are required to prevent discharge of radioactive gases to the outside environment during a LOCA, the seat and seal materials are critical to the operation of the valves. During normal operation the valves are exposed to the air in the containment and outside air, but during a LOCA the media may be made up of steam and air [redacted] which may be radioactive and at elevated temperatures. The seat material selected for this application was SA479 316L SST. The 316 grade was selected due

to its corrosion resistance and ability to withstand all of the possible medias that may come in contact with the seat. The L grade of 316 SST was further specified because the seat is welded to the body (SA516 GR70) and the L grade has a lower carbon content that will reduce the carbide precipitation in the heat affected zone of the seat. The seal is a laminate of 316 SST and asbestos. The 316 SST was chosen in the "straight" grade since no welding is done on the seal. The asbestos used is made of John Manville style 60 or Klinger K-61 material. The laminated type seal was selected for its ability to seal with less torque than would be required for a solid seal. The laminate allows each SST member to act independently and to conform to the contour of the machined seat as seating torque is applied. The asbestos member not only allows each SST member to act independently but also reduces the seal area in contact with the seal and therefore, results in application of higher normal stresses to the seal for any given seating torque.

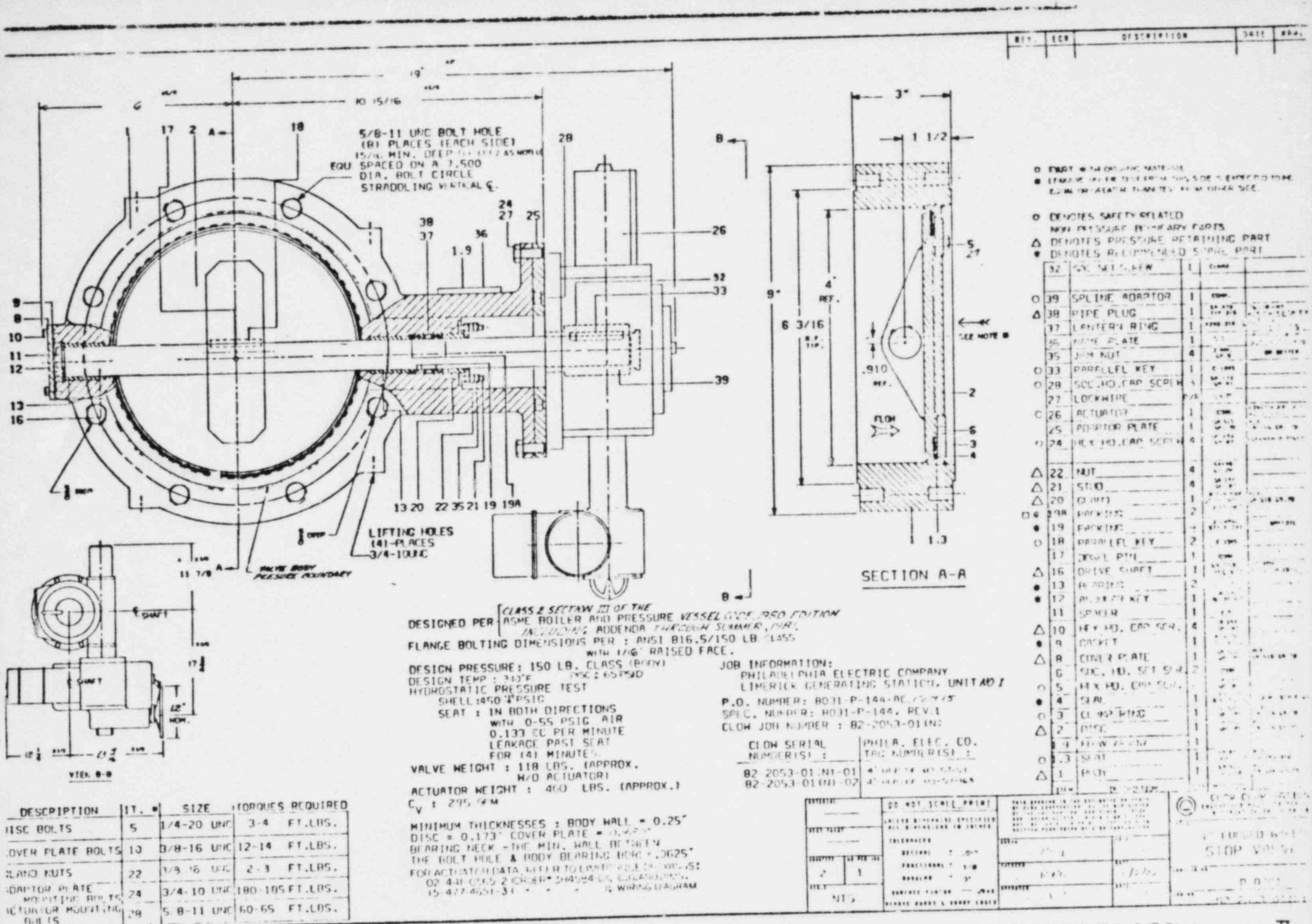


FIGURE 2 - 4" MOTOR OPERATED VALVE ASSEMBLY AND MATERIALS

Page 1

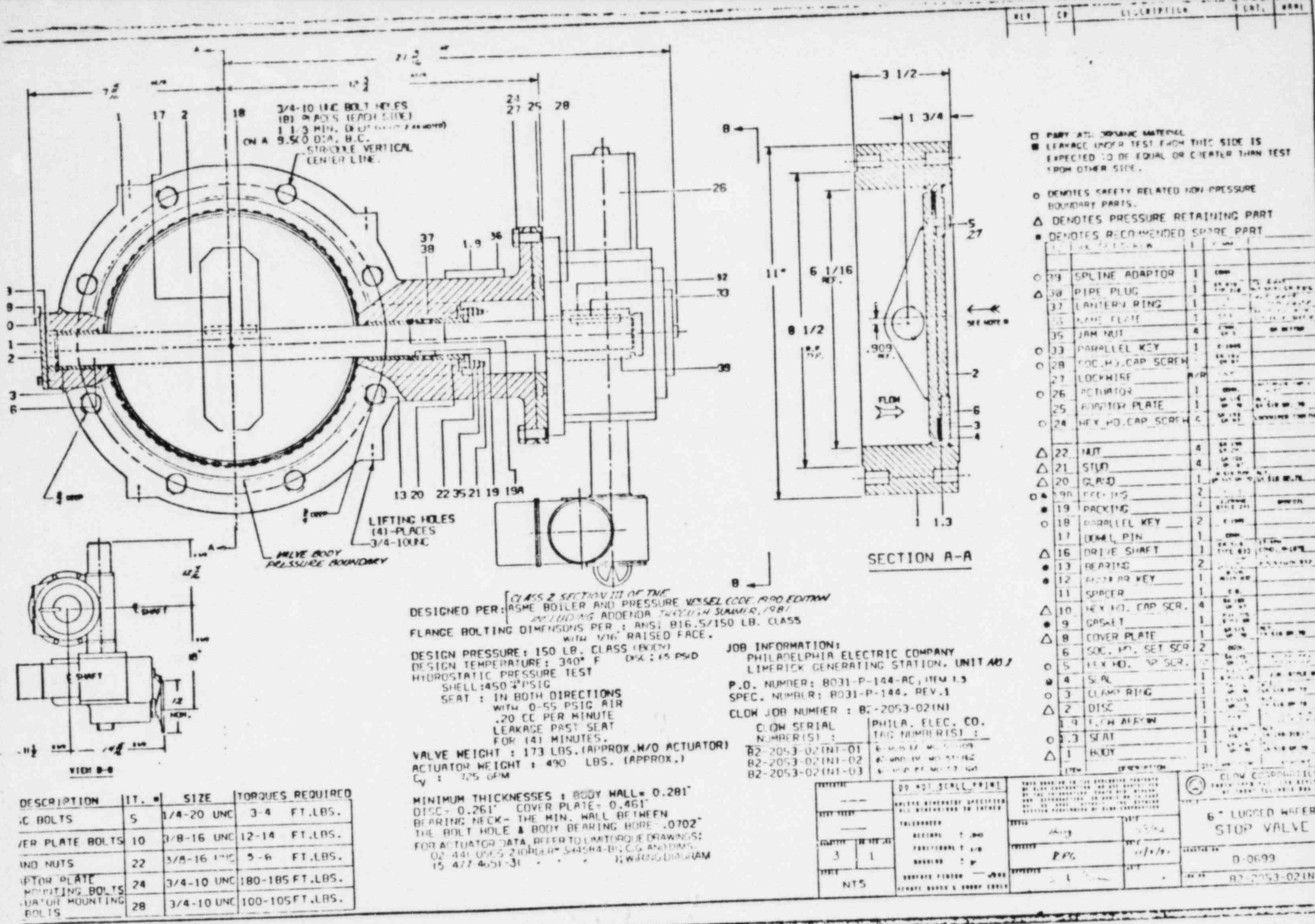
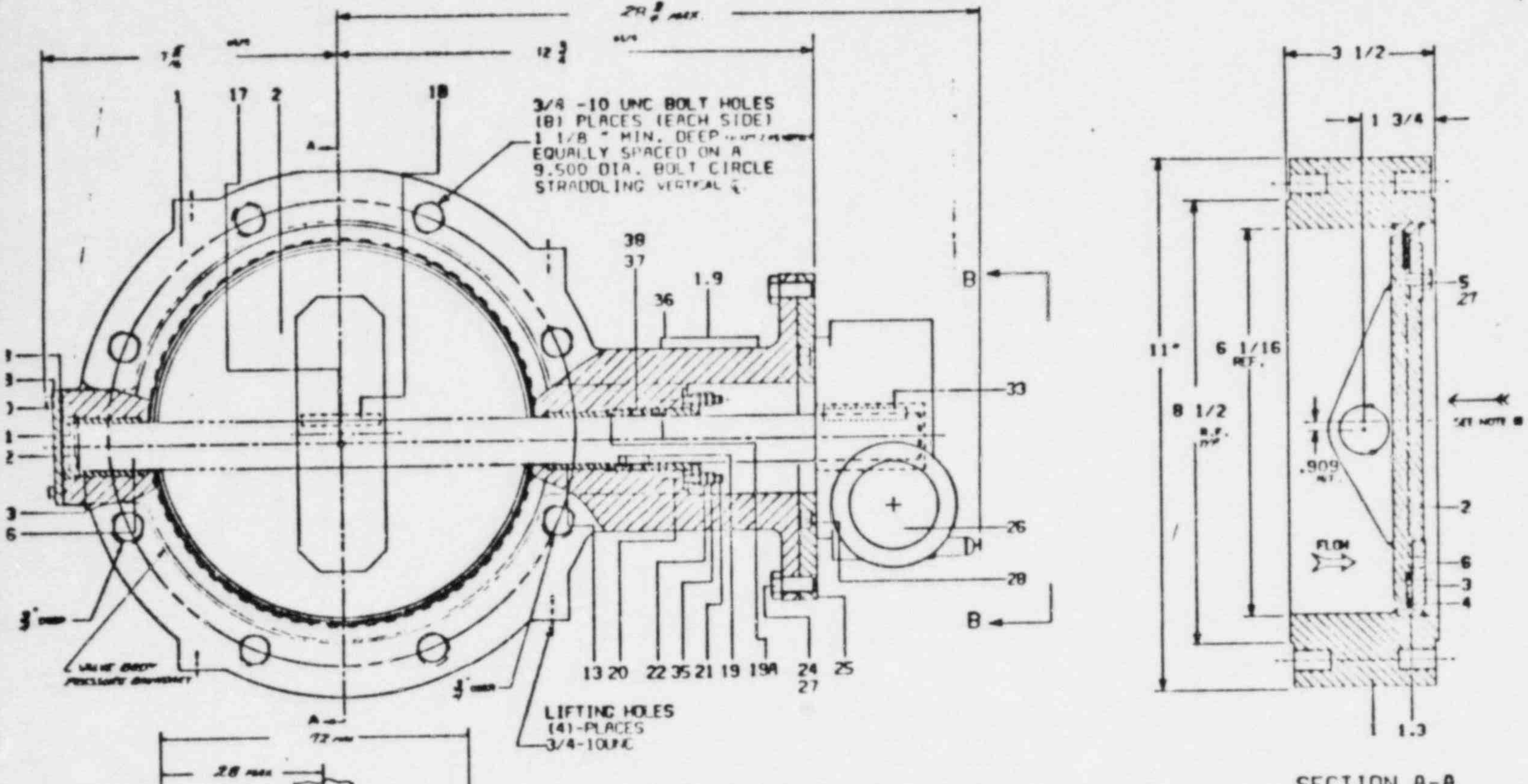
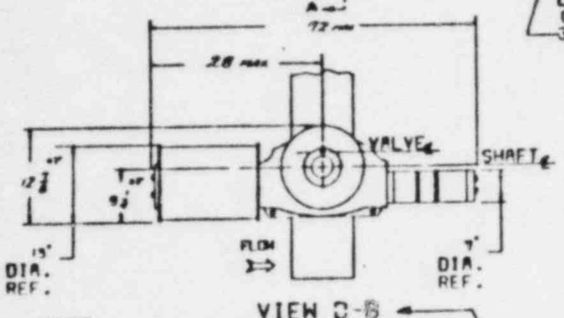


FIGURE 3 - 6" MOTOR OPERATED VALVE ASSEMBLY AND MATERIALS





- ■ LEAKAGE UNDER TEST FROM THIS SIDE IS EXPECTED TO BE EQUAL OR GREATER THAN TEST FROM OTHER SIDE.
- □ PART WITH ORGANIC MATERIAL
- ○ DENOTES SAFETY RELATED NON-PRESSURE BOUNDARY PARTS.
- △ DENOTES PRESSURE RETAINING PART
- ● DENOTES RECOMMENDED SPARE PART

NOTE:
ACTUATOR CAN BE ROTATED TO POSITION 180° FROM THAT SHOWN.

DESIGNED PER: CLASS 2 SECTION III OF THE ASME BOILER AND PRESSURE VESSEL CODE, 1980 EDITION INCLUDING ADDENDA THROUGH SUMMER, 1981

FLANGE BOLTING DIMENSIONS PER: ANSI B16.5/150 LB. CLASS WITH 1/16" RAISED FACE

DESIGN PRESSURE: 150 LB. CLASS (BODY)
DESIGN TEMPERATURE: 340° F DISC 25 PSID
HYDROSTATIC PRESSURE TEST
SHELL: 450 PSIG
SEAT: IN BOTH DIRECTIONS WITH 0-55 PSIG AIR .20 CC PER MINUTE LEAKAGE PAST SEAT FOR (4) MINUTES.

VALVE WEIGHT: 173 LBS. (APPROX. W/O ACTUATOR)
ACTUATOR WEIGHT: 533 LBS. (APPROX.)
C_v: 725 GPM

MINIMUM THICKNESSES: BODY WALL = 0.281"
DISC = 0.261" COVER PLATE = 0.461"
BEARING NECK - THE MIN. WALL BETWEEN THE BOLT HOLE & BODY BEARING BORE = .0702"
ACTUATOR AIR SUPPLY: 80 PSIG MIN 150 PSIG MAX

FOR ACTUATOR DATA, (C.G. AND MOUNTED ACCESSORIES) REFER TO G.H. BETTIS DRAWING SPC-963.

JOB INFORMATION:
PHILADELPHIA ELECTRIC COMPANY
LIMERICK GENERATING STATION, UNIT NO 1

P.O. NUMBER: B031-P-144-AC, ITEM NO 1/3
SPEC. NUMBER: B031-P-144, REV.1

CLOW JOB NUMBER: 82-2053-03(N)

CLOW SERIAL NUMBER(S):	PHILA. ELEC. CO. TAG NUMBER(S):
82-2053-03(N)-01	[illegible]
82-2053-03(N)-02	[illegible]

DESCRIPTION	IT. #	SIZE	TORQUES REQUIRED
SC BOLTS	5	1/4-20 UNC	3-4 FT.LBS.
VER PLATE BOLTS	10	3/8-16 UNC	12-14 FT.LBS.
AND NUTS	22	[illegible] UNC	5-6 FT.LBS.
APTOR PLATE MOUNTING BOLTS	24	3/4-10 UNC	180-185 FT.LBS.
TUATOR MOUNTING HOLES	28	3/4-10 UNC	100-105 FT.LBS.

	ITEM	DESCRIPTION	QTY	MATERIAL	
△	38	PIPE PLUG	1	[illegible]	[illegible]
	37	LANTERN RING	1	[illegible]	[illegible]
	36	NAME PLATE	1	[illegible]	[illegible]
	35	JAM NUT	4	[illegible]	[illegible]
○	33	PARALLEL KEY	1	[illegible]	
○	28	SOC. HD. CAP SCREW	1	[illegible]	
	27	LOCKWIRE	A/R	[illegible]	
○	26	ACTUATOR	1	[illegible]	[illegible]
	25	ADAPTOR PLATE	1	[illegible]	[illegible]
○	24	HEX HD. CAP SCREW	4	[illegible]	[illegible]
△	22	NUT	4	[illegible]	
△	21	STUD	4	[illegible]	
△	20	GLAND	1	[illegible]	[illegible]
□●	19A	PACKING	2	[illegible]	
●	19	PACKING	[illegible]	[illegible]	[illegible]
○	18	PARALLEL KEY	2	[illegible]	
	17	DOWEL PIN	1	[illegible]	
△	16	DRIVE SHAFT	1	[illegible]	[illegible]
●	13	BEARING	2	[illegible]	[illegible]
●	12	BEARING KEY	1	[illegible]	
	11	SPACER	1	[illegible]	
△	10	HEX HD. CAP SCR.	[illegible]	[illegible]	
●	9	GASKET	1	[illegible]	
△	8	COVER PLATE	1	[illegible]	[illegible]
	6	SOC. HD. SET SCR.	2	[illegible]	
○	5	[illegible] HD. CAP SCR.	1	[illegible]	
●	4	SEAL	1	[illegible]	[illegible]
○	3	CLAMP RING	1	[illegible]	[illegible]
△	2	DISC	1	[illegible]	[illegible]
	1.3	[illegible]	1	[illegible]	
○	1.3	SEAT	1	[illegible]	[illegible]
△	1	BODY	1	[illegible]	[illegible]

DO NOT SCALE PRINT

CLOW CORPORATION

6" LUGGED WAFER STOP VALVE

D-0700

NTS

FIGURE 4 - 6" AIR OPERATED VALVE ASSEMBLY AND MATERIALS

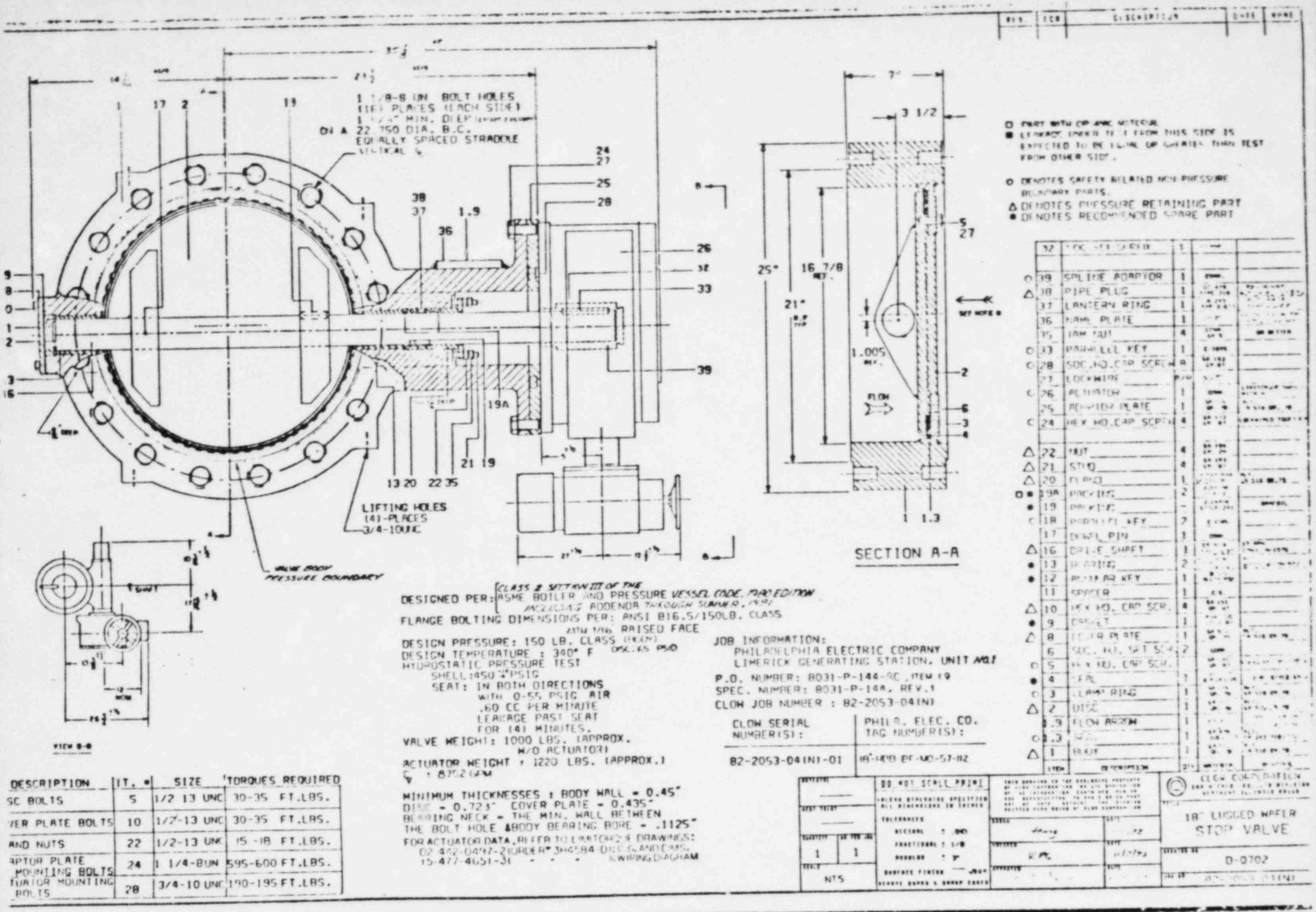


FIGURE 5 - 18" MOTOR OPERATED VALVE ASSEMBLY AND MATERIALS

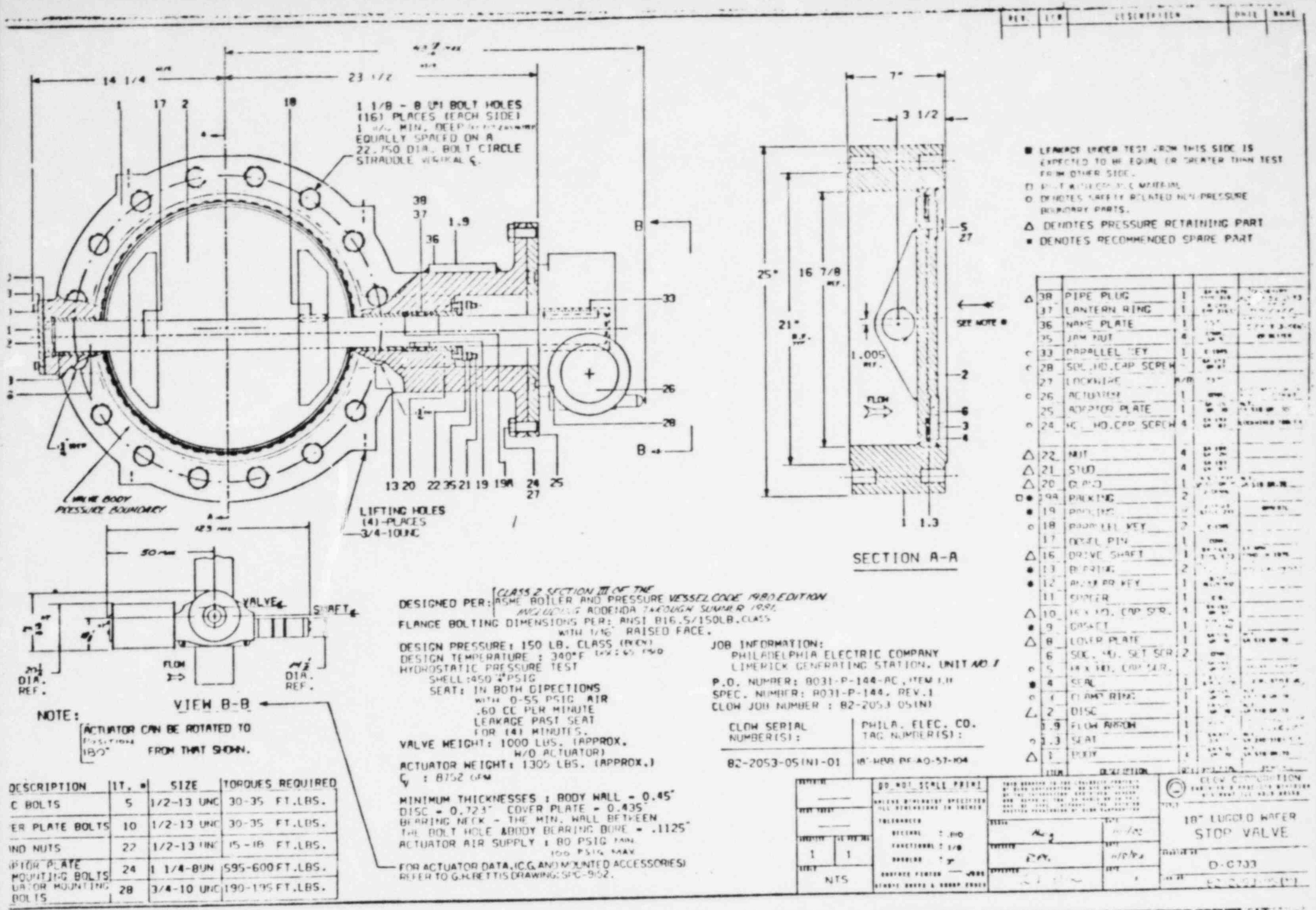


FIGURE 6 - 18" AIR OPERATED VALVE ASSEMBLY AND MATERIALS

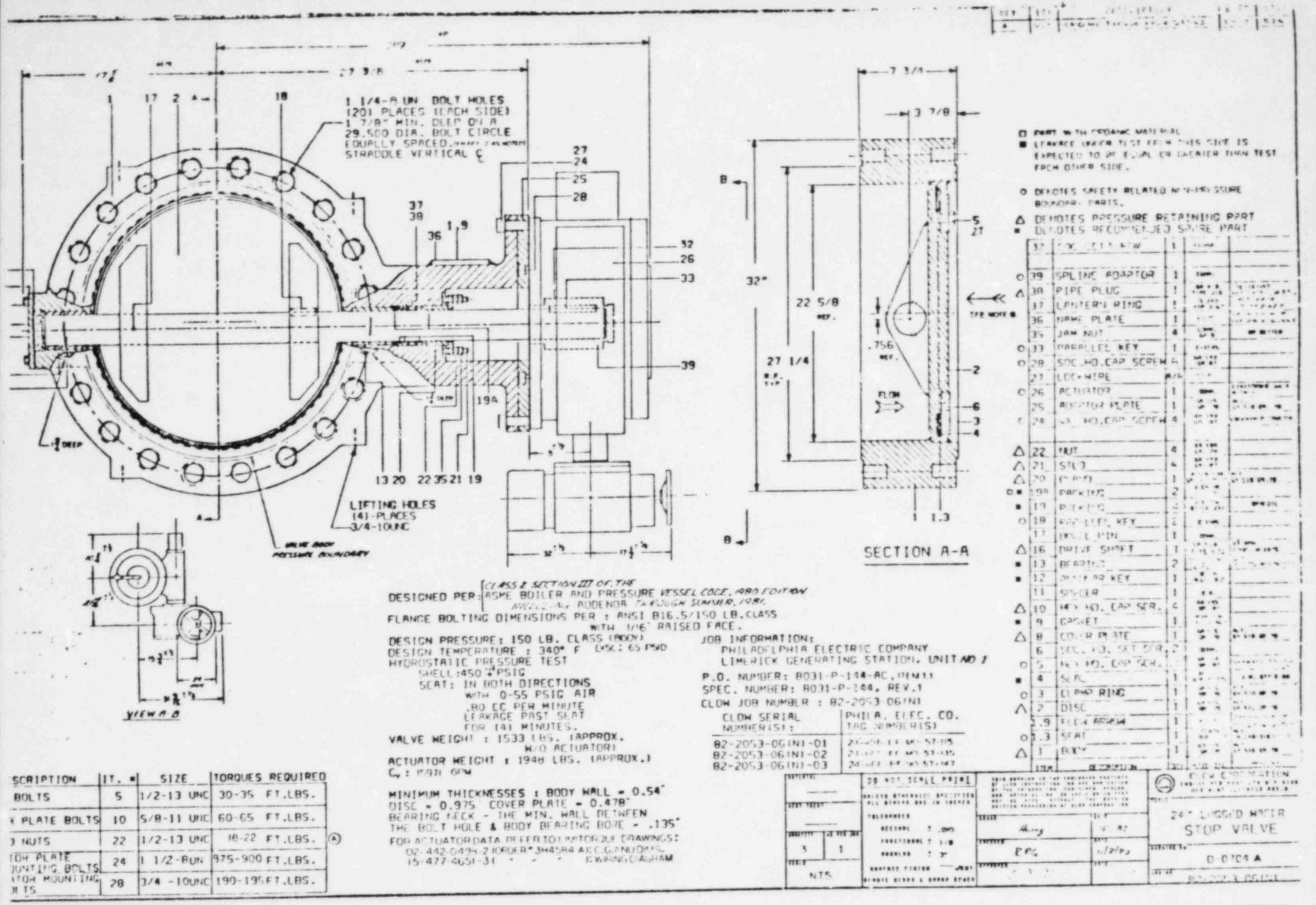


FIGURE 7 - 24" MOTOR OPERATED VALVE ASSEMBLY AND MATERIALS

Page

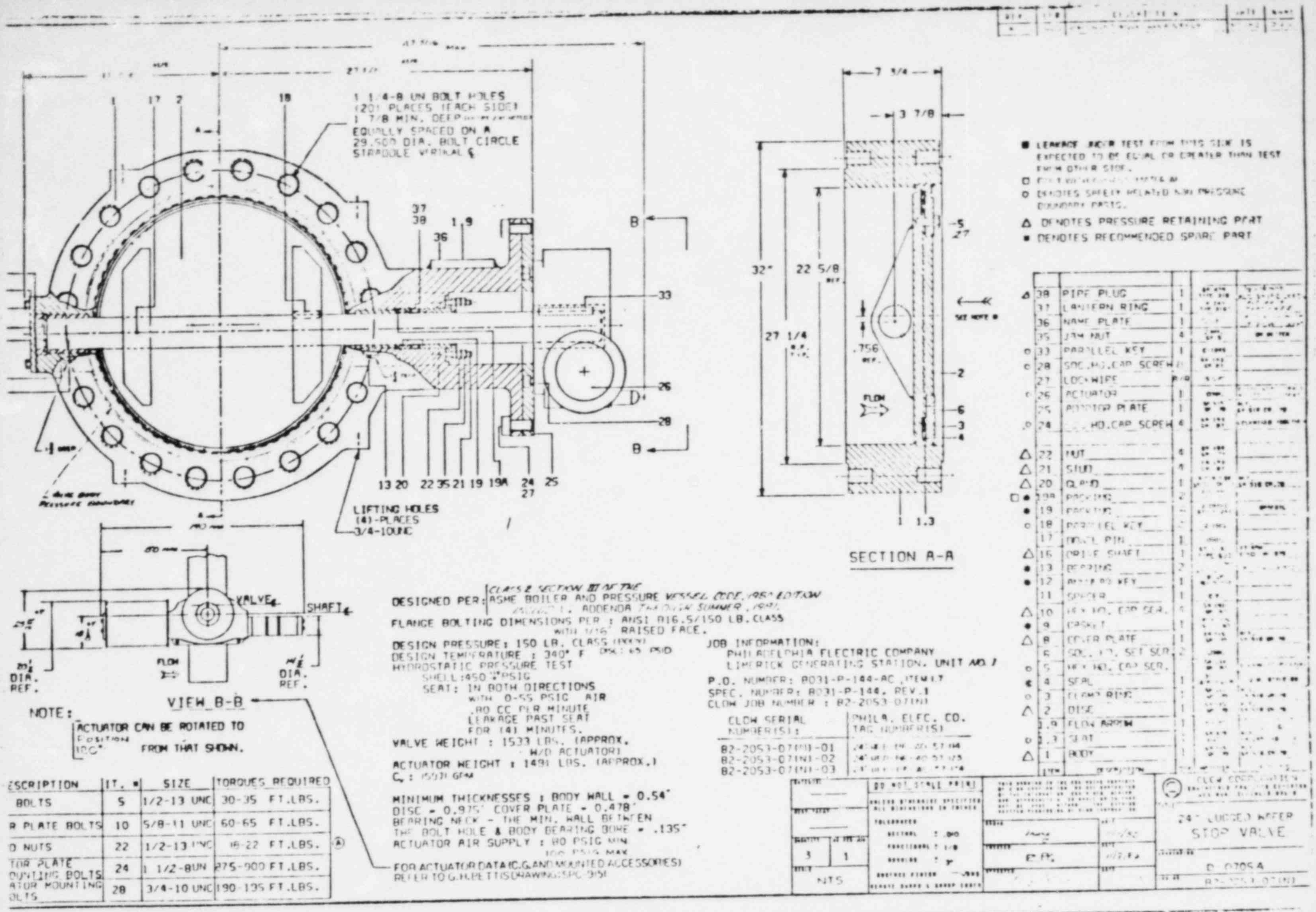


FIGURE 8 - 24" AIR OPERATED VALVE ASSEMBLY AND MATERIALS

2.1.3 Operation

The operation of the Tricentric valve is extremely simple since there are only 2 moving parts, the disc assembly and the shaft. The valve operates by changing the position of the disc relative to the seat. This is accomplished through the application 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.

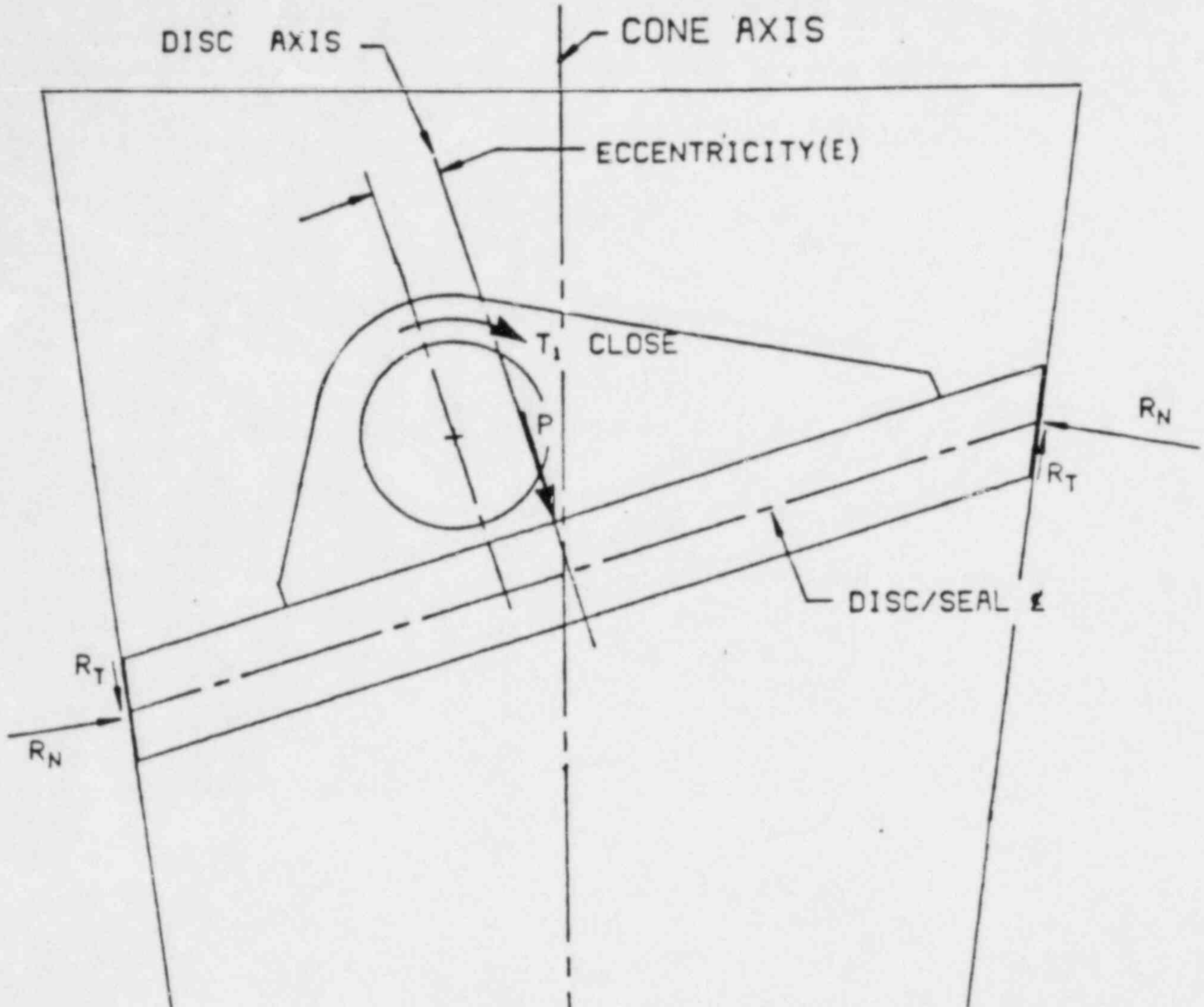
1. Bearing friction torque is the result of the flow or pressure forces acting on the disc which are transmitted to the bearing through the shaft which supports the disc. The bearing friction torque is proportional to these forces acting on the disc and the coefficient of friction between the shaft and 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. 9) 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. 9, as the valve is seated by applying a closing moment T_1, the normal forces R_N will increase. Since the seal angle varies around the seal circumference, R_N also varies, thus the point where R_N is a minimum must be loaded sufficiently to effect a seal. Sealing characteristics will be further discussed in the section under Valve Sealing Characteristics (Section 6.0).



T_1 = Closing torque applied by actuator

P = Force equivalent to disc pressure loading

R_N = Normal seat reaction force due to torque application

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

DISC WITH CLOSING FORCES APPLIED

FIGURE 9

5. Unseating torque is the torque required to move the seal out of contact with the seat. Unseating torque is also dependent on the sealing materials, seal thickness, valve geometry, valve size, differential pressure, and also the seating torque. As described in the section under Valve Design, when no pressure was applied to the valve, the unseating torque was small relative to the applied seating torque. However, when pressure is applied to the shaft side of the disc, not only does the normal force (R_N) increase but also the frictional force (R_T) which resists opening. This increase in frictional force may exceed the PAM torque. Thus an actuator is selected to provide an output torque greater than PAM torque. Typically 1.2 to 1.5 times the PAM torque is required to unseat the valve.

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

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

*Center of Gravity

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

2.2 Actuator Design (Pneumatic Spring Return)

2.2.1 Geometry

The basic actuator is a device by which air pressure is converted to thrust through a linear cylinder and then converted to a rotary (90^0) 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 10.

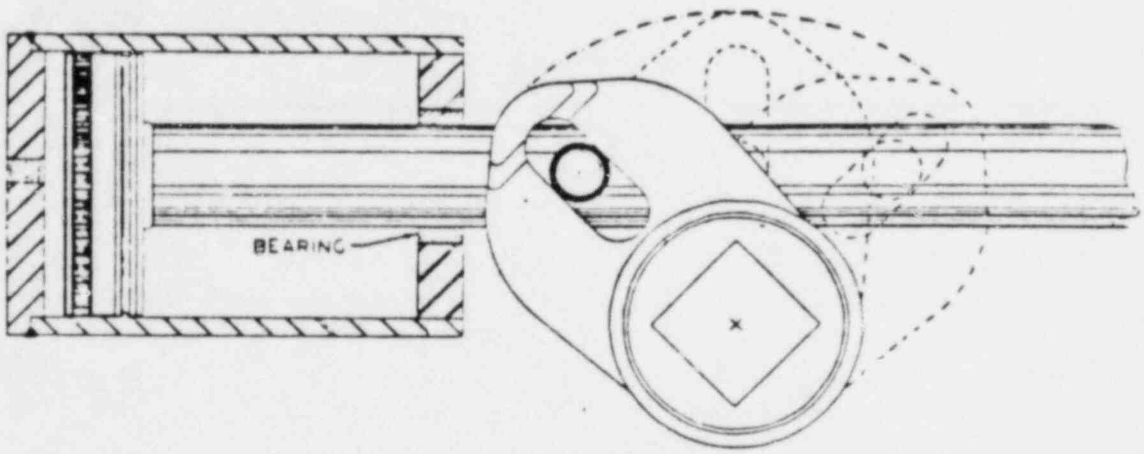


FIGURE 10 - ACTUATOR SCOTCH YOKE DESIGN

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° 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°.

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

TORQUE AT CENTER OF STROKE

$T = P \times A \times MA$

Where:

T = Torque in in-lb

P = Operating pressure in p.s.i.

MA = Moment arm in inches at center

A = Area of the piston in square inches

TORQUE AT BEGINNING AND END OF STROKE

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

Where:

T = Torque in in-lb

F = Resultant total force in pounds = $P \times A$

$\frac{MA}{\text{Cos}.\ 45^{\circ}}$ = 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 11.

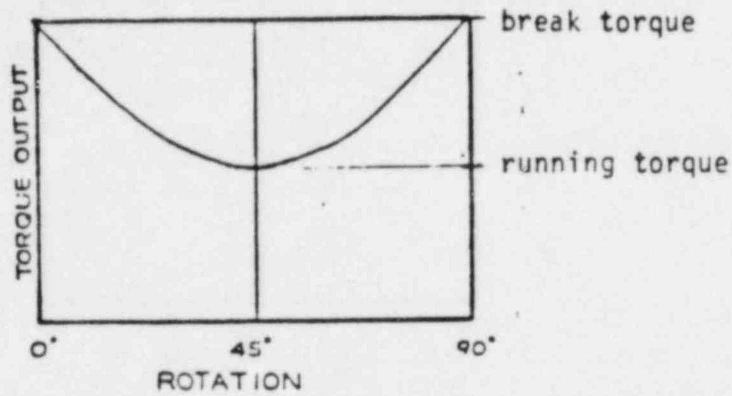


FIGURE 11- Typical torque output for double acting scotch yoke actuator.

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.

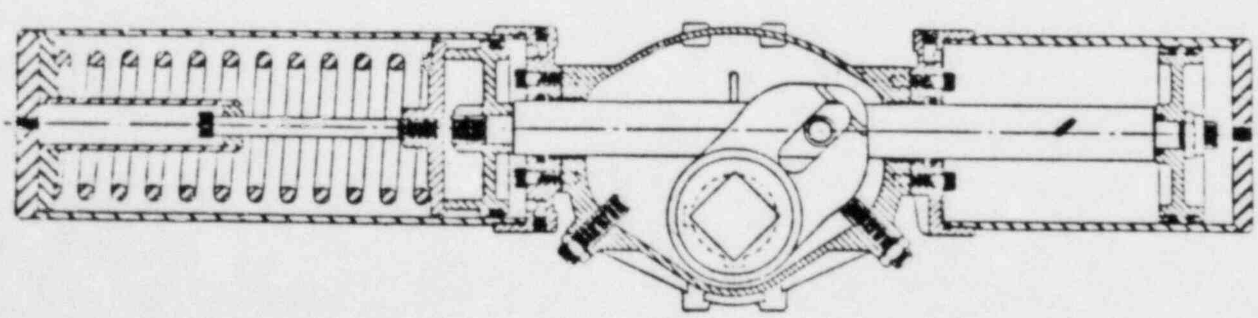


FIGURE 12 - Fail safe, spring return actuator design

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

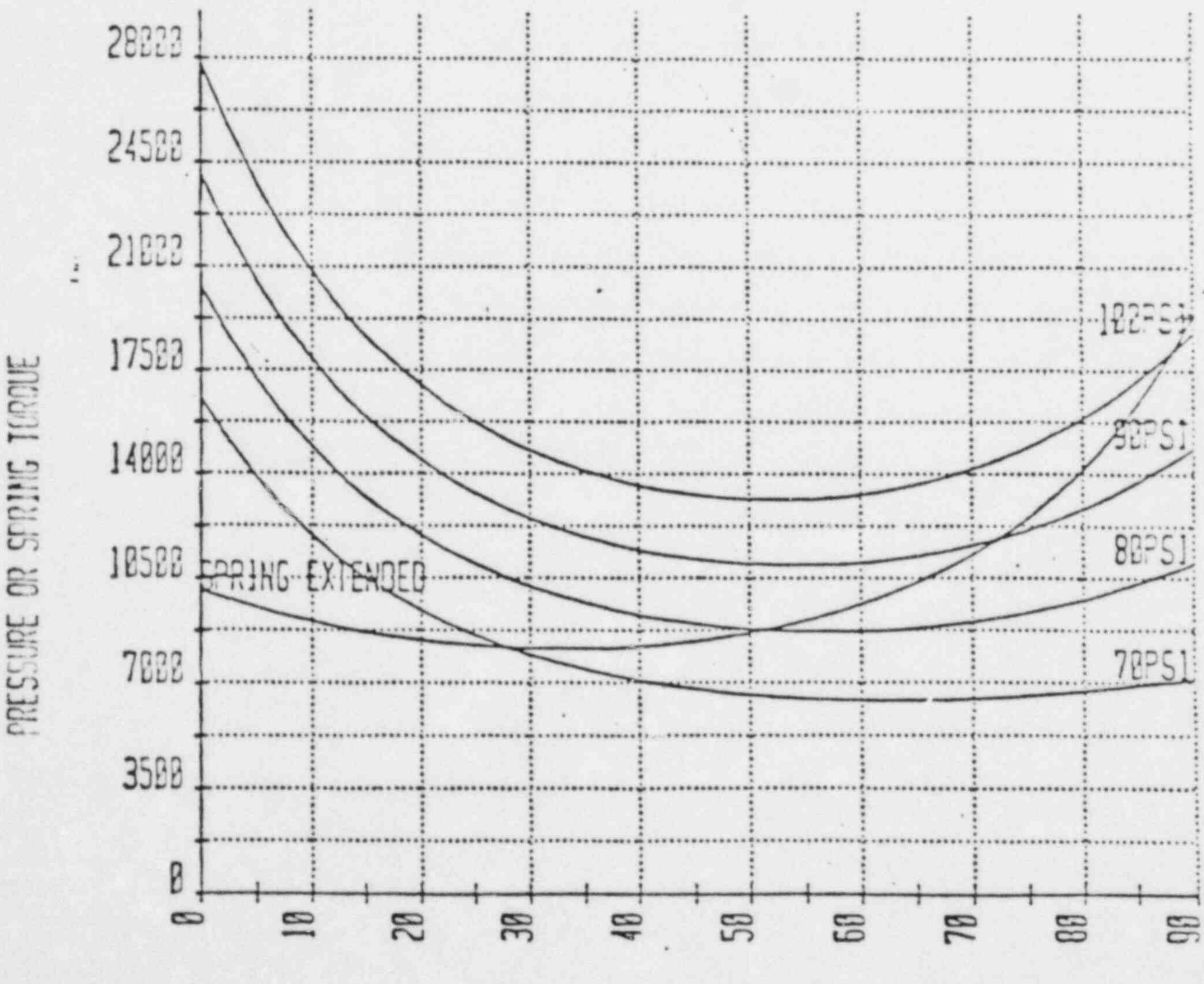


FIGURE 13 - Typical torque output curves for a spring return actuator

2.2.2 Actuator Design Materials

The Bettis actuators used for this job are the T series actuators. These were further specified to be the N version for nuclear service and qualified per IEEE 323-1974, IEEE 344-1975, and IEEE 382. These actuators incorporate use of special materials for nuclear service as listed below.

Special Material:

- Grease - Mobil 28
- Seals - Ethylene Propylene (certified to 1.4×10^8 rads)
- Internal cylinder coating - Molybdenum disulfide
- Yoke pin and rollers - Ryton coated

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

2.2.3 Actuator and Valve Operation (Pneumatic/Spring Return)

2.2.3.1 Actuators and Accessories Supplied

A complete list of all accessories used on each pneumatically actuated valve can be found in Table 1 and each is further described here.

An Asco solenoid valve(s) is used on each actuator to control the air supply to the actuator and, to "dump" the air in the cylinder which allows the valve to open or close as required. The solenoid valves are 3 way, internal piloted diaphragm valves. The solenoid valves are controlled by a 120 VAC coil. When the coil is de-energized by intentional or faulted conditions, the cylinder port is allowed to discharge through the exhaust port and thereby allow the spring return actuator to perform its required function. When the coil is energized, the supply pressure is directed into the cylinder and rotates the valve in a direction opposite to spring induced rotation. Two solenoid valve models are used, one is a NP831664E. This valve is designated for use in nuclear power applications which consists of providing IEEE compliance and a waterproof solenoid enclosure.

It is also 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. The other solenoid valve used is a NP8316E34E which is identical to the NP831664E except the port size is 1 in. NPT and the orifice is 1 in.

Limit switches are also provided, mounted on the actuator to indicate full open or closed position. One of each model no. switch is provided, one set for the open position and the other set for the closed position. The switch model Nos. are Namco

EA 180-31302 and EA 180-32302 which are DPDT switches with 2 NO and 2 NC contacts and are quick make-quick break type. The switches meet NEMA 1, 4, and 13 and also all applicable IEEE requirements. The switches are of the spring return type with one model being CW operation and the other CCW operation. Both switches use the same lever arm which is a Namco model EL-060-56500.

TABLE 1

PNEUMATIC ACTUATED UNITS

Valve Size (in.)	Mark Nos.	Clow Job No.	Bettis Actuator Model No.	Fail-safe Rotation (viewed from top of unit)	Fail-safe Valve Position	Actuator Accessories: Asco Solenoid Valve Model No.	Actuator Accessories: Namco limit switches and lever arm Model Nos. (2 closed position switches) (2 open position switches)	Other Accessories
6	AO-57-121 AO-57-131	82-2053-03(N)	NT312-SR5	CW	Close	NP831664E (Qty. 1)	EA 180-31302 L.S. EA 180-32302 L.S. EL 060-56500 L.A.	Rosedale Filter Y6-3/4-25-B Fisher Regulator 95H-41 Hoffman Enclosure A-1008 CHNF
18	AO-57-104	82-2053-05(N)	NT820-SR5	CW	Close	NP8316E34E (Qty. 2)	EA 180-31302 L.S. EA 180-32302 L.S. EL 060-56500 L.A.	Rosedale Filter Y15-40-BO Fisher Regulator 95H-53 Hoffman Enclosure A-1008 CHNF
24	AO-57-114 AO-57-123 AO-57-124	82-2053-07(N)	NT820-SR4	CW	Close	NP8316E34E (Qty. 2)	EA 180-31302 L.S. EA 180-32302 L.S. EL 010-53337 L.A.	Rosedale Filter Y15-40-BO Fisher Regulator 95H-53 Hoffman Enclosure A-1008 CHNF

2.2.3.2 Pneumatic Actuator Output Torques

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

Table 2

Actuator Model	Guaranteed Torque/Calculated Torque	%
NT312-SR5	5,510/5,810	.95
NT820-SR5	63,300/62,160	1.02
NT820-SR4	95,500/93,098	1.03

The graphs which follow show how the torque output varies for the pressure stroke as a function of supply pressure. It can also be seen that the spring output torque is not a function of supply pressure. The graphs also demonstrate that the output torque (pressure on spring stroke) is a function of yoke position. The graphs provided are based on the numerical data provided from the actuator manufacturer.

FIGURE 14A

T312 SR5

DATA INPUT

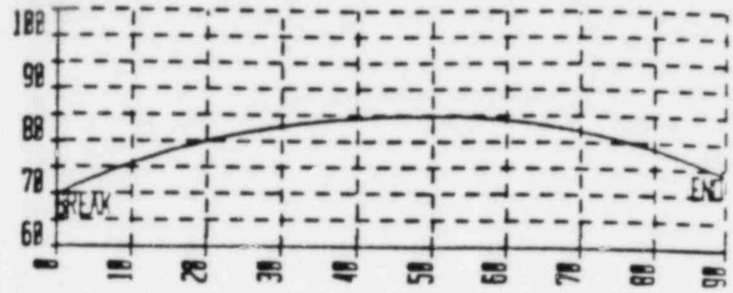
CYLINDER DIAMETER (in)=	12.33
CENTER OR TIE BAR DIAMETER (in)=	0.875
PISTON ROD DIAMETER (in)=	1.375
NUMBER OF PISTONS =	1
MOMENT ARM (in)=	2.812
SPRING LOAD A (lbs)=	1396
SPRING LOAD B (lbs)=	3409
BREAK EFFICIENCY (%)=	70
RUNNING EFFICIENCY (%) =	85
ENDING EFFICIENCY (%) =	74
PRESSURES (psi) =	40 50 60 70
ACTUATOR TYPE,CB=1,HD=2,T=3,R&P=4, =	3

YOKE ARM ANGLE (degrees)	SPRING TORQUE (in lb)	PRESSURE TORQUE (40)psi	PRESSURE TORQUE (50)psi	PRESSURE TORQUE (60)psi	PRESSURE TORQUE (70)psi	EFFICIENCY SPR. %	EFFICIENCY PRES. %
0	5810	12884	17479	22073	26668	74	70
5	5719	10910	15004	19098	23191	77	73
10	5608	9453	13166	16880	20594	79	76
15	5508	8353	11777	15201	18625	81	78
20	5437	7509	10713	13917	17120	82	80
25	5403	6855	9894	12933	15973	83	82
30	5411	6346	9266	12187	15107	84	83
35	5466	5949	8791	11632	14474	85	84
40	5573	5644	8442	11240	14038	85	85
45	5738	5413	8200	10988	13776	85	85
50	5971	5245	8055	10866	13676	85	85
55	6281	5131	7998	10865	13733	84	85
60	6687	5065	8026	10987	13948	83	84
65	7210	5041	8138	11235	14332	82	83
70	7883	5053	8336	11619	14901	80	82
75	8753	5096	8626	12155	15685	78	81
80	9887	5157	9013	12868	16723	76	79
85	11389	5219	9504	13788	18073	73	77
90	13421	5242	10100	14957	19815	70	74

FIGURE 14B

EFFICIENCY PLOT

EFFICIENCY vs ANGLE



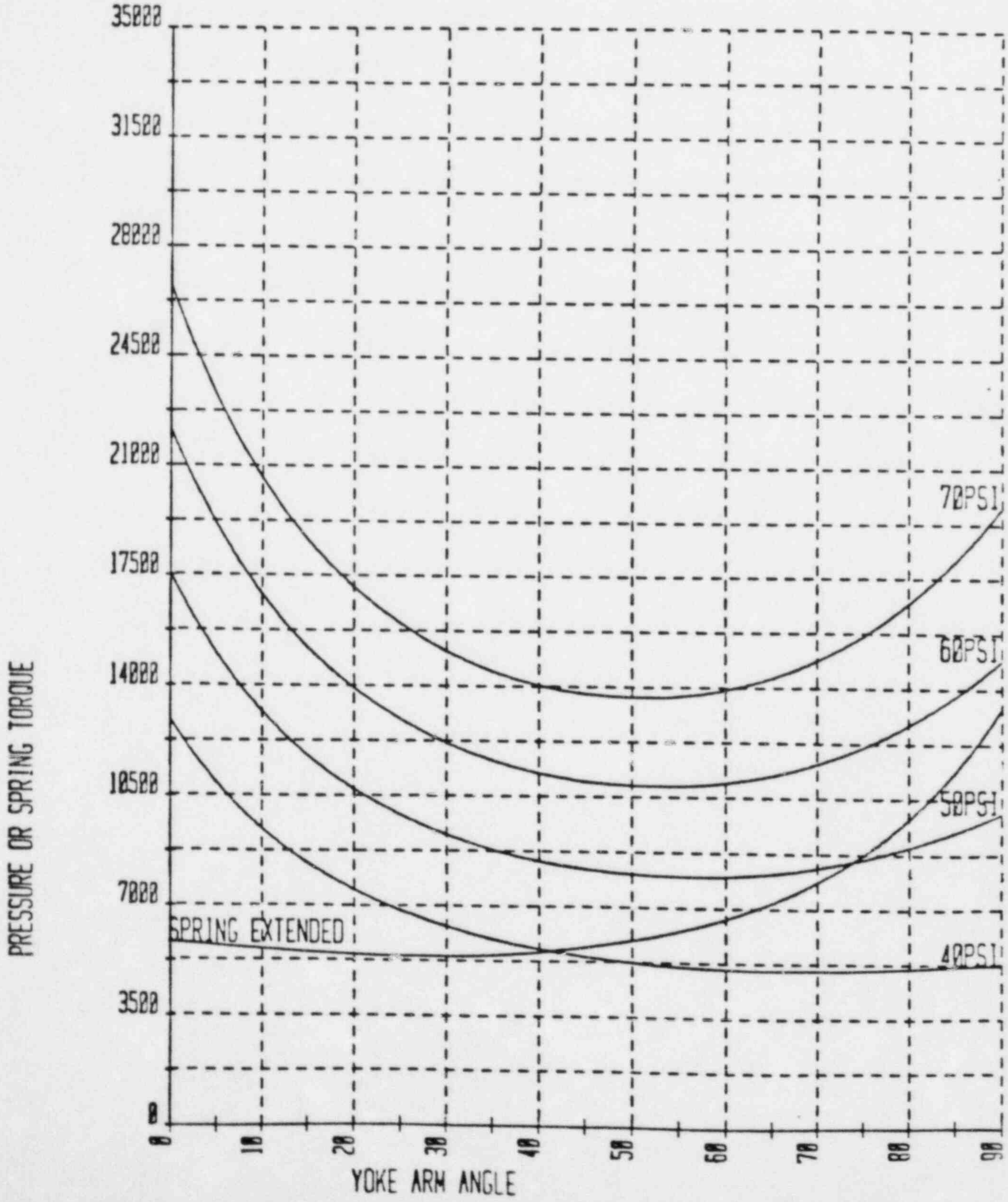


FIGURE 15A

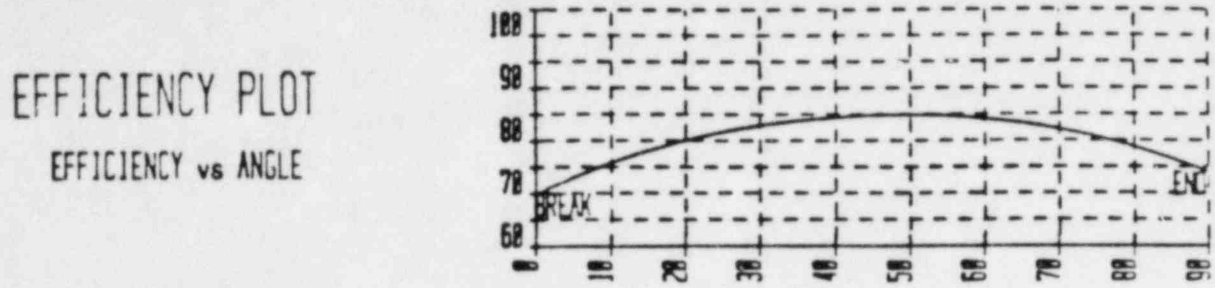
T820 SR5

DATA INPUT

CYLINDER DIAMETER (in)=	19.58
CENTER OR TIE BAR DIAMETER (in)=	1.000
PISTON ROD DIAMETER (in)=	1.750
NUMBER OF PISTONS =	1
MOMENT ARM (in)=	8.000
SPRING LOAD A (lbs)=	5250
SPRING LOAD B (lbs)=	7834
BREAK EFFICIENCY (%)=	70
RUNNING EFFICIENCY (%) =	85
ENDING EFFICIENCY (%) =	74
PRESSURES (psi) =	40 50 60 70
ACTUATOR TYPE,CB=1,HD=2,Y=3,R&P=4, =	3

YOKE ARM ANGLE (degrees)	SPRING TORQUE (in lb)	PRESSURE TORQUE (40)psi	PRESSURE TORQUE (50)psi	PRESSURE TORQUE (60)psi	PRESSURE TORQUE (70)psi	EFFICIENCY SPR. %	EFFICIENCY PRES. %
0	62160	74313	107591	140869	174148	74	70
5	57004	64133	93781	123429	153078	77	73
10	52975	56555	83451	110348	137244	79	76
15	49868	50819	75617	100415	125213	81	78
20	47525	46431	69635	92839	116043	82	80
25	45834	43067	65078	87090	109102	83	82
30	44719	40502	61655	82808	103961	84	83
35	44129	38586	59166	79746	100326	85	84
40	44042	37211	57475	77740	98004	85	85
45	44454	36307	56498	76688	96878	85	85
50	45388	35828	56183	76538	96893	85	85
55	46890	35750	56516	77282	98048	84	85
60	49038	36065	57511	78956	100401	83	84
65	51948	36784	59213	81642	104072	82	83
70	55794	37931	61706	85480	109254	80	82
75	60825	39552	65116	90680	116244	78	81
80	67408	41709	69631	97552	125474	76	79
85	76096	44482	75514	106547	137580	73	77
90	87741	47965	83145	118324	153504	70	74

FIGURE 15B


EFFICIENCY PLOT
EFFICIENCY vs ANGLE
100
90
80
70
60
BREAK
END
0
10
20
30
40
50
60
70
80
90

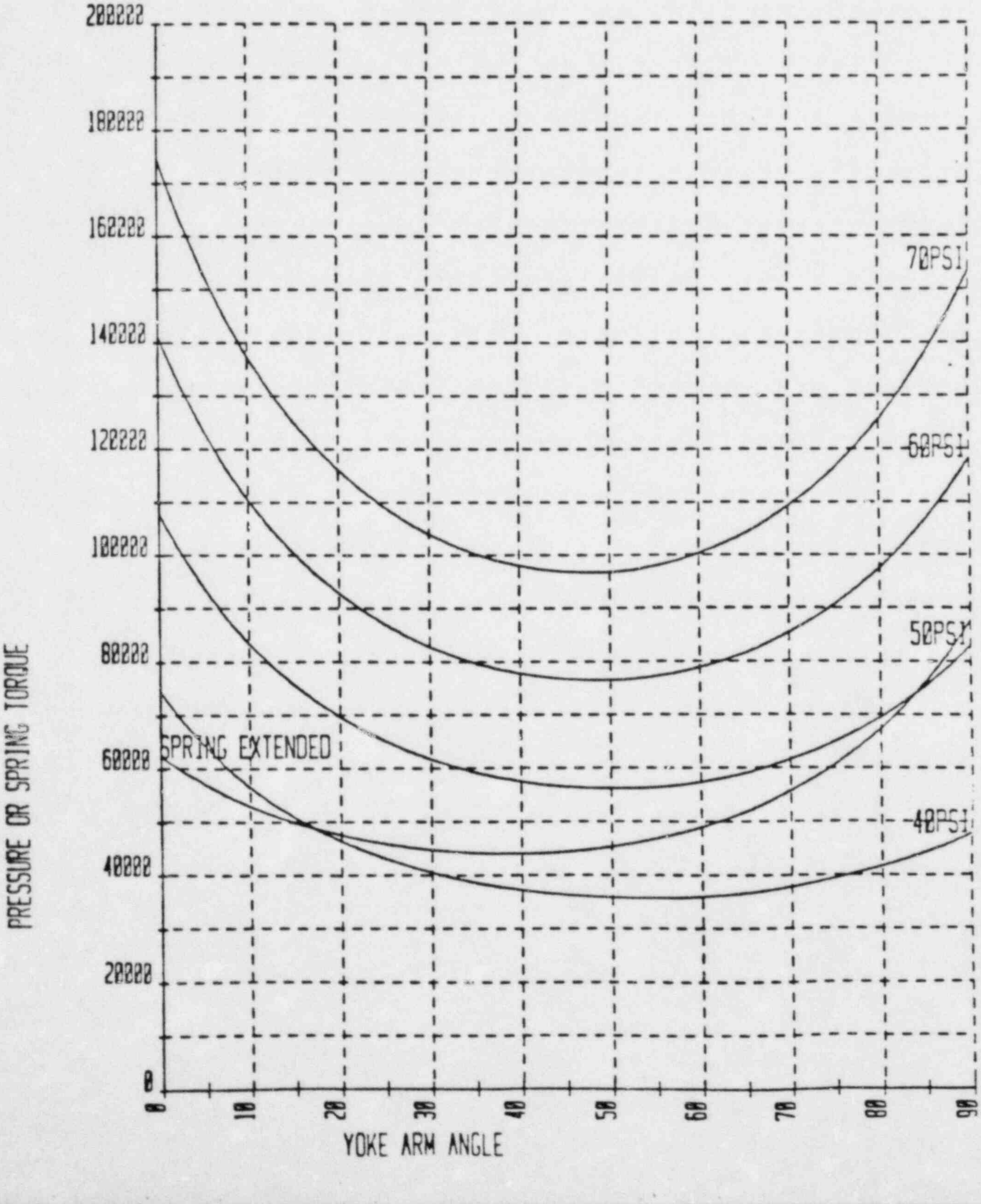

200000
180000
160000
140000
120000
100000
80000
60000
40000
20000
0
70PSI
60PSI
50PSI
40PSI
SPRING EXTENDED
PRESSURE OR SPRING TORQUE
0
10
20
30
40
50
60
70
80
90
YOKE ARM ANGLE

FIGURE 16A

T820 SR4

DATA INPUT

CYLINDER DIAMETER (in)=				19.58
CENTER OR TIE BAR DIAMETER (in)=				1.000
PISTON ROD DIAMETER (in)=				1.750
NUMBER OF PISTONS =				1
MOMENT ARM (in)=				8.000
SPRING LOAD A (lbs)=				7863
SPRING LOAD B (lbs)=				12902
BREAK EFFICIENCY (%)=				70
RUNNING EFFICIENCY (%) =				85
ENDING EFFICIENCY (%) =				74
PRESSURES (psi) =	60	70	80	90

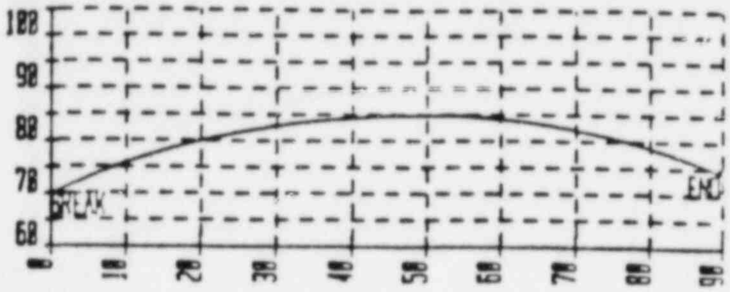
ACTUATOR TYPE,CB=1,HD=2,T=3,R&P=4, = 3

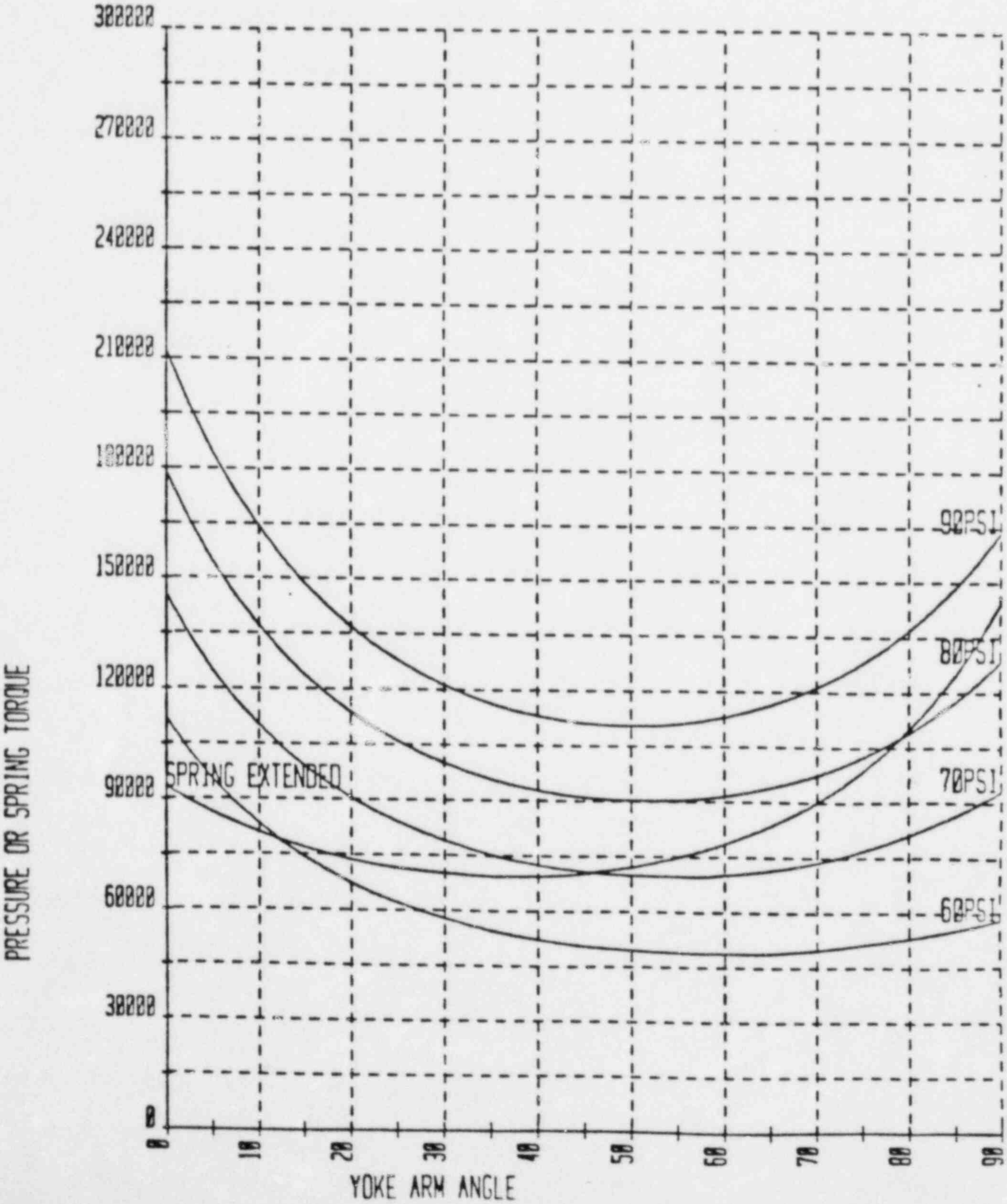
YOKE ARM ANGLE (degrees)	SPRING TORQUE (in lb)	PRESSURE TORQUE (60)psi	PRESSURE TORQUE (70)psi	PRESSURE TORQUE (80)psi	PRESSURE TORQUE (90)psi	EFFICIENCY SPR. %	EFFICIENCY PRES. %
0	93098	111604	144882	178160	211438	74	70
5	86357	95385	125034	154682	184330	77	73
10	80989	83363	110260	137156	164052	79	76
15	76813	74277	99075	123873	148671	81	78
20	73674	67317	90521	113725	136928	82	80
25	71452	61949	83961	105973	127984	83	82
30	70064	57809	78961	100114	121267	84	83
35	69457	54644	75224	95804	116384	85	84
40	69615	52280	72544	92808	113073	85	85
45	70551	50591	70781	90972	111162	85	85
50	72313	49493	69848	90202	110557	85	85
55	74990	48928	69694	90460	111226	84	85
60	78720	48863	70308	91754	113199	83	84
65	83710	49279	71708	94137	116567	82	83
70	90256	50171	73945	97719	121494	80	82
75	98792	51540	77104	102668	128232	78	81
80	109953	53385	81307	109229	137151	76	79
85	124695	55678	86711	117744	148777	73	77
90	144502	58319	93499	128679	163859	70	74

FIGURE 16B

EFFICIENCY PLOT

EFFICIENCY vs ANGLE





2.2.3.3 Operating Time

Bench Test - The following is a summary of the operating times recorded during the operational test performed on each valve. The tests were performed using a 100 psig air supply with a maximum flow rate of approximately 70 SCFM. There was no flow through the valve during this test.

TABLE 3

Mark no. of Valve	Valve Size (inch)	Bettis Actuator Model No.	Opening Time * Sec.	Closing Time Sec.
AO-57-121	6	NT312-SR5	#	#
AO-57-131	6	"	2.7	2.6
AO-57-104	18	NT-820-SR5	22.1	2.5
AO-57-114	24	NT-820-SR4	#	#
AO-57-123	24	"	26.9	1.8
AO-57-124	24	"	26.1	1.7

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

#will be furnished at a later date

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

2.3 Actuator Design (Electric)

2.3.1 Basic Electric Actuator Description (Limitorque Design)

The electric actuator is a device by which electrical energy is converted to a controllable rotary (90^0) motion through the use of a motor, gearing, and electrical control elements. This type actuator has a constant torque output capability over the entire output rotation of 90^0.

The actuator assembly consists of two major subassemblies, these are the manual type HBC and the electric type SMB. The HBC is directly coupled to the valve shaft and provides the output torque to position the valve disc in the required position. The SMB is coupled to the input shaft of the HBC and provides the required torque input to the HBC as well as containing the motor, electrical control elements, and a manual override feature.

A typical HBC is shown in Figure 17.

The HBC is a worm gear type unit that has a self locking gear set. A self locking gear set has a specified input shaft, and output drive. The output drive cannot be rotated to produce a rotation of the input shaft.

This self locking feature prevents the valve disc from "driving" or rotating the HBC input shaft when fluid dynamics forces act upon the valve disc. The HBC also contains mechanical stops for the open and closed position limits of the actuator . Since the Tricentric is a torque seated valve and the valve seat serves as the closed position stop, the mechanical closed position

A TYPICAL HBC GEAR OPERATOR

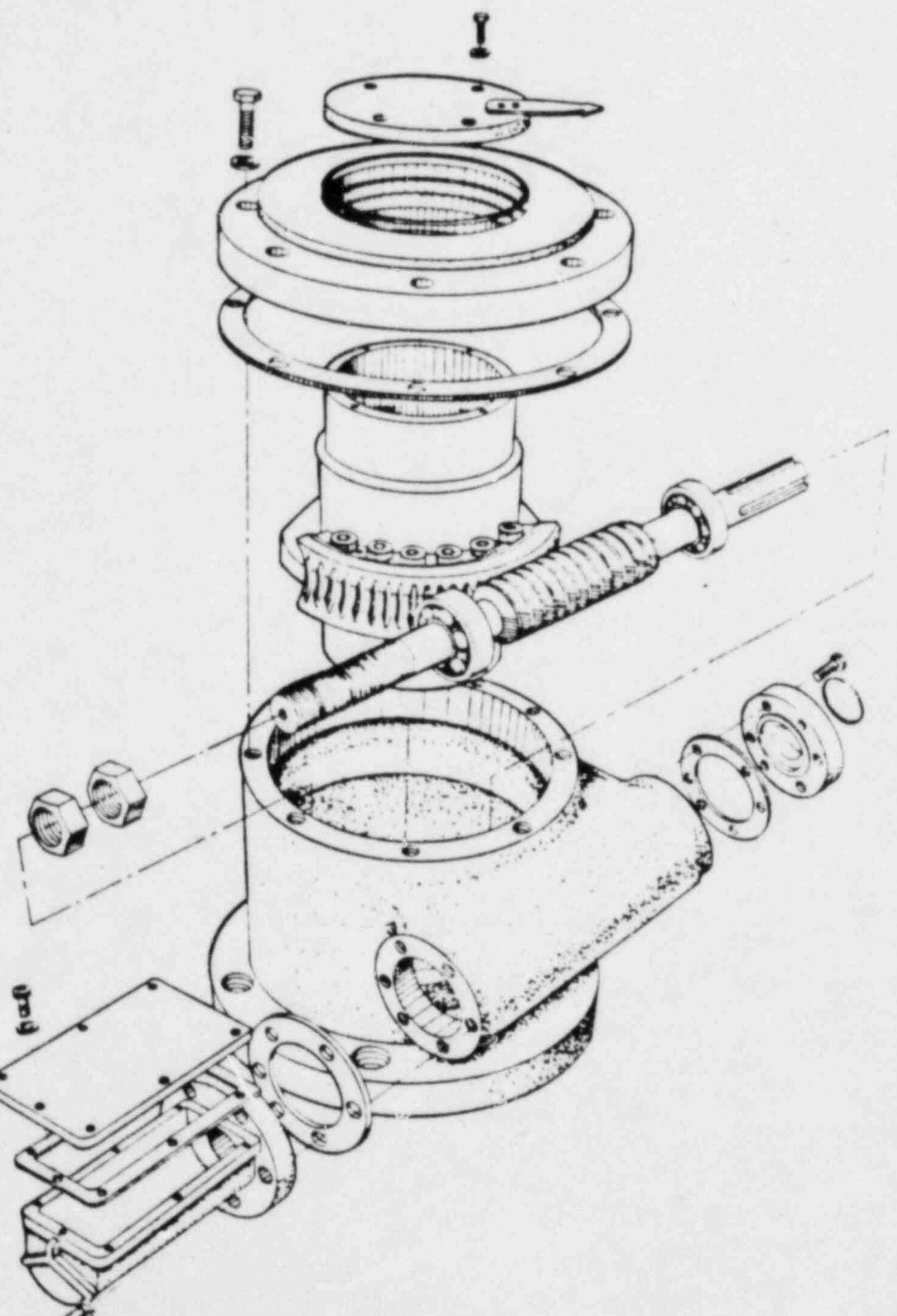
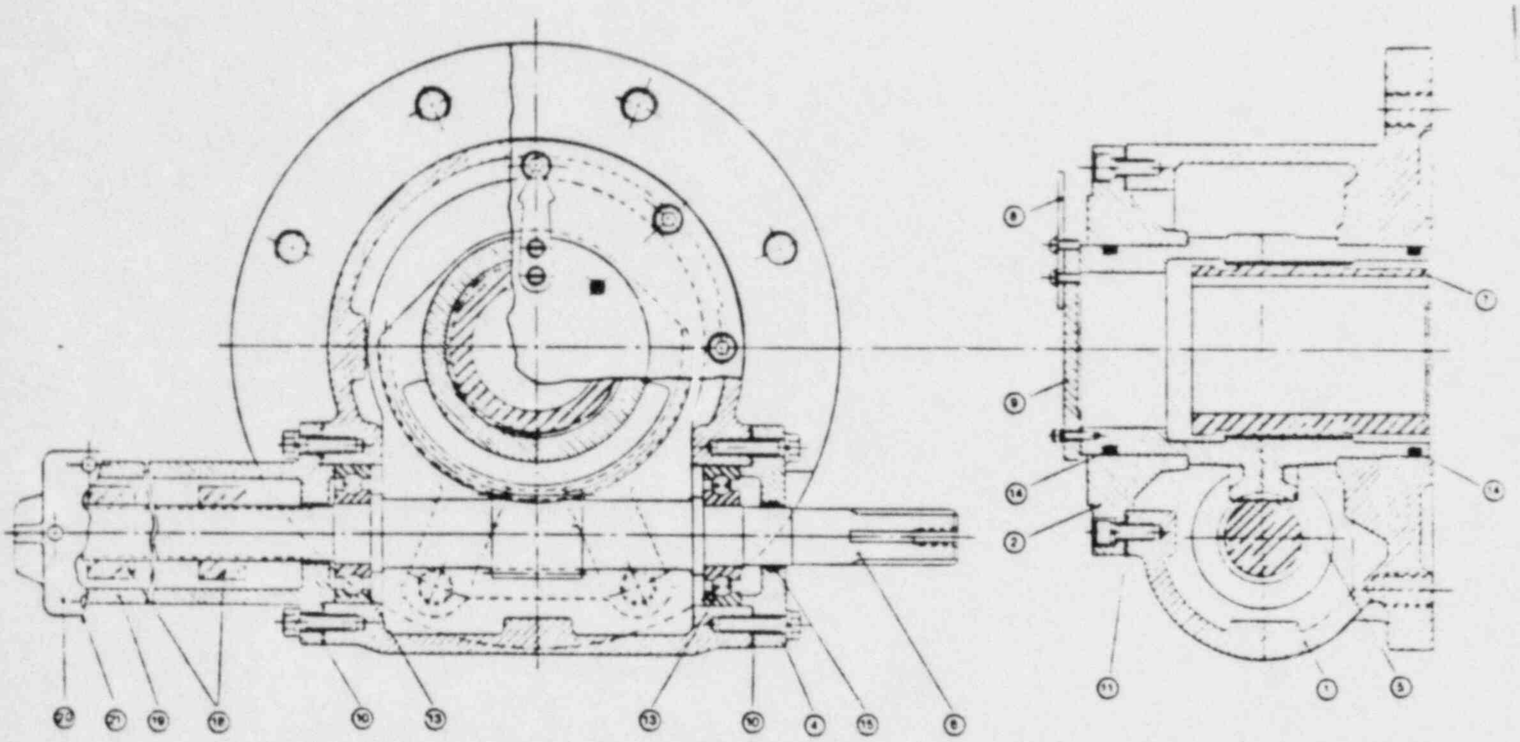


FIGURE 17

stop in the HBC is adjusted to be used as a backup stop only. There are two types of mechanical stops used on HBC actuators. The type of stop used depends on the unit size, with the HOBC thru H3BC using the "Hex Nut" type stops and the H4BC thru H7BC using the "Stop Screw" type stops. The HBC units also have an indicator arrow on the pointer cap that is to show disc position.

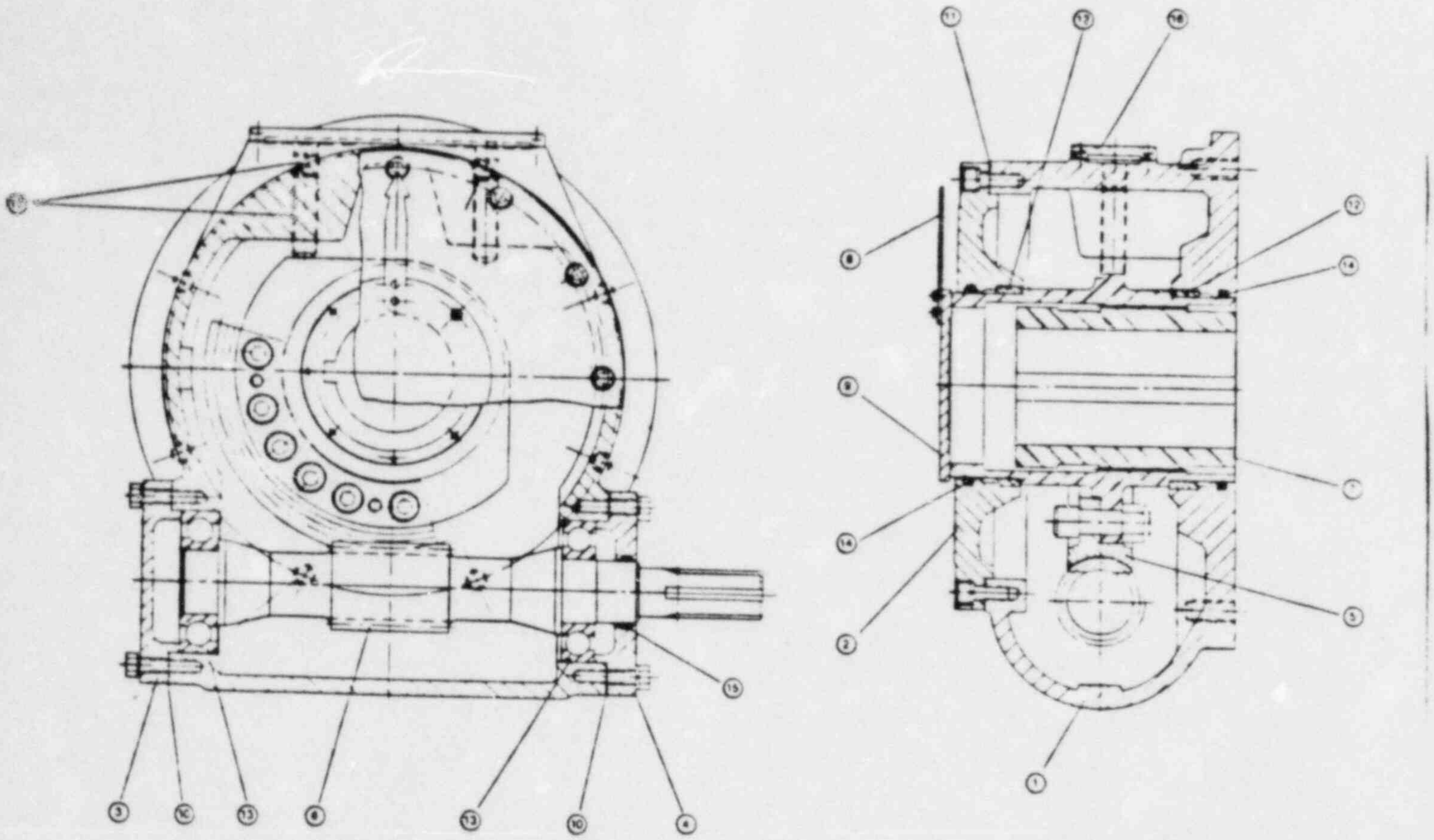
A cutaway drawing showing the construction and component parts for the HOBC thru the H3BC is shown in Figure 18 and the H4BC thru the H7BC is shown in Figure 19.



PC. NO.	DESCRIPTION	PC. NO.	DESCRIPTION	PC. NO	DESCRIPTION
1	HOUSING	8	POINTER	15	WORM SHAFT O RING
2	HOUSING COVER	9	POINTER CAP	16	STOP SCREW COVER
3	END CAP	10	END & THRU CAP GASKET	17	STOP SCREW & LOCKSCREW
4	THRU CAP	11	HSG COVER GASKET	18	HEX STOP NUT
5	DRIVE SLEEVE & WORM GEAR	12	DRIVE SLEEVE BUSHING	19	LIMIT STOP HOUSING
6	WORM SHAFT	13	WORM SHAFT BEARING	20	CAP LIMIT STOP HSG
7	SPLINE ADAPTER	14	DRIVE SLEEVE O RING	21	LIMIT STOP HSG GASKET

FIGURE 18 - HOBC THRU H3BC COMPONENT LISTING

H4BC - H7BC



PARTS LIST

PC. NO.	DESCRIPTION
1	HOUSING
2	HOUSING COVER
3	END CAP
4	THRU CAP
5	DRIVE SLEEVE & WORM GEAR
6	WORM SHAFT
7	SPLINE ADAPTER
8	POINTER
9	POINTER CAP
10	END & THRU CAP GASKET
11	HSG. COVER GASKET
12	DRIVE SLEEVE BUSHING
13	WORM SHAFT BEARING
14	DRIVE SLEEVE O RING
15	WORM SHAFT O RING
16	STOP SCREW COVER
17	STOP SCREW & LOCKSCREW

FIGURE 19 - H4BC THRU H7BC COMPONENT LISTING

Since the HBC is a worm gear device, the output torque will be constant over the operating range of approximately 90^o for a constant torque input. The output torque is directly proportional to the input torque, gear ratio, and efficiency of the unit. The formulas used for determining torque output or required torque input are given below in formulas 1 and 2 respectively.

<u>NOMENCLATURE</u>

T_{out} = Output torque (inch-pounds)

T_{in} = Input torque (inch-pounds)

R = Worm gear ratio (dimensionless)

E = Unit efficiency (dimensionless)

Formula 1

$$T_{out} = T_{in} \times R \times E$$

Formula 2

$$T_{in} = \frac{T_{out}}{R \times E}$$

The maximum output torque of an HBC is dependent on the structural strength of the HBC unit. Table 4 below lists the unit type and size, maximum torque output, gear ratio, and the efficiency. The efficiencies listed are the "Break-away" values since these represent a worst case value used in sizing the units.

TABLE 4

Unit Type & Size	Maximum Output Torque (in. lbs.)	Gear Ratio	Efficiency Break-away
HBC-0	5,340	71:1	0.26
HBC-1	15,600	70:1	0.28
HBC-2	26,400	70:1	0.26
HBC-3	67,800	70:1	0.33
HBC-4	153,600	60:1	0.34
HBC-5	235,000	65:1	0.34
HBC-6	552,000	66:1	0.36
HBC-7	760,000	69:1	0.36

A typical SMB is shown in Figure 20.

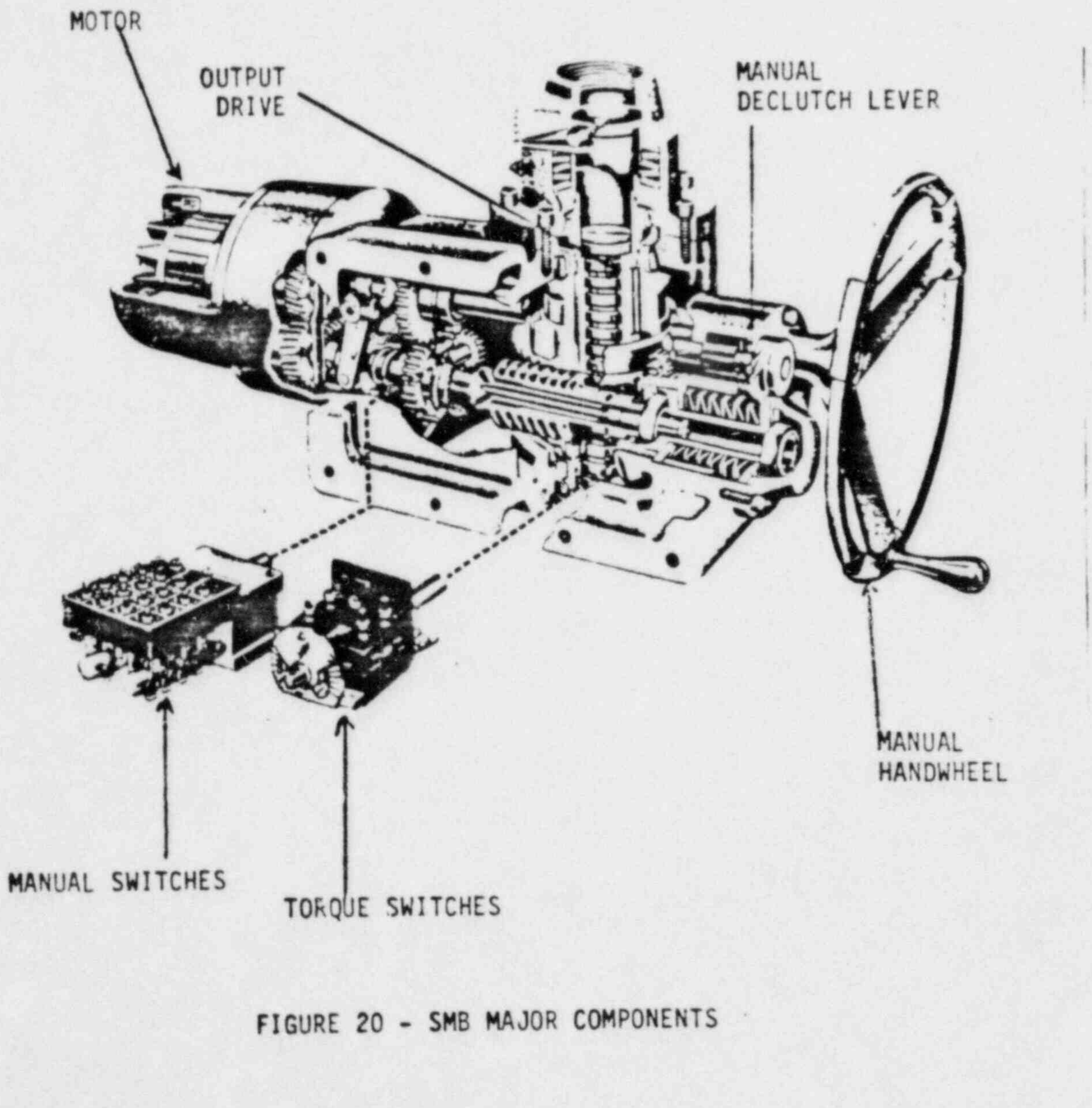


FIGURE 20 - SMB MAJOR COMPONENTS

The electric motor on a SMB drives the worm shaft thru a helical gear set (Figure 21) with the pinion being fixed to the motor shaft. This helical gear set makes up the first reduction from the motor. The worm shaft drives the worm which drives the worm gear. This worm gear set (Figure 22) which is usually self locking, makes the second reduction of the SMB. The worm gear drives the output of the SMB usually thru a "hammer blow" device to form the final drive output which is then coupled to the input shaft of the HBC. The torque switches and limit switches are driven off the worm shaft and provide for both torque and position control of the actuator. The complete drive including the torque switches and limit switches, is shown in Figure 23 . The SMB also provides for manual operation in the event of an electrical power failure. The handwheel on the SMB can be engaged by pulling the declutch lever to the manual position. The SMB also has an electrical enclosure box were all electrical components, including torque switches, limit switches, space heaters, and terminal strips are located.

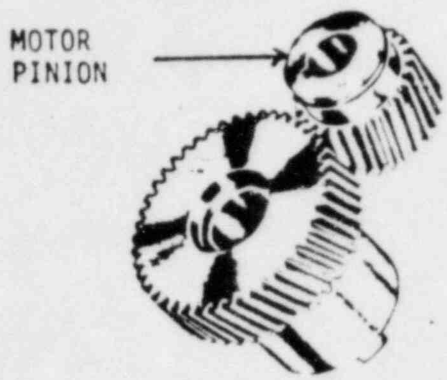


FIGURE 21 - **Helical Gear Set**

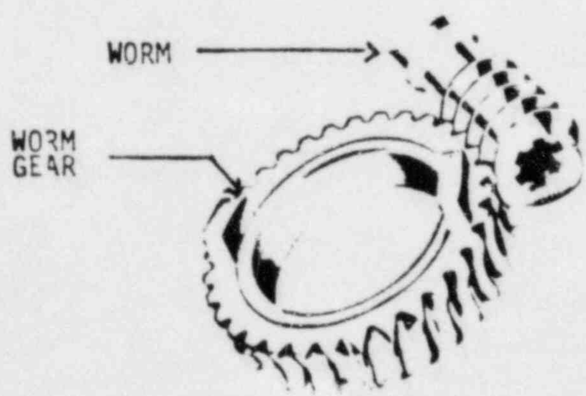


FIGURE 22 - **Worm Gear Set**

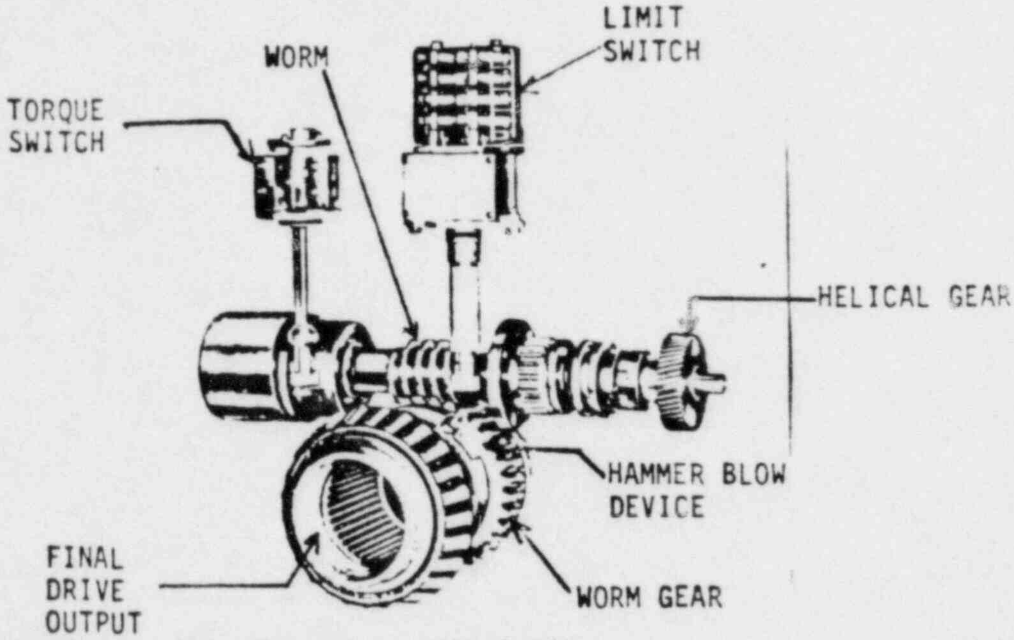


FIGURE 23 - SMB - **Internal assembly showing position of Limit Switches**

2.3.2 Nuclear Electric Actuator Materials of Construction

The Limitorque actuators furnished for this project are all SMB electric actuators with HBC manual units. These actuators were specified for nuclear service per Bechtel Spec. No. 8031-G-11 and qualified per IEEE 323-1974, IEEE 344-1975, and IEEE 382-1972. These actuators incorporate use of special materials for nuclear service as listed below.

SMB seals - Viton
HBC seals - Viton
Lubricant - Exxon Nebula EP-0 or EP-1
Limitswitch gearing lubricant - Mobil 28
Exterior paint - Carbozinc 11
Limitswitch rotor & base - Fribrite
Torque switch - Fribrite
Limitswitch gear box - Bronze
Terminal strip - Marathon 300
Gaskets - Anchorite

2.3.3 Actuator And Valve Operation

2.3.3.1 Description

The actuators supplied for this project are not self contained units, and therefore, require customer supplied control devices. The customer supplied control system must be compatible with the actuators furnished to assure proper operation and prevent damage to the actuators. During operation, the valve must be able to perform several functions that include closing tightly, opening fully, and stopping at any intermediate position. The control of these functions is accomplished thru the use of a motor control unit used in conjunction with the torque and limit switches.

Since the motor controller is a customer provided unit its selection and design cannot be covered in this report.

During operation of the valve from any position to the full open position, the open limit switch will trip allowing the motor controller to stop the valve disc in the open position. Wired in series with the open limit switch is a torque switch(es) which protects the valve and actuator from damage due to overtorquing. This torque switch is adjusted and set to meet the design requirements of the valve and actuator. In the event of an open limit switch failure, the SMB would drive the HBC against it's mechanical stop. At that time, the torque would increase causing the torque switch to trip and stop the actuator. The mechanical stop is a protection means only and should not be used for normal operation.

When operating the valve from any position to the full closed position, two switches will trip unlike going to the full open position. The first switch that will trip is the closed limit switch. This switch will not stop the valve from closing, but serves another function. The closed limit switch is set to trip at approximately 3 degrees from the full closed position. When this switch trips. two of the contacts on the switch will parallel the open torque switch. While this switch is tripped, the open torque switch(es) is bypassed and allows full actuator output torque to be applied to the valve when opening the first 3 degrees. At the closed position, the valve disc seal will make contact with

the seat. As this occurs, the actuator output torque will increase until the closed torque switch(es) setting is exceeded and causes the closed torque switch(es) to trip allowing the motor controller to stop the valve disc in the closed position. In the closing direction, the torque switch(es) is the only electrical device used to stop the actuator. In the event the closed torque switch(es) failed, the actuator motor would go to locked rotor until any thermal overload device would trip out. If any of the above failure conditions were to occur, it would indicate that severe damage had occurred to the valve or actuator.

Shown in Figure 24 is a general view of the torque switch and limit switch assemblies.

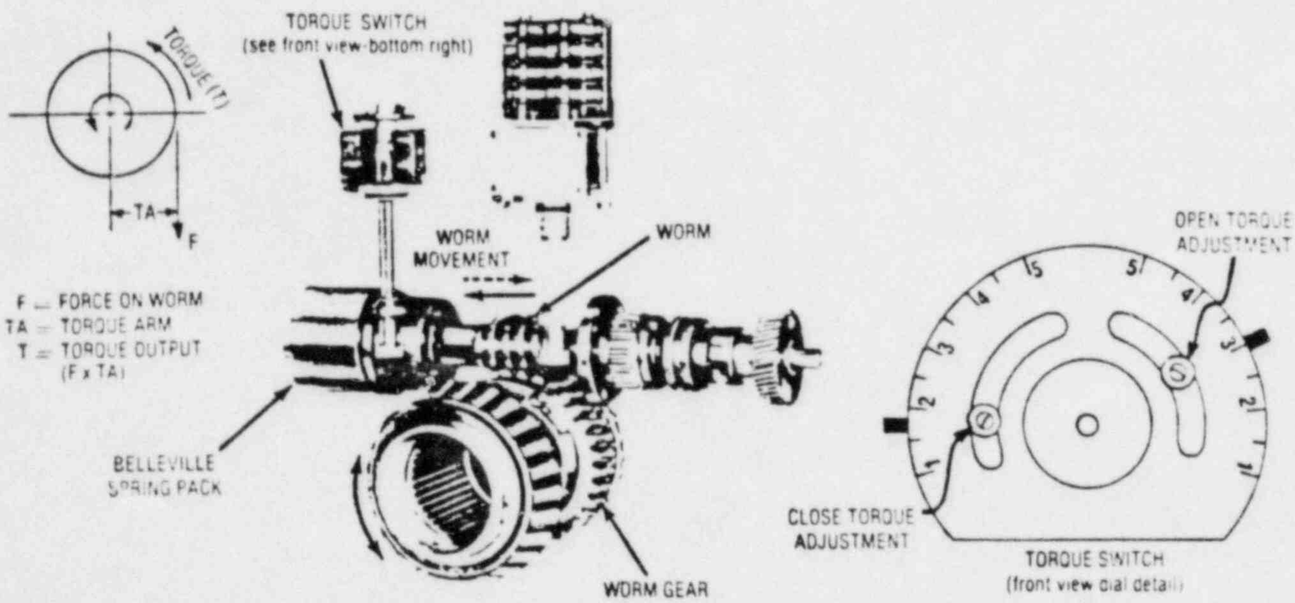


FIGURE 24 - TORQUE SWITCH ACTIVATION

Several of the SMB units supplied for this project do have double torque switches, however, when the switches are wired in parallel per the customer specification, they do not provide redundancy. When the switches are wired parallel, both switches must trip to stop the actuator output. Wiring the torque switches in this manner may lead to potential problems. If one switch undergoes a mechanical failure and cannot trip, even though the other switch may trip, it will not stop the actuator and locked rotor will occur.

Another item, which must be addressed in the SMB units, is the non-self locking worm gear sets in the SMB-2 units. Although most SMB units are self locking, the SMB-2 units could not be made self locking and still meet the required operating times. A problem may occur in the SMB-2 when the torque switch trips in the closed position. Since the torque switch(es) is driven by the worm shaft, when the worm shaft moves back after applying the required torque, the closed torque switch(es) may remake contact. When this occurs, the torque output of the HBC unit is not affected, since it has self locking gearing; therefore the valve will still be fully closed. It is possible, after the torque switch remakes contact, to electrically drive the HBC further closed. This could result in excessive torque output of the HBC, which could damage the actuator and/or the valve itself. It is therefore, necessary that the motor controller be designed with sufficient protection to prevent this from happening.

At any time when the actuator is not being operated electrically, the manual override may be engaged. To engage the handwheel, it is necessary to pull the declutch lever to the manual position. The handwheel may then be used to position the valve disc in the required position. The declutch lever will automatically disengage when any electrical actuation occurs.

2.3.3.2 Actuator Output Torques And Operating Times Design Requirements

The design requirement for operating times was 5 seconds or less. This was achieved through the selection of the proper gear ratios and motor RPM used in the actuators.

The output torque of the actuators is dependent on several variables that include motor sizing, gear ratios, and torque switch setting. Of these three variables, two are fixed by the sizing of the actuator and provide no adjustment. The torque switch setting is an adjustable variable that allows precise setting of the sensed output torque.

Due to the fast operating times required, one other effect must be considered and that is inertia. An article describing fast closing actuators is included in Appendix C that was prepared by the Limitorque Corp. Inertial effects continue after the torque switch(es) has tripped. Since the motor and gearing are moving at a relatively high speed with a given mass, the actuator will continue to increase torque output even though the electric power to the actuator has stopped.

The inertial effects cannot be determined empirically, since the system rigidity plays an important factor. A torque magnification of 1 to 3 times the torque at which the torque switch(es) trips, may be produced. It is therefore, necessary to either design the entire system to withstand this higher torque output, or adjust the torque switch(es) to a lower setting so the net torque produced will not exceed the required amount.

Design and measured actuator output torques are shown in the table below. Subsequent sections describe how these torques relates to safe operation of the subject valves.

TABLE 5

ELECTRIC ACTUATOR TORQUE CHARACTERISTICS

Valve Size (inches)	Design Torque in-lb	Actuator Model	Torque Switch Trip Torque (in-lb)	Torque Switch Setting Required	Torque Applied# To Valve (in-lb)	Torque To Fail Weakest* Component (Key)	Safety Factor*
4	2,112	SMB-00-10-H1BC		1.0		7,594	+
6	7,800	SMB-00-10-H2BC		1.5		27,585	+
18	63,120	SMB-1-60-H5BC	34,000	1.0	86,000	207,563	2.41
24	135,000	SMB-2-60-H5BC		1.0		362,496	+

*Based on key machanical properties measured by test. Valve operator compatibility forms supplied separately indicate failure point for key based on mechanical properties lower than actually used.

#At normal voltage 460 AC (higher than design or torque switch trip due to inertia).

+Will be furnished at a later date

2.3.3.3 Operating Time

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

TABLE 6

Mark No. of Valve	Valve Size (inch)	Limitorque Actuator Model No.	Opening Time Sec. Voltage Min.	Opening Time Sec. Voltage Norm.	Closing Time Sec. Voltage Min.	Closing Time Sec. Voltage Norm.
MO-57-161	4	SMB-00-10-H1BC	*	*	*	*
MO-57-163	4	SMB-00-10-H1BC	*	*	*	*
MO-57-109	6	SMB-00-10-H2BC	*	*	*	*
MO-57-162	6	SMB-00-10-H2BC	*	*	*	*
MO-57-164	6	SMB-00-10-H2BC	*	*	*	*
MO-57-112	18	SMB-1-60-H5BC	*	*	*	*
MO-57-115	24	SMB-2-60-H5BC	*	*	*	*
MO-57-135	24	SMB-2-60-H5BC	*	*	*	*
MO-57-147	24	SMB-2-60-H5BC	*	*	*	*

*Will be furnished at a later date

3.0 VALVE OPERATING AND INSTALLATION REQUIREMENTS

3.1 Valve Operating Conditions

The normal and accident operating conditions for the subject valves are taken from Bechtel Power Corp. Specification 8031-P-144, Rev. 1, Appendix 17, Paragraph 1A and 3B. Leakage requirements are per spec. paragraph 10.0. This data is presented in summarized form in Tables 7 thru 9.

TABLE 7

Seismic Loadings For All Valves

Condition	Loading Condition	Acceleration Values (g) Horiz.	Vert.
Normal operation	gravity load only (no seismic acceleration)	0.0	1.0
Upset Emergency Faulted	All loads per 8031-P-144 Appendix 17, Table C-3 (worst case basis)	4.5	4.5

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

TABLE 8

Pressure Differentials Applied to Valves

VALVE SIZE	VALVE MARK NO.	NORMAL OPERATING PRESSURE (PSIG)	OPER. TEMP. RANGE (^{o}F)	DESIGN DIFFERENTIAL PRESSURE (PSIG)	NORMAL FLOW SCFM	FAILURE MODE
4"	MO-57-161,163	1.5	65-340	55	150	in position
6"	MO-57-109,162,164	1.5	65-340	55	150	in position
6"	AO-57-121,131	1.5	65-340	55	150	closed
18"	MO-57-112	1.5	65-340	55	4400	in position
18"	AO-57-104	1.5	65-340	55	4400	closed
24"	MO-57-115,135,147	1.5	65-340	55	6600	in position
24"	AO-57-114,123,124	1.5	65-340	55	6600	closed

TABLE 9

Allowed Seat Leakage Rates

(Per Spec at 5, 25, and 55 PSIG Pneumatic)

VALVE SIZE	VALVE MARK NO.	ALLOWED LEAKAGE cc/min
4"	MO-57-161,163	.133
6"	MO-57-109,162,164	.20
6"	AO-57-121,131	.20
18"	MO-57-112	.60
18"	AO-57-104	.60
24"	MO-57-115,135,147	.80
24"	AO-57-114,123,124	.80

3.2 Valve Installation Configurations

In addition to the pressure and flow conditions specified in 3.0, the valve performance is effected by the as installed orientation. Upstream and downstream, tees, elbows, reducers, and other valves can effect the aerodynamic torque characteristics of butterfly valves. These effects are discussed in Section 5.0 The installed configurations for the subject valves as derived from Bechtel prints are summarized in Figures 25 thru 35. with appropriate print references.

NOTE: All valve discs in the figures are shown in the partially open (approximately 20° off of seat) position.

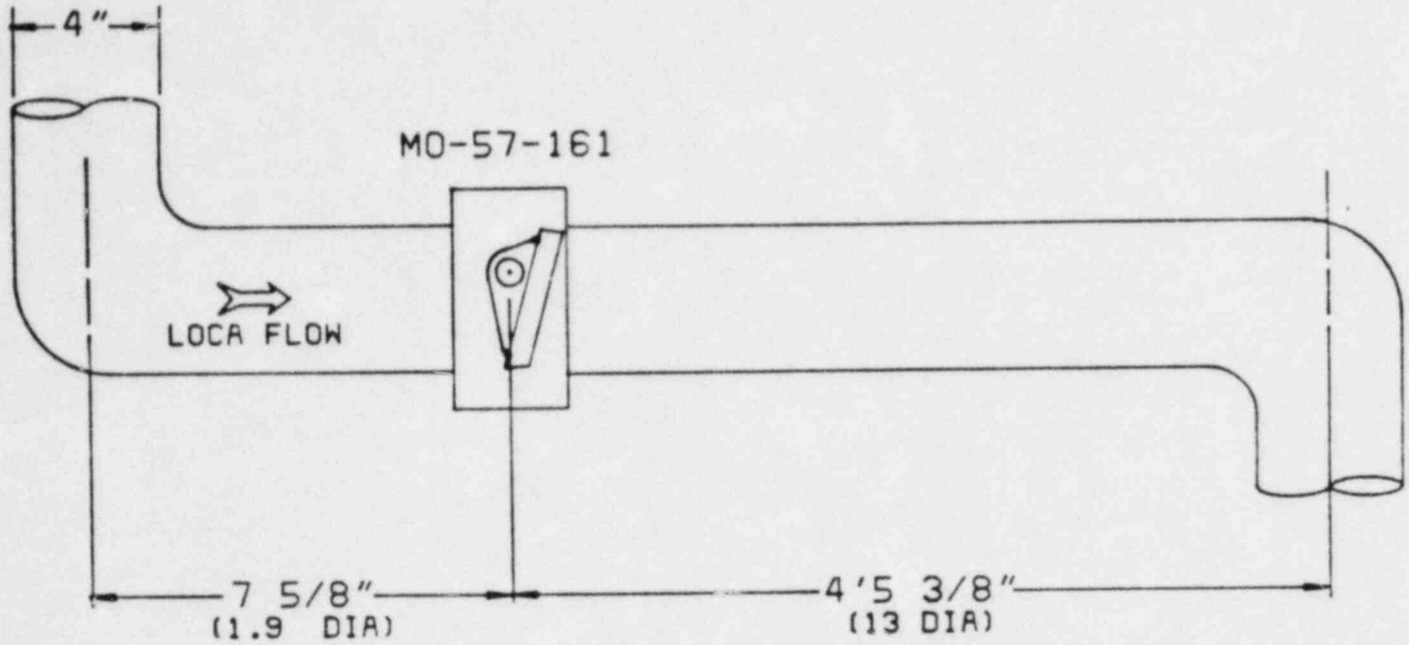


FIGURE 25: VALVE MO-57-161 AS INSTALLED PIPING CONFIGURATION PER DWG. HBB-127-2 REV.10. DISC ORIENTATION SHOWN IS WORST CASE, NOT NECESSARILY AS INSTALLED.

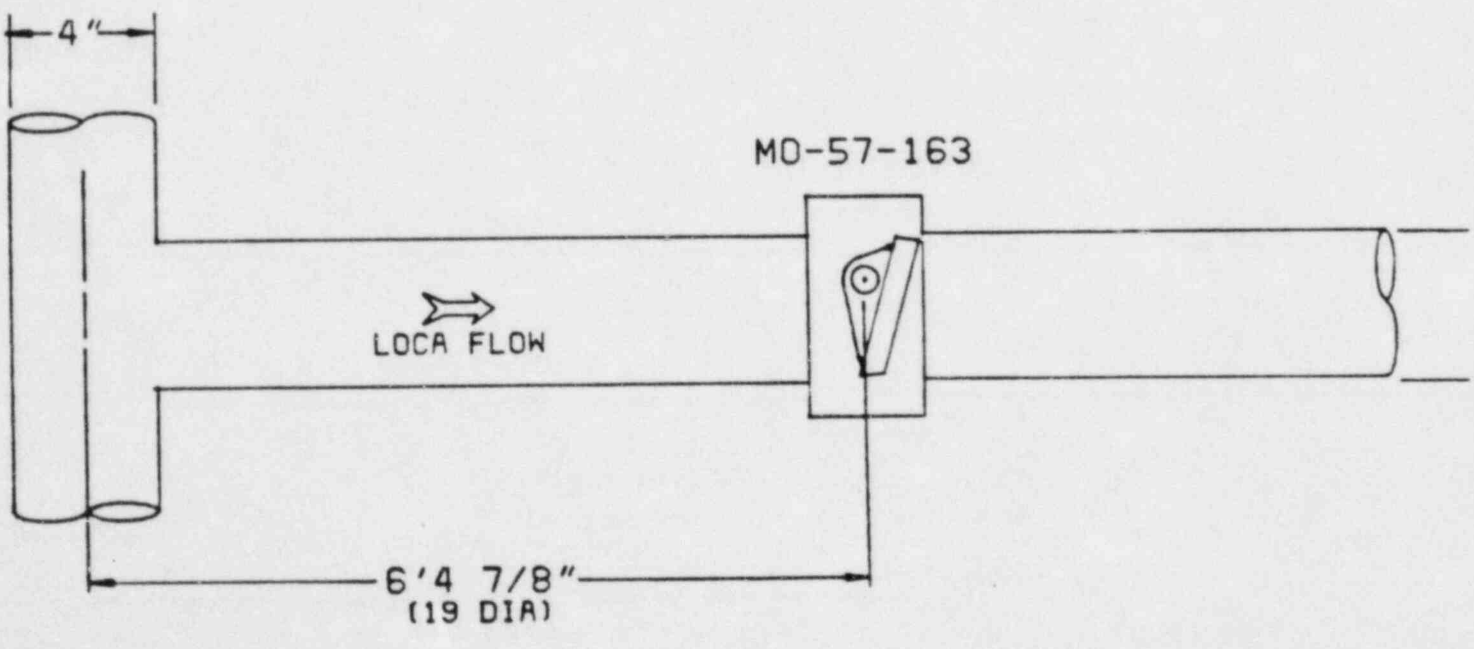


FIGURE 26: VALVE MO-57-163 AS INSTALLED PIPING CONFIGURATION PER DWG. HBB-124-2 REV.17. DISC ORIENTATION SHOWN IS WORST CASE, NOT NECESSARILY AS INSTALLED.

DRAWINGS FOR QUALIFICATION REPORT FOR 82-2053(N)

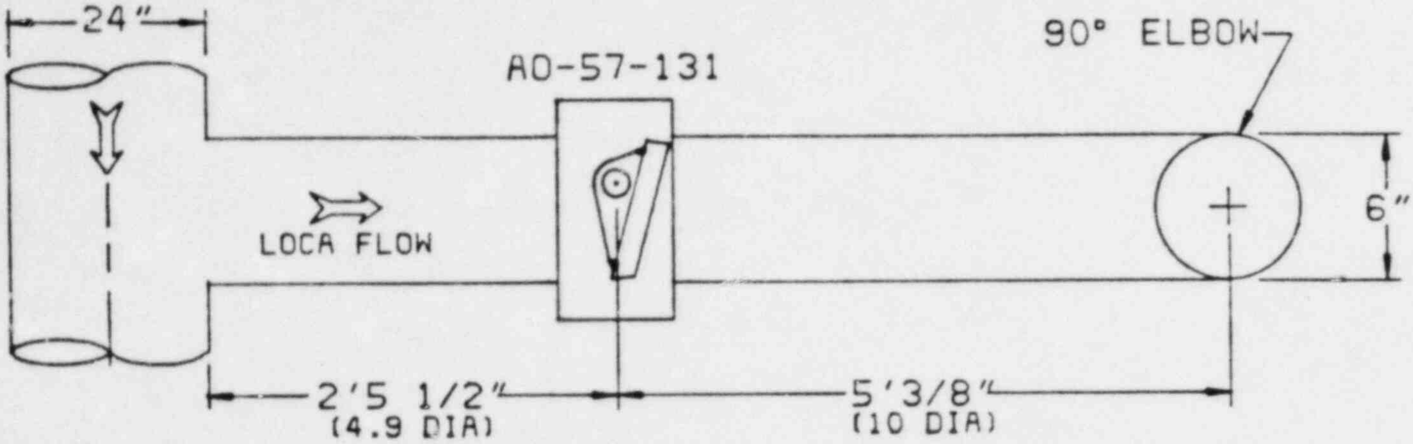


FIGURE 27: VALVE AO-57-131 AS INSTALLED PIPING CONFIGURATION PER DWG. HBB-124-1 REV.7. DISC ORIENTATION SHOWN IS WORST CASE, NOT NECESSARILY AS INSTALLED.

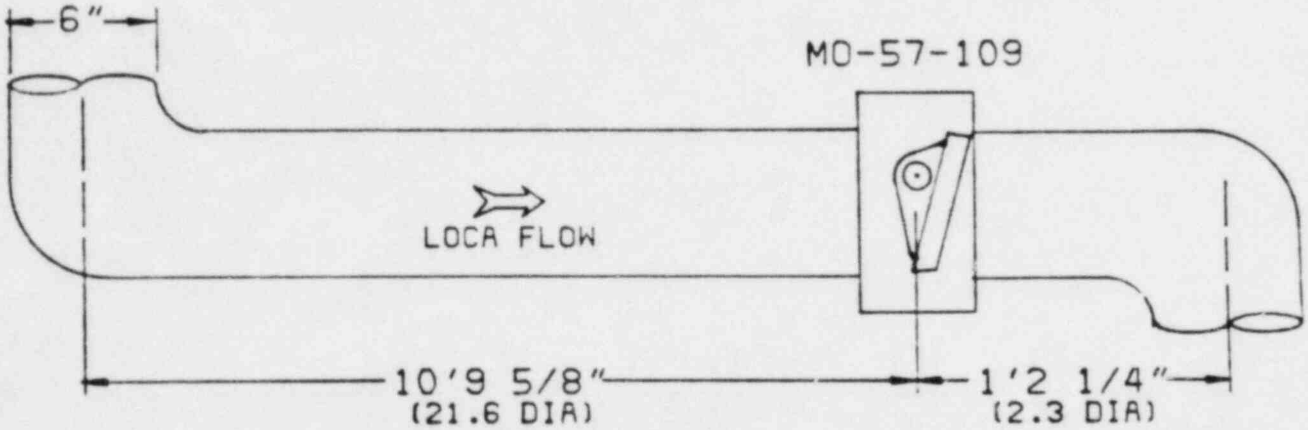


FIGURE 28: VALVE MO-57-109 AS INSTALLED PIPING CONFIGURATION PER DWG. HBB-125-1 REV.7. DISC ORIENTATION SHOWN IS WORST CASE, NOT NECESSARILY AS INSTALLED.

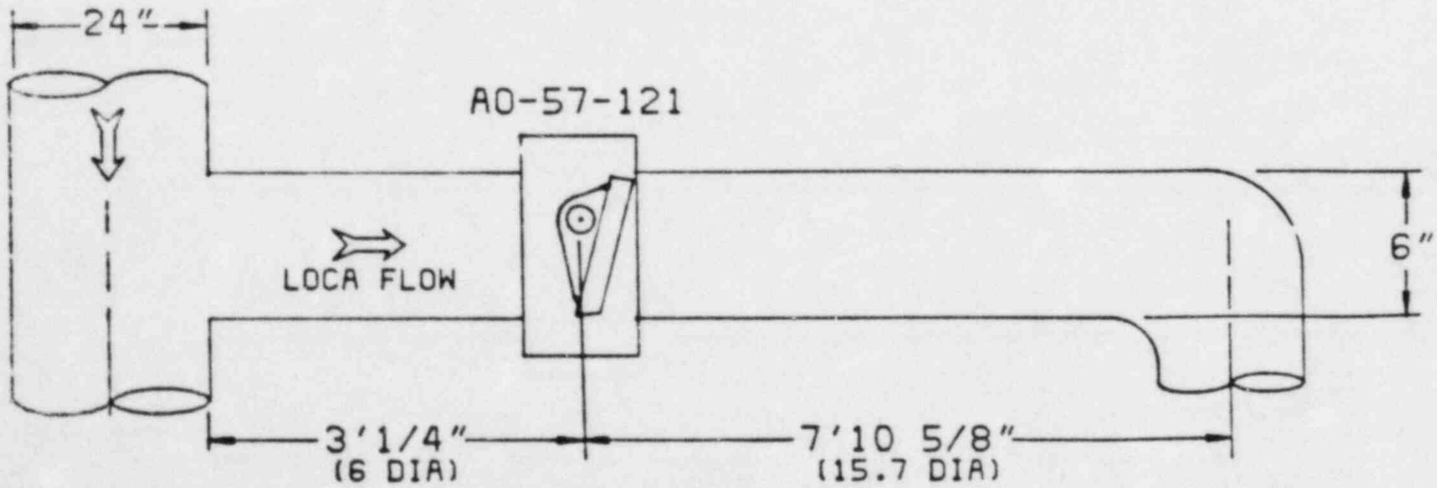


FIGURE 29: VALVE AO-57-121 AS INSTALLED PIPING CONFIGURATION PER DWG. HBB-125-1 REV.7. DISC ORIENTATION SHOWN IS WORST CASE, NOT NECESSARILY AS INSTALLED.

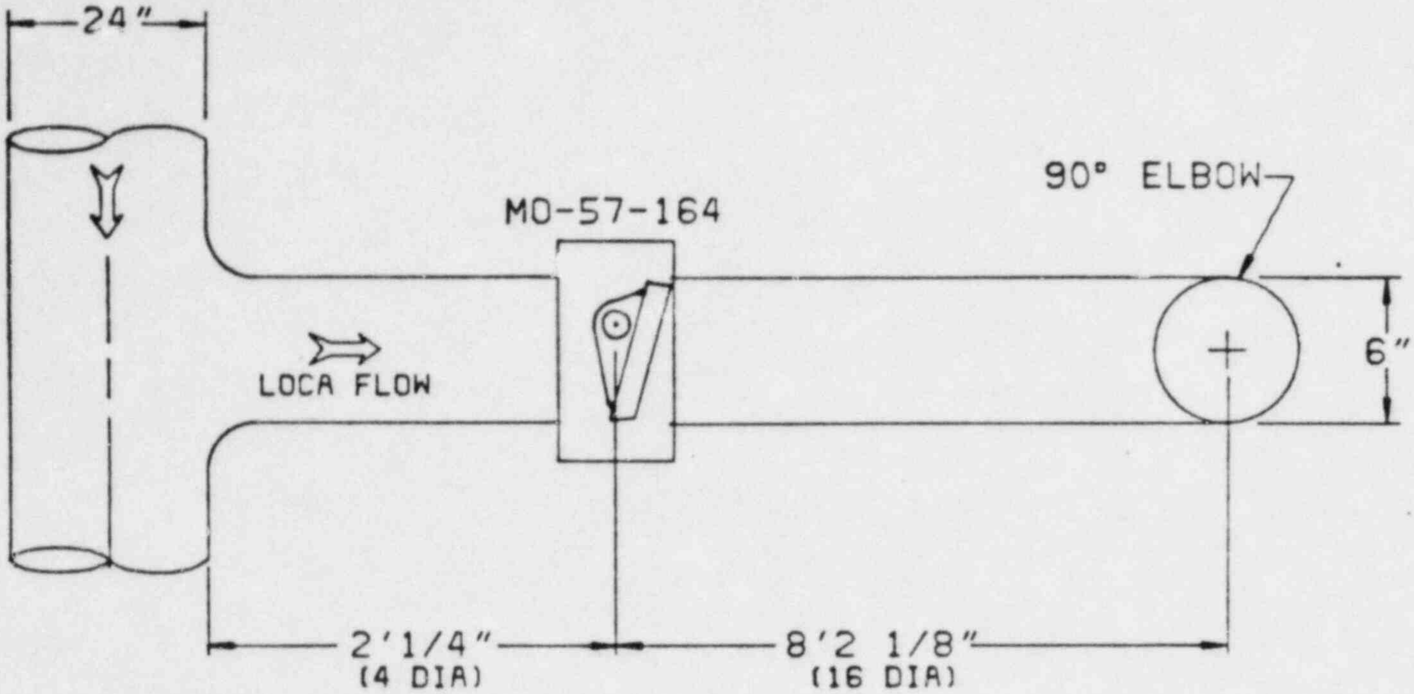


FIGURE 30: VALVE MO-57-164 AS INSTALLED PIPING CONFIGURATION PER DWG. HBB-126-3 REV.7. DISC ORIENTATION SHOWN IS WORST CASE, NOT NECESSARILY AS INSTALLED.

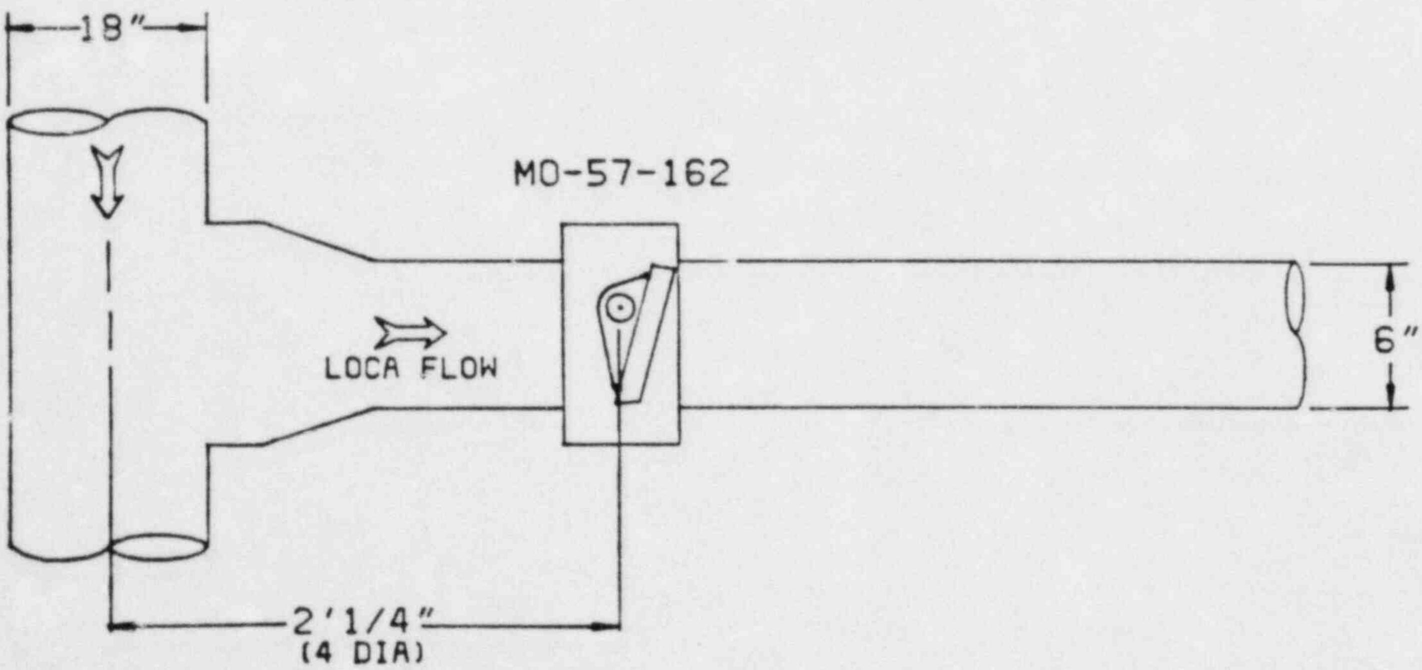


FIGURE 31: VALVE MO-57-162 AS INSTALLED PIPING CONFIGURATION PER DWG. HBB-128-3 REV.5. DISC ORIENTATION SHOWN IS WORST CASE, NOT NECESSARILY AS INSTALLED.

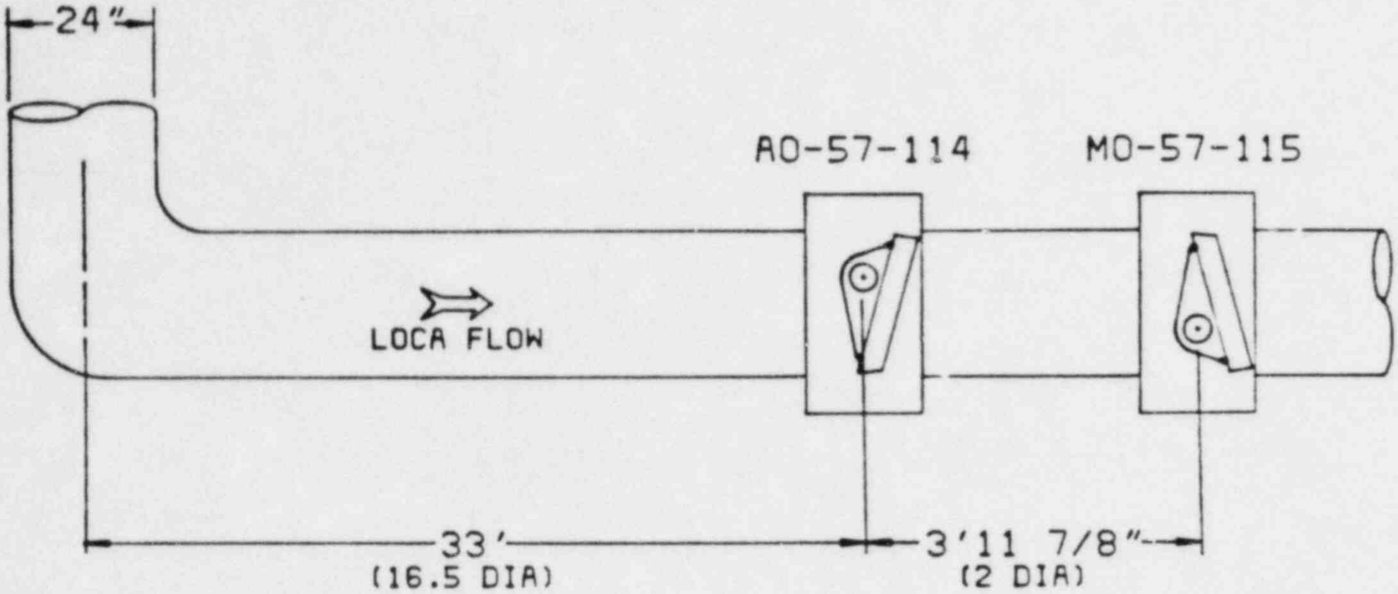


FIGURE 32 : VALVES AO-57-114 & MO-57-115 AS INSTALLED PIPING CONFIGURATION PER DWG. HBB-127-1 REV.12. DISC ORIENTATIONS SHOWN ARE WORST CASE, NOT NECESSARILY AS INSTALLED.

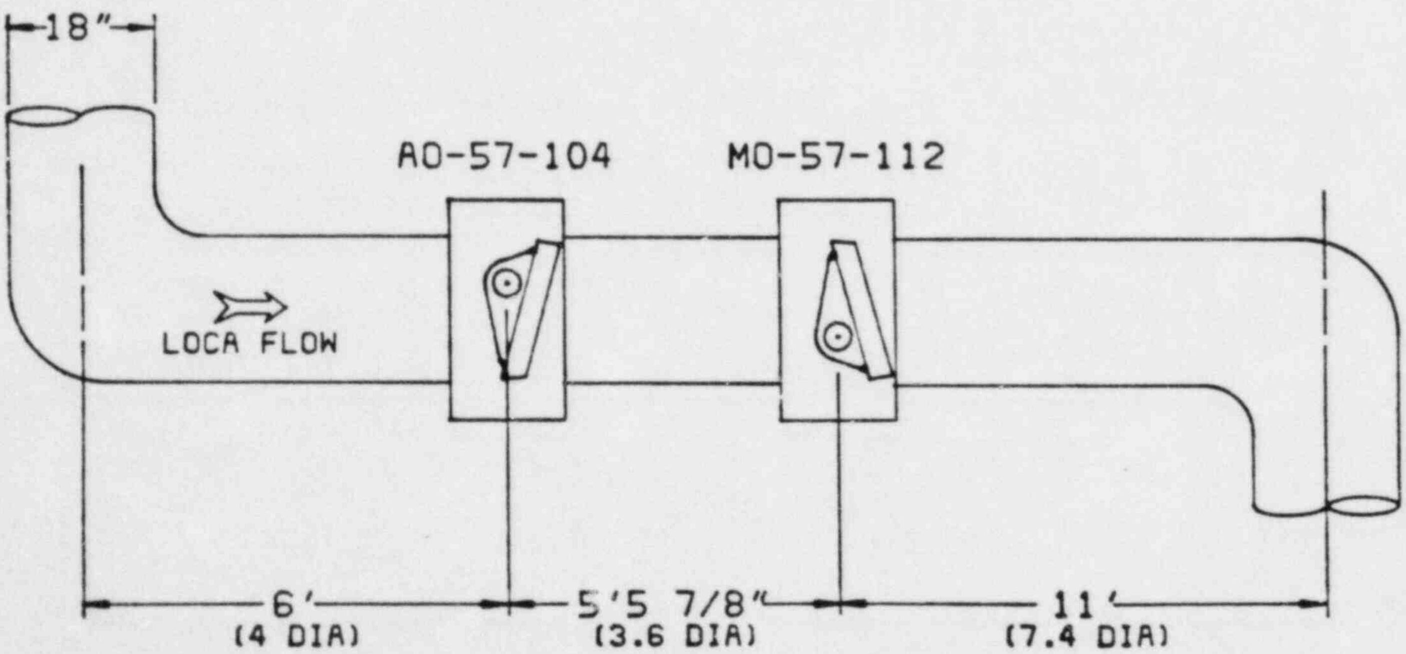


FIGURE 33 : VALVES AO-57-104 & MO-57-112 AS INSTALLED PIPING CONFIGURATION PER DWG. HBB-128-1 REV.11. DISC ORIENTATIONS SHOWN ARE WORST CASE, NOT NECESSARILY AS INSTALLED.

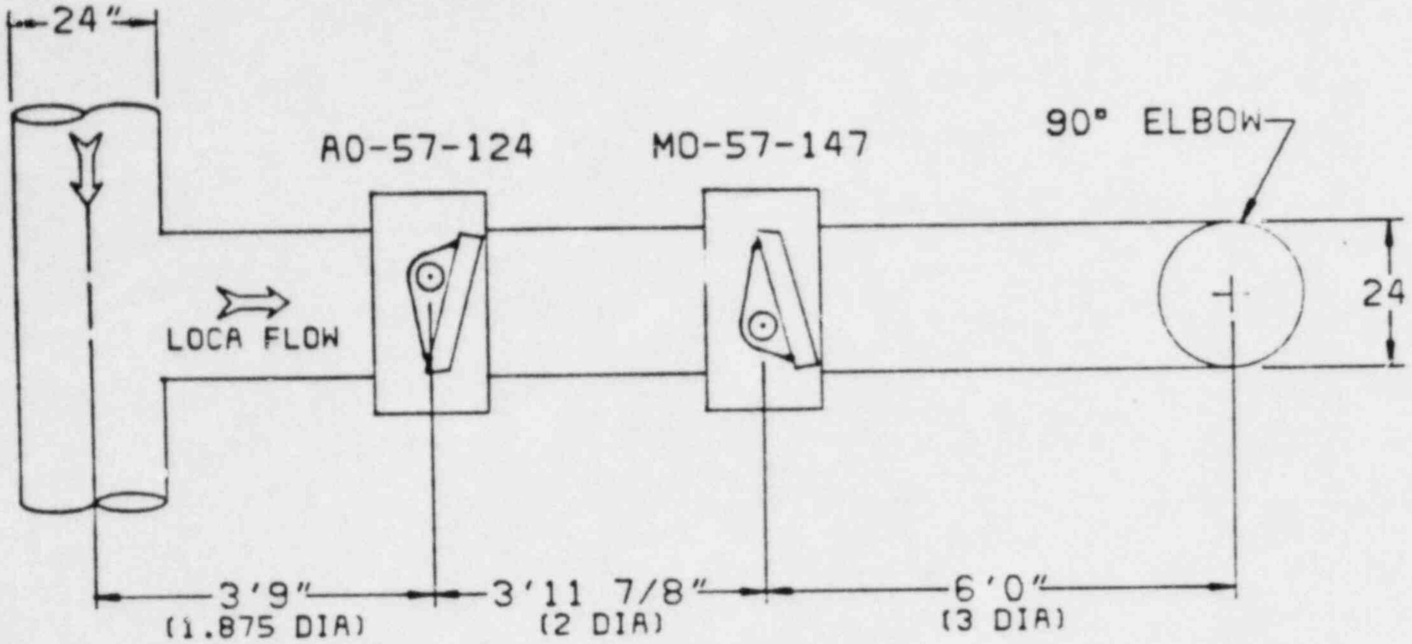


FIGURE 34: VALVES AO-57-124 & MO-57-147 AS INSTALLED PIPING CONFIGURATION PER DWG. HBB-126-1 REV.7. DISC ORIENTATIONS SHOWN ARE WORST CASE, NOT NECESSARILY AS INSTALLED.

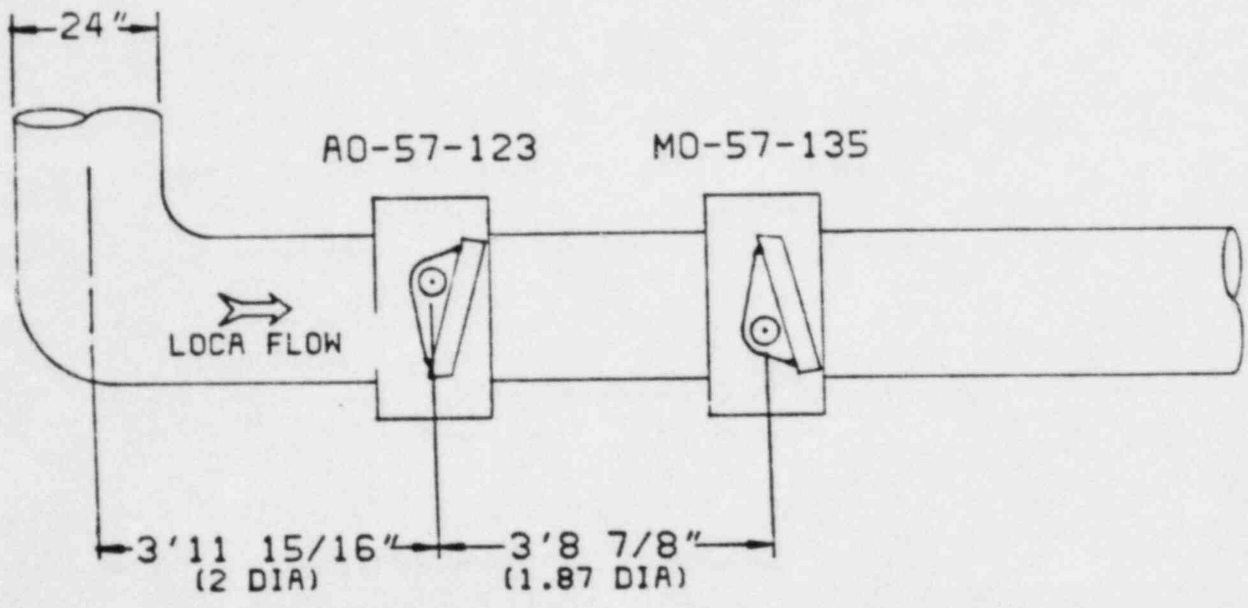


FIGURE 35: VALVES AO-57-123 & MO-57-135 AS INSTALLED PIPING CONFIGURATION PER DWG. HBB-124-2 REV.17. DISC ORIENTATIONS SHOWN ARE WORST CASE, NOT NECESSARILY AS INSTALLED.

4.0 VALVE STRUCTURAL INTEGRITY UNDER SEISMIC AND OPERATIONAL LOADINGS

Operability of the subject valves has been verified by a combination of testing and analysis in accord with Bechtel Power Corp. Specification 8031-P-144, Rev.1. Separate reports have been prepared and provided demonstrating suitability of valve components and the assembly. A listing is provided in the references (7.0) at the end of the report. This section summaries the results of such tests and analyses in meeting the conditions as presented in Section 3.0.

4.1 Valve Frequency And Stress Analysis

Valve frequency and stress analysis was performed by Patel Engineers, Huntsville, Alabama for each valve size. The analysis was made using the ANSYS finite element computer program developed by Swanson Analysis System, Inc., Houston, Pa. This public domain program has had a sufficient history of use to justify its applicability and validity. The analyses were made for the seismic conditions stated in Section 3.0 and for pressure and temperature as specified in Table 11. The obtained lowest resonant frequencies for the valve assembly are presented in Table 10. The lowest resonant frequency as determined by a conservative analysis approach is 85 Hz. For stress analysis, allowable stresses were in accord with ASME Section III requirements and Table C-3 of 8031-P-144 Appendix 17. Table 13 thru 16

summarize the maximum stresses in the valve elements and how these relate to allowed values.

4.2 Bettis Actuator Resonant Frequency Test

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

4.3 Asco Solenoid Valve Resonant Frequency Test

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

*National Technical System

The test demonstrated no major resonances between one and 130 Hz. One orientation showed a system resonance between 130 and 140 Hz which was outside of the required operability range. The sine beat test which consisted of 6260 beats per orientation at 5 to 100 Hz and accelerations of 2.0 to 11.0 g (within test table acceleration limits) showed the solenoid valve to be operable before, during, and after the test. No detectable leakage occurred during any phase of the tests. Although these tests were not performed for the subject contract (no report is submitted for review or approval), the report is available for review at Clow's Westmont, Illinois facility.

4.4 Static Load Test During Simulated LOCA Flow

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

4.5 Bettis Actuator Seismic/Hydrodynamic Operability Test

A Dynamic Test was performed on a NT312-SR5 and NT820-SR4 actuator to demonstrate operability under anticipated loadings which may be encountered in service. The test was performed at NTS, Saugus, Ca., in accord with NTS Test Procedure 528-0951. The test demonstrated the units would operate as required before, during, and after the test. This information is supplied by a report under separate cover (see Reference 7.0).

TABLE 10

Lowest Valve Resonant Frequencies
(Per Analysis)

VALVE SIZE	TYPE	FREQUENCIES (Hz)
4	MO	87
6	MO	107
6	AO	105
18	MO	139
18	AO	131
24	MO	85*
24	AO	86

*Analysis assumed use of SMB-3 electric actuator which is a worse case than the SMB-2 shipped with the valves.

TABLE 11

Condition Applied For Stress Analysis

Valve Size	Valve Mark Nos.	Body Design Pressure (PSIG)	Disc Differential Pressure (PSI)	Design Temp. °F	Design Seating Torque (in-lb)
4	MO-57-161,163	285	65	340	2,112
6	MO-57-109,162,164 AO-57-121,131	285	65	340	7,800
18	MO-57-112 AO-57-104	285	65	340	63,300
24	MO-57-115,135,147 AO-57-114,123,124	285	65	340	135,000

TABLE 12

Allowed Stress

Applicable Valves	Condition	Membrane Stress Limit	Membrane + Bending Stress Limit
All valves	Upset	1.1 S	1.65 S
	Emergency	1.5 S	1.8 S
	Faulted	2.0 S	2.40 S

S = Stress allowed per ASME, Sect. III, Tables 1-7.1 thru 1-7.3 and Appendix XVII 2460 (as applicable)

4

Table 13
4" Valve
Summary of Allowable Stresses (worst case)

LOCATION	MATERIAL	ALLOWABLE STRESS (psi) (PER ASME SECTION III, TABLES I-7.1 THROUGH I-7.3)	STRESS VALUE (psi)	ELEMENT	STRESS RATIO
Valve Body	SA 516 GR. 70	17500	4109	61	0.24
Disc	SA 516 GR. 70	17500	3309	191	0.19
Drive Shaft	SA 564 Type 630 H-1045	34550	28020	202	0.81
Operator Adapter Plate	SA 516 GR. 70	17500	1847	212	0.11
Adapter Plate Bolts	SA 193 GR. B7	25000	9284 σ_N 6198 τ	N/A	0.08*
Operator/Adapter Bolts	SA 193 GR. B7	25000	5616 σ_N 4871 τ	N/A	0.04*
Cover Plate	SA 516 GR. 70	(1.5)(17500) =26250	3258	N/A	0.19
Cover Plate Bolts	SA 193 GR. B7	25000	3286 σ_N 18 τ	N/A	0.003*

*Per ASM ection III, Appendix XVII, Subsubarticle 2460.

6.

Table 14
6" Valve
Summary of Allowable Stresses (worst case)

LOCATION	MATERIAL	ALLOWABLE STRESS (psi) (PER ASME SECTION III, TABLES I-7.1 THROUGH I-7.3)	STRESS VALUE (psi)	ELEMENT	STRESS RATIO
Valve Body	SA 516 GR.70	17500	3836	81	0.22
Disc	SA 516 GR.70	17500	8817	202	0.50
Drive Shaft	SA 564 Type 630 H-1045	34550	23038	216	0.67
Operator Adapter Plate	SA 516 GR.70	17500	4377	226	0.25
Adapter Plate Bolts	SA 193 GR.B7	25000	$15090\sigma_N$ 7764τ	N/A	0.15*
Operator/Adapter Bolts (Hybrid Worst Case Assembly)	SA 193 GR.B7	25000	$17284\sigma_N$ 11194τ	N/A	0.27*
Cover Plate	SA 516 GR. 70	(1.5)(17500) =26250	6613	N/A	0.39
Cover Plate Bolts	SA 193 GR.B7	25000	$5490\sigma_N$ 21τ	N/A	0.01*

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

Table 15
18" Valve
Summary of Allowable Stresses (worst case)

LOCATION	MATERIAL	ALLOWABLE STRESS (psi) (PER ASME SECTION III, TABLES I-7.1 THROUGH I-7.3)	STRESS VALUE (psi)	ELEMENT	STRESS RATIO
Valve Body	SA 516 GR.70	17500	2156	81	0.12
Disc	SA 516 GR.70	17500	5043	214	0.29
Drive Shaft	SA 564 Type 630 H-1045	34550	22168	232	0.64
Operator Adapter Plate	SA 516 GR.70	17500	3053	261	0.18
Adapter Plate Bolts	SA 193 GR.B7	25000	9839 6645	N/A	0.09*
Operator/Adapter Bolts (Hybrid Worst Case Assembly)	SR 193 GR.B7	25000	10479 11806	N/A	0.24*
Cover Plate	SA 516 GR. 70	(1.5)(17500) =26250	17641	N/A	0.68
Cover Plate Bolts	SA 193 GR.B7	25000	8072 26	N/A	0.02*

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

Table 16
24" Valve
Summary of Allowable Stresses (worst case)

LOCATION	MATERIAL	ALLOWABLE STRESS (psi) (PER ASME SECTION III, TABLES I-7.1 THROUGH I-7.3)	STRESS VALUE (psi)	ELEMENT	STRESS RATIO
Valve Body	SA 516 GR. 70	17500	4089	106	0.23
Disc	SA 516 GR. 70	17500	6930	244	0.40
Drive Shaft	SA 564 Type 630 H-1045	34550	27263	267	0.79
Operator Adapter Plate	SA 516 GR. 70	17500	5891	296	0.34
Adapter Plate Bolts	SA 193 GR. B7	25000	$17215\sigma_N$ 8884τ	N/A	0.19*
Operator/Adapter Bolts (Limitorque)	SR 193 GR. B7	25000	$15566\sigma_N$ 19076τ	N/A	0.61*
Operator/Adapter Bolts (Bettis)	SA 193 GR. B7	25000	$16141\sigma_N$ 18276τ	N/A	0.57*
Cover Plate	SA 516 GR. 70	(1.5)(17500) =26250	24052	N/A	0.92
Cover Plate Bolts	SA 193 GR. B7	25000	$7000\sigma_N$ 102τ	N/A	0.01*

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

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 17) 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^o 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^o (¼ turn) valve design. Therefore, at small opening angles (0^o to 20^o) there are some differences in torque. For angles over this amount, the aerodynamics are the same. Also, at small angles the torque approaches the value of the pressure area torque (as explained in Section 2.1.3) thus, differences between the two designs are not significant. With reasonable similarity between the test models and the full size valves, the data may be used to predict torque characteristics for produced valves.

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

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

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

where

T = predicted aerodynamic torque (in lb)

C_T = torque coefficient developed in model tests

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

D_V = nominal valve diameter (in.)

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

Table 17 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 18 shows a comparison between the provided size valves and the interpolated sizes.

Linear interpolation was used to predict torque characteristics in Clow Program CVAP, thus a similar interpolation of sizes is applicable for size comparison purposes. It can be seen in the table that very good (less than 8% deviation) correlation was obtained for torque critical items for the 18" and 24" valves. For the 6" valves correlation is good (less than 9% deviation) for all critical dimensions other than disc thickness. From test data, greater disc thickness would reduce the potential for torques tending to resist valve closure. For the 4" valves interpolated dimensions are larger than production dimensions for the I.D., A_2, and K_2. This would lead to higher calculated torques than would actually be experienced. The offset E used for calculation is only 69% of the production size which would indicate a lower torque than in service. However, service torques would not be higher than 44% of those indicated in Tables 27 and 28.

TABLE 17

Test Valve Scaled Sizes (Critical Elements)

ELEMENT	12" Full Size	12" Model Size	24" Full Size	24" Model Size	48" Full Size	48" Model Size	96" Full Size	96" 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 off[illegible]om body centerline

* Ear is element welded to disc which shaft is [illegible]ced 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 18

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

ELEMENTS	VALVE SIZES 4"		6"		18"		24"	
	Size	Ratio	Size	Ratio	Size	Ratio	Size	Ratio
*I.D.	4.026	1.20	6.07	1.09	16.88	1.02	22.62	1.0
*A_2	3.289	1.15	5.33	1.07	16.08	1.02	21.71	1.01
*K_2	3.40	1.12	5.20	1.05	15.70	1.01	21.25	.98
Shaft Dia.	.75	1.17	1.25	1.02	2.5	1.10	3.0	1.08
Shaft ℄ to Seal ℄ ,L	.75	1.07	1.0	1.16	2.19	1.07	2.50	1.08
*Disc Thickness	.625	.99	.625	1.43	1.63	1.04	1.75	1.07
*Shaft Offset E	.88	.69	.88	.96	.95	1.08	.69	1.07
Shaft Offset LC	.91	NA	.91	NA	1.00	NA	.76	NA
Ear Width	1.0	.88	1.5	.85	2.5	1.10	3.50	.93
Ear Height	1.06	1.25	1.56	1.23	3.25	1.27	3.75	1.30

*Elements considered important to torque characteristics

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

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

All dimensions in inches

5.1.2 Tests With An Upstream Elbow

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

The mitered elbow produces the greatest separated flow region at the inside of the turn and biases the flow to the outside corner to a maximum (see photo from water table study Figure 37).[1] Further, the mitered elbow produces flow patterns more severe than expected for tee flow (see Figures 37 and 40). The testing performed has given added evidence in support of this assumption. (See report reference 7.0 C-3) Flow around the corner produces a lower local pressure around the inside of the turn and higher local pressure to the outside. This will oppose closure for geometry 1 (see Figure 41) and aid closure for geometry 2.

Based on these considerations, models of a 12", 24", and 48" valve (per Table 17) were tested for torque characteristics. All valve models were tested for geometries 1, 2, and 3 at 2 diameters downstream from the mitered elbow. In addition, the

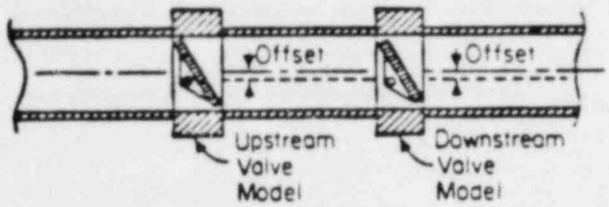
[1] See reference 7.0 E-3 .

12" model was tested at 4 and 8 diameters downstream. The test showed the greatest variation of torque from that obtained for straight-line flow occurred at 2 diameters downstream from the elbow. Differences due to valve orientation were small at 4 diameters downstream and were just detectable at 8 diameters downstream.

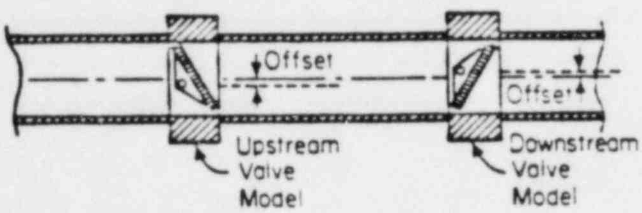
For the subject job some valves are installed closer than 2 diameters from an elbow. Since the mitered elbow used in the model tests is a worst case condition and radius type elbows are typically used for in plant installation, use of the test data for 2 diameters downstream for determining installed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

5.1.3 Tests With Two Valves In Series

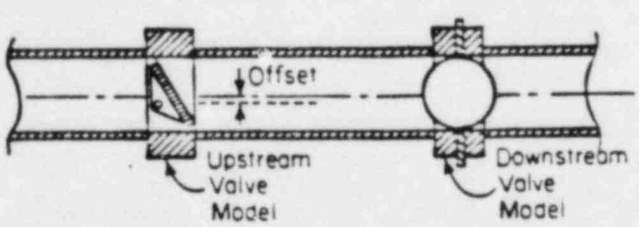
When two valves are installed in series in a pipe run at a relatively close distance (less than 8 diameters) some level of interaction will occur. Several different orientations of the two valves relative to one another are possible as shown in Figure 36. Model tests performed to determine aerodynamic torque characteristics, indicate that orientation 2 with the upstream valve failed (stuck) at 60° open and the downstream valve at full open would represent a worst case condition (highest torques resisting downstream valve closure). These model tests are more fully described in a separate report indicated in the references (Section 7.0).



Orientation 1



Orientation 2



Orientation 3

FIGURE 36 POSSIBLE ORIENTATION OF TWO CLOW VALVES INSTALLED IN SERIES

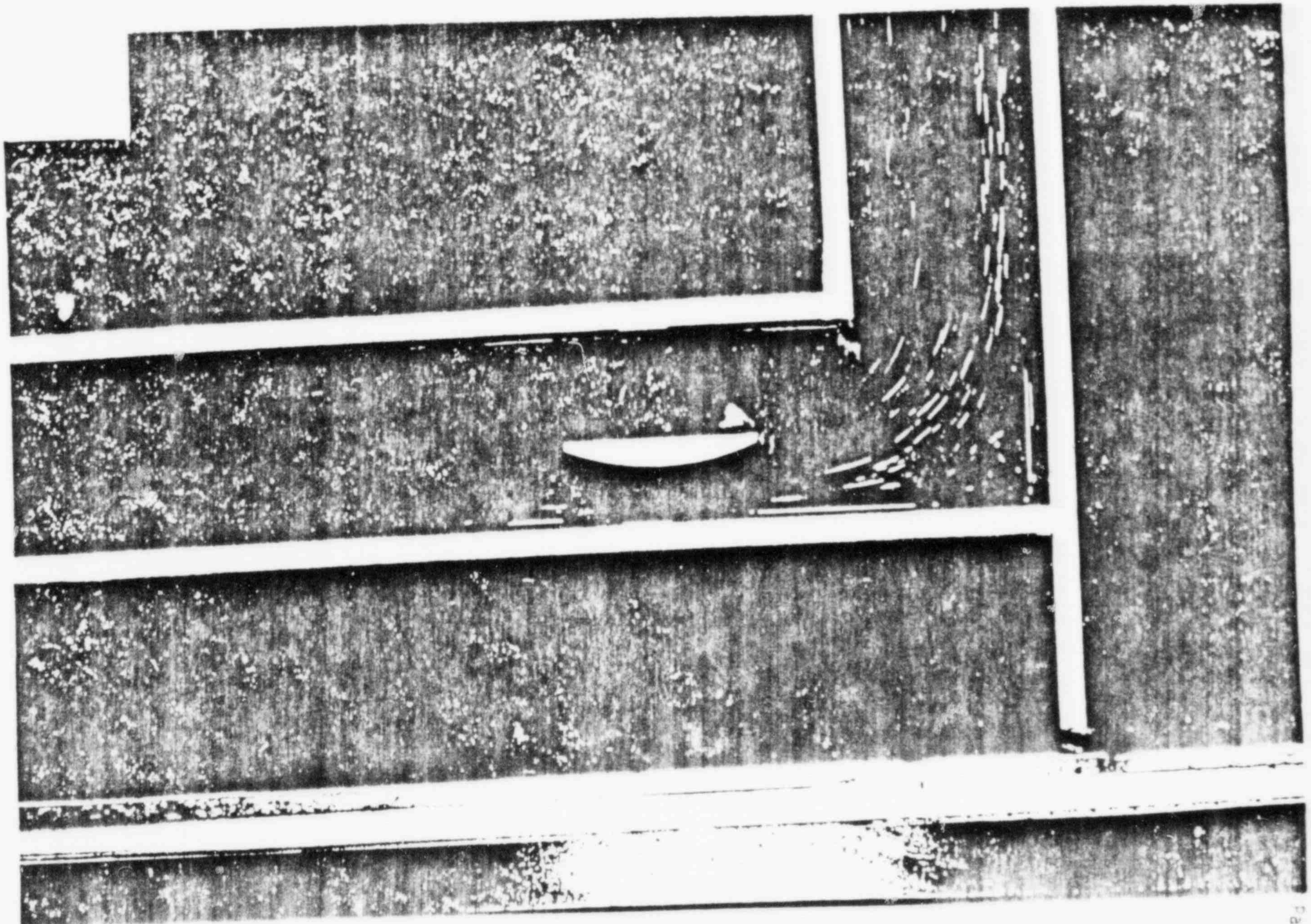


FIGURE 37 - Water Table Study of Choked Flow Pattern With Disc Full Open (90°)

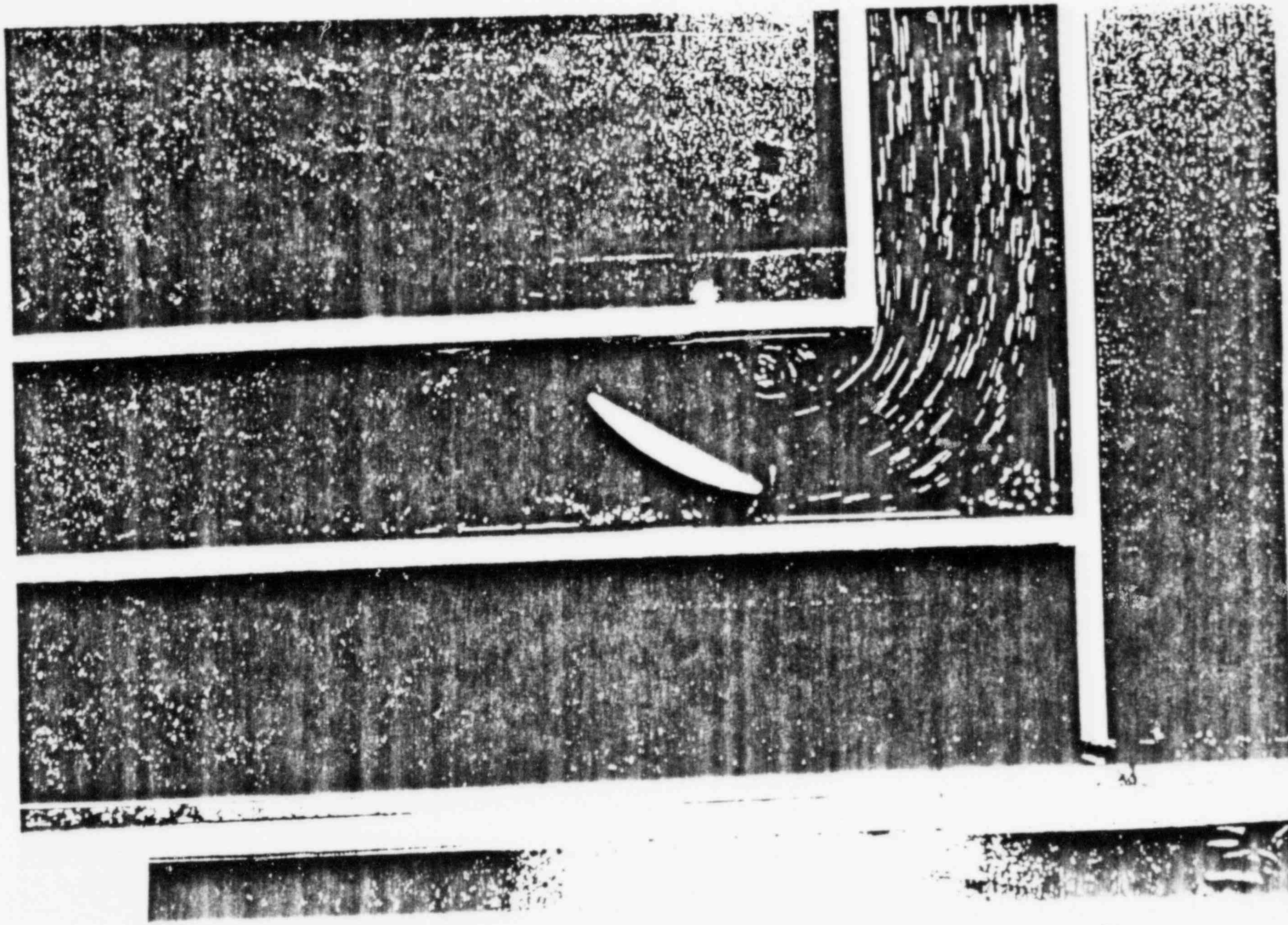


FIGURE 38 - Water Table Study of Choked Flow Pattern With Disc Partially Open (60^{o})

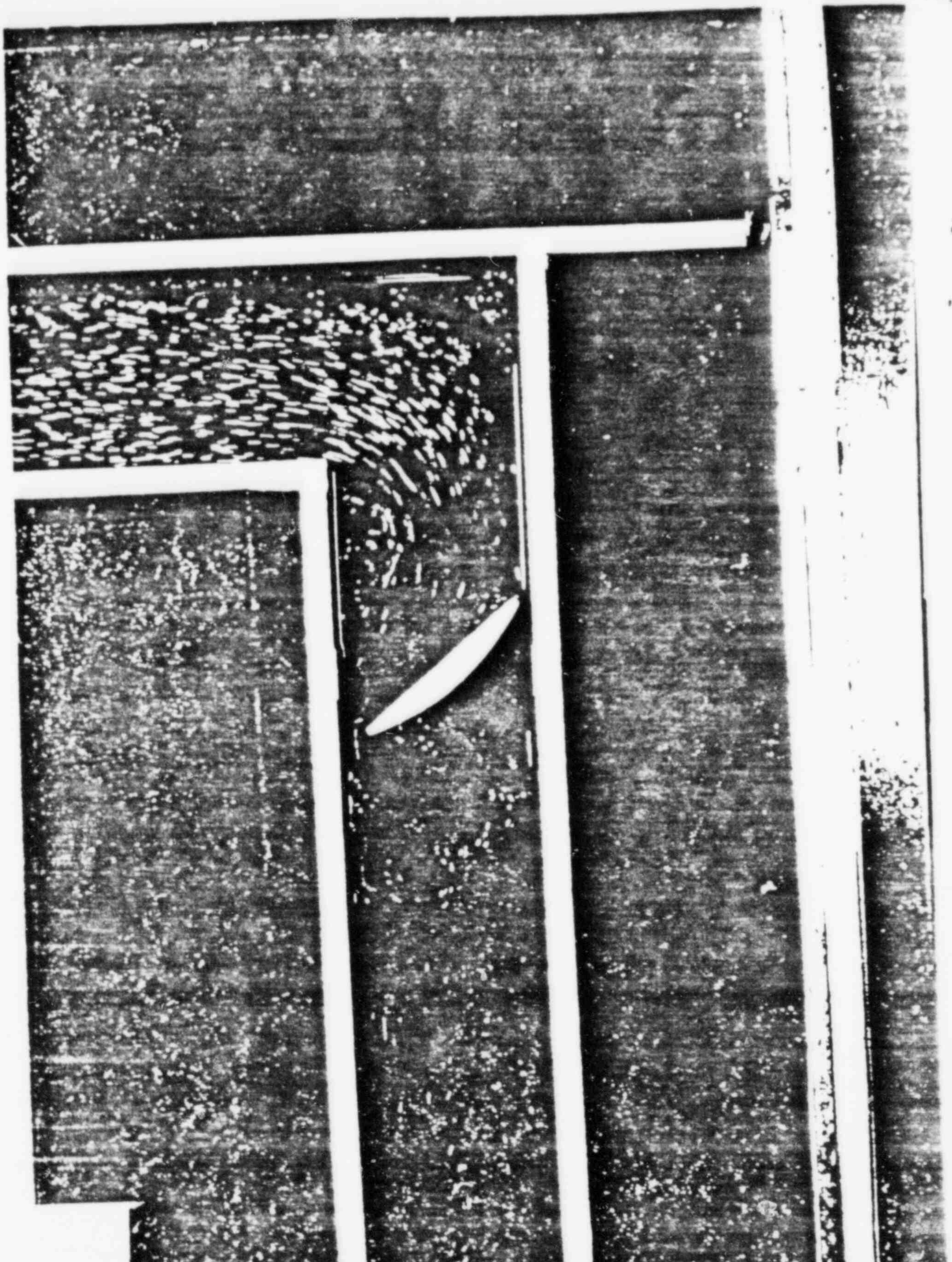


FIGURE 39 - Water Table Study of Choked Flow Pattern With Disc Partially Open (40^0)

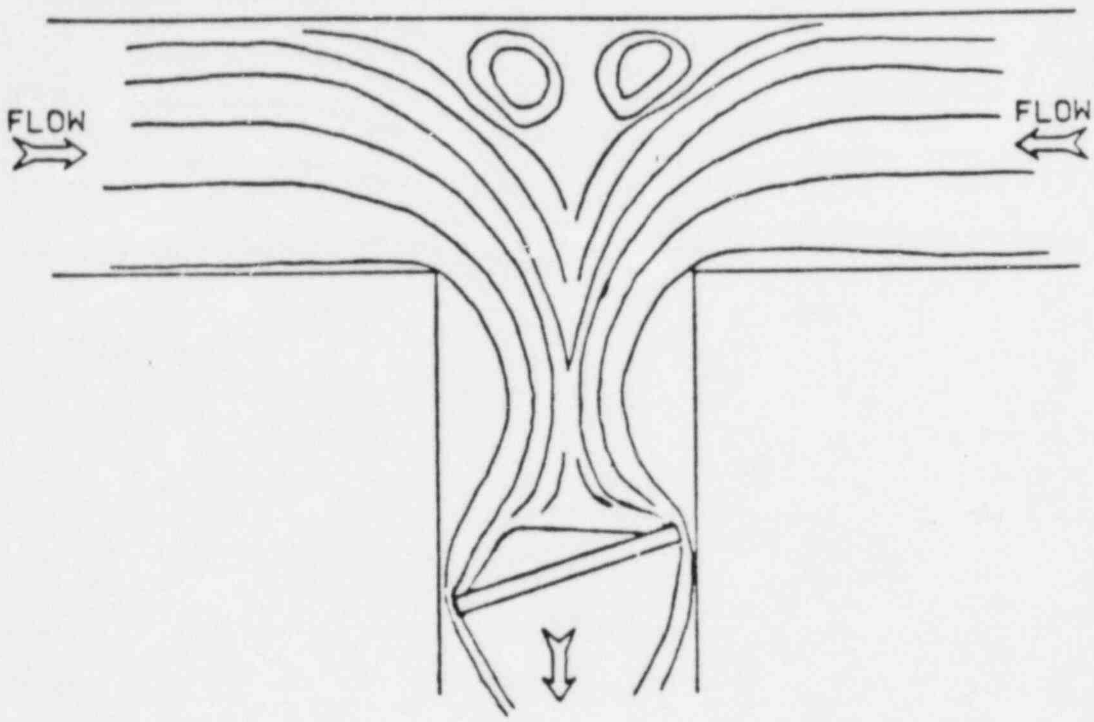


FIGURE 40A
TEE WITH FLOW FROM TWO SIDES

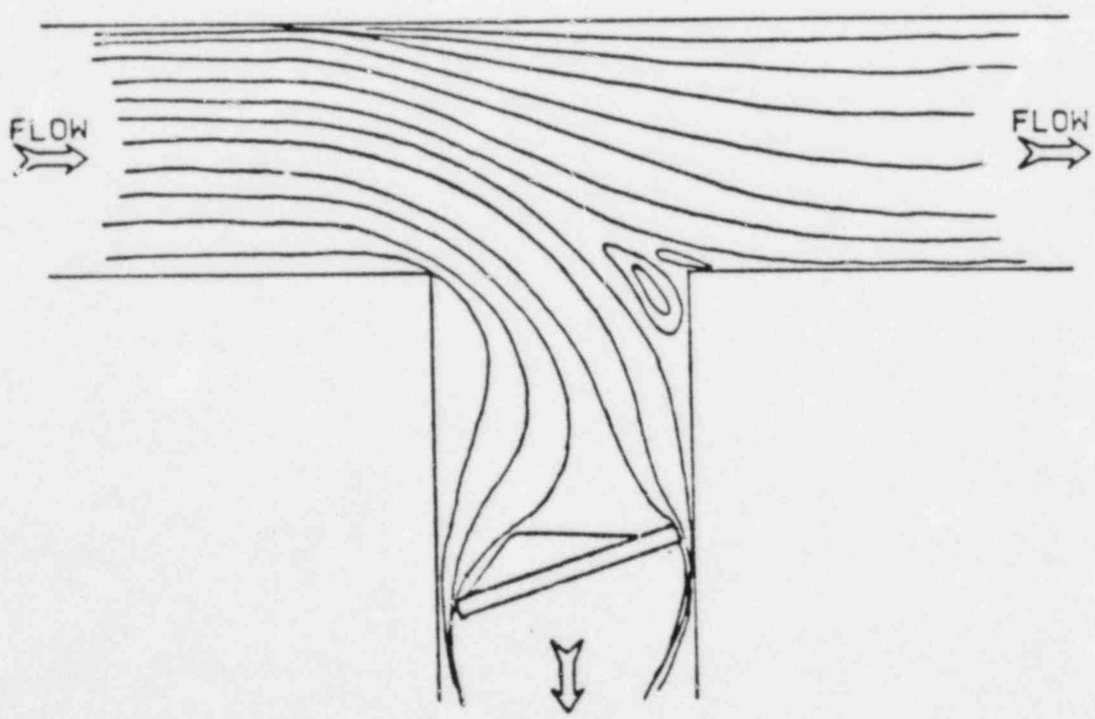
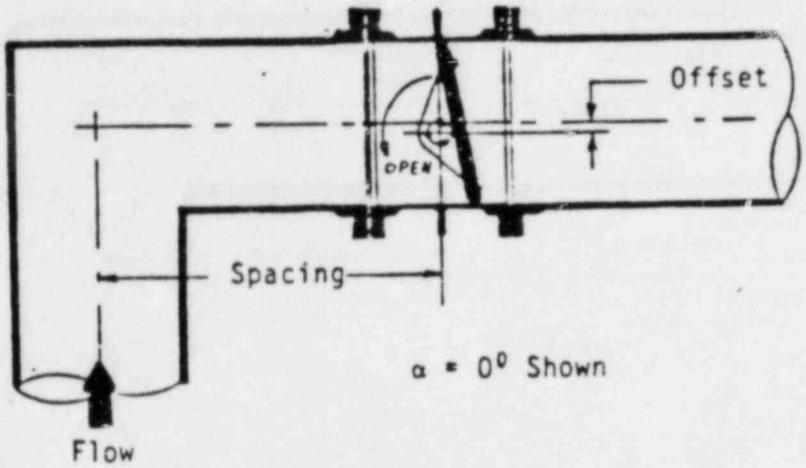


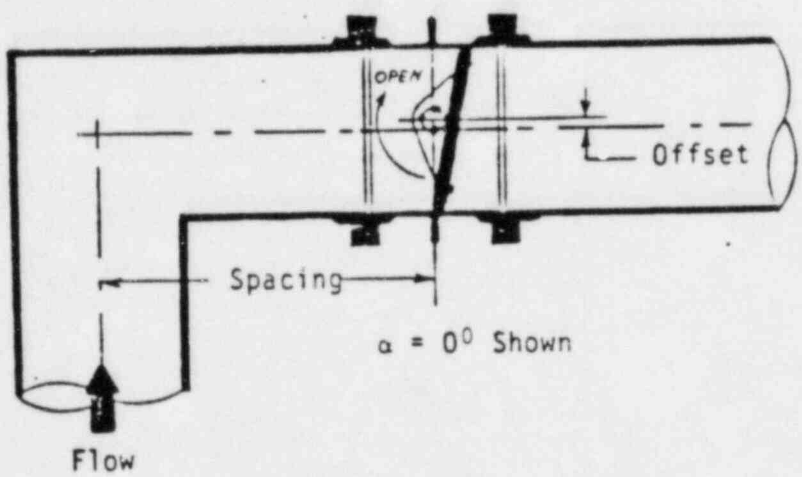
FIGURE 40B
TEE WITH FLOW FROM ONE SIDE

Geometry 1



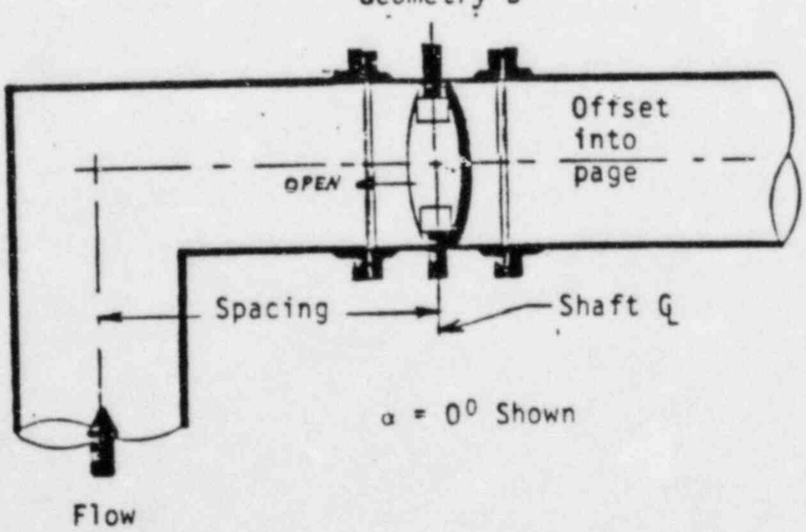
α = 0° Shown

Geometry 2



α = 0° Shown

Geometry 3



α = 0° Shown

FIGURE 41 - Valve Orientations Relative to Upstream Elbow

5.1.4 Downstream Piping Effects

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

5.2 Model Data Verification

A test of a full size 12" valve was run at Vought's High Speed Wind Tunnel in Dallas, Texas (see reference 7.0 B3) to demonstrate operability and substantiate model test data. The tests demonstrated the valve would operate in the required 5 second period. It further showed that torque values were less than predicted from model data. The valve used for the test incorporated a one piece thru shaft design while the model had a two piece shaft. To verify the torque effect due to this change, another test was made (data not put into a formal report form) in which a 2 piece shaft was installed in place of the thru shaft. The test was made with the disc held in a stationary position by a manual worm gear type actuator. The result was that the peak torque was the same for both the one and two piece shaft design. The only difference was that the two piece shaft design showed a peak torque closer (by 5 to 10 degrees) to the full open position. A test was also run with the one piece shaft design with the disc held in a stationary position. This was done to provide direct correlation with the model tests which were done in this manner. It also allowed a comparison to the torques measured during the dynamic test with the shaft connected to the pneumatic actuator. A summary of the operability test is included in Appendix B.

5.3 Application of Model Aerodynamic Test To Full Size Valve Operability

5.3.1 Valve Operating Times Expected In Service

All valves were designed to close within 5 seconds for flow conditions produced by maximum differential pressure (see 3.0, Table 8). These are the maximum conditions expected in the event of a LOCA. The valves were designed to fully open within 5 seconds for conditions of normal flow, though most are capable of opening fully within this time for maximum pressure differential. All air actuated valves will fail closed through use of a return spring in the actuator. They will open within 5 seconds if the air supply to the actuator is adequate. All electric actuated valves will fail in position on loss of power. The electric actuated valves may operate full open to closed in a period greater than 5 seconds due to the design provided by the actuator manufacturer. Any such deviation has been approved by the customer prior to shipment.

In the Vought test, which used a pneumatic/spring return actuated valve, (Reference 7.0) closing times were shown to improve slightly with flow through the valve. Opening times were retarded on the order of 1/2 to 2 seconds depending on flow conditions. These changes are of a conservative nature since it was necessary to restrict both the valve opening and closing air supplies to prevent pressure upstream of the valve from increasing to an unreasonable level during the test. The conduct of the test would suggest that

opening times in actual service for similar valve/actuator assemblies might be retarded about .3 to .5 (since normal flows are much lower than tested flows) and closing times might be improved by the same amount under maximum differential pressure conditions relative to the Clow bench test data. Since the electric actuated valves have a self locking worm gear set, little change in operating time (.2 sec. or less) would be expected under full flow conditions as compared to no flow conditions.

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 19 thru 26 present torque and other data for the subject valves at various operating conditions. The item of concern for valve operability is TQ (for normal operating conditions, open cycle) and TQA (for maximum operating conditions, closing cycle). All torque values shown are positive, tending to close the valves. The meanings of the other listings can be found in 7.0 References C-1.

To obtain torque conditions for the as installed valves a judgement must be made as to what set of test data most nearly represents the actual conditions.

TABLE 19
NORMAL FLOW CALCULATIONS 4" VALVE

CASE: BECHTEL,LIMERICK,82-2053(N)
DATE: 4-7-83
PATM: 14.70(PSIA)
PSU = 15.70(PSIA)
MEDIUM: GAS = A
FLOW = UF
WBO = 150.00(SCFM)
DV = 4.000(IN)

TSU = 609.67(R)
GAMMA = 1.40
OPTION = 1

UNITS SYSTEM: ES
SHAFT: US

MW = 29.0

OUTPUT DATA

SOLUTION: DPBO = .02(PSIG)

PSD/POU = .9980

ALPHA	CF	WR	DPS/PSU	PSU/POU	PSC/POU	POD/POU	TQR1
80.0	.4710	1.0000	.0016	.9995	.7397	.9971	.1222
75.0	.4623	.9814	.0016	.9996	.7378	.9928	.1581
70.0	.4448	.9444	.0016	.9996	.7338	.9894	.1856
65.0	.4201	.8918	.0017	.9996	.7277	.9867	.2052
60.0	.3893	.8264	.0017	.9997	.7193	.9848	.2175
55.0	.3537	.7510	.0018	.9997	.7084	.9837	.2233
50.0	.3147	.6682	.0018	.9998	.6951	.9834	.2234
45.0	.2736	.5808	.0019	.9998	.6795	.9839	.2189
40.0	.2316	.4917	.0019	.9999	.6619	.9851	.2107
35.0	.1901	.4036	.0020	.9999	.6427	.9872	.2002
30.0	.1504	.3193	.0020	1.0000	.6225	.9901	.1887
25.0	.1137	.2414	.0020	1.0000	.6024	.9937	.1776
20.0	.0814	.1729	.0020	1.0000	.5832	.9982	.1685
15.0	.0548	.1164	.0020	1.0000	.5664	1.0000	.1632
10.0	.0352	.0747	.0020	1.0000	.5532	1.0000	.1634
5.0	.0238	.0505	.0020	1.0000	.5456	1.0000	.1710

ALPHA (DEG)	YCV (...)	W (LBM/HR)	TQ (IN-LBF)
80.0	255.17	672.40	.20
75.0	249.14	659.89	.26
70.0	237.40	635.03	.31
65.0	221.27	599.68	.35
60.0	202.01	555.70	.38
55.0	180.73	504.95	.40
50.0	158.51	449.28	.41
45.0	135.93	390.56	.42
40.0	113.78	330.65	.41
35.0	92.56	271.41	.40
30.0	72.70	214.68	.38
25.0	54.71	162.34	.36
20.0	39.05	116.24	.34
15.0	26.23	78.24	.33
10.0	16.81	50.20	.34
5.0	11.38	33.97	.35

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

TABLE 20
NORMAL FLOW CALCULATIONS 6" VALVE

CASE: BECHTEL.LIMERICK,82-2
UNITS SYSTEM: ES
SHAFT: US

DATE: 4-19-83
PATM: 14.70(PSIA)
PSU = 15.70(PSIA) TSU = 609.67(R) MW = 29.0
MEDIUM: GAS = A GAMMA = 1.40
FLOW = UF OPTION = 1
WBO = 150.00(SCFM)
DV = 6.000(IN)

OUTPUT DATA

SOLUTION: DPBO = .00(PSIG)

PSD/POU = .9996

ALPHA	CF	WR	DPS/PSU	PSU/POU	PSC/POU	POD/POU	TQR1
80.0	.4970	1.0000	.0003	.9999	.7446	.9980	.1124
75.0	.4877	.9814	.0003	.9999	.7430	.9938	.1504
70.0	.4693	.9444	.0003	.9999	.7393	.9904	.1786
65.0	.4432	.8918	.0003	.9999	.7334	.9878	.1981
60.0	.4107	.8264	.0003	.9999	.7252	.9860	.2098
55.0	.3732	.7510	.0003	.9999	.7145	.9849	.2148
50.0	.3321	.6682	.0003	1.0000	.7012	.9847	.2142
45.0	.2887	.5808	.0003	1.0000	.6854	.9852	.2092
40.0	.2444	.4917	.0003	1.0000	.6674	.9866	.2008
35.0	.2006	.4036	.0003	1.0000	.6477	.9887	.1905
30.0	.1587	.3193	.0004	1.0000	.6269	.9916	.1794
25.0	.1200	.2414	.0004	1.0000	.6059	.9953	.1688
20.0	.0859	.1729	.0004	1.0000	.5860	.9998	.1603
15.0	.0578	.1164	.0004	1.0000	.5683	1.0000	.1550
10.0	.0371	.0747	.0004	1.0000	.5547	1.0000	.1546
5.0	.0251	.0505	.0004	1.0000	.5464	1.0000	.1606

ALPHA (DEG)	YCV (...)	W (LBM/HR)	TQ (IN-LBF)
80.0	616.22	672.40	.10
75.0	601.09	659.89	.14
70.0	572.54	635.03	.17
65.0	532.40	599.68	.20
60.0	484.10	555.70	.21
55.0	432.60	504.95	.23
50.0	379.16	449.28	.23
45.0	324.59	390.56	.24
40.0	270.93	330.65	.23
35.0	220.24	271.41	.22
30.0	172.79	214.68	.22
25.0	129.92	162.34	.20
20.0	92.63	116.24	.20
15.0	62.25	78.24	.19
10.0	39.92	50.20	.19
5.0	27.03	33.97	.20

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

N/

TABLE 21
NORMAL FLOW CALCULATIONS 18" VALVE

CASE: BECHTEL,LIMERICK,82-2053(N)
DATE: 4-7-83
PATM: 14.70(PSIA)
PSU = 15.70(PSIA) TSU = 609.67(R)
MEDIUM: GAS = A GAMMA = 1.40
FLOW = UF OPTION = 1
WBO = 4400.00(SCFM)
DV = 18.000(IN)

UNITS SYSTEM: ES
SHAFT: US

MW = 29.0

OUTPUT DATA

SOLUTION: DPBO = .03(PSIG)

PSD/POU = .9971

ALPHA	CF	WR	DPS/PSU	PSU/POU	PSC/POU	POD/POU	TQR1
80.0	.5747	1.0000	.0019	.9991	.7558	.9968	.0532
75.0	.5640	.9814	.0020	.9991	.7547	.9926	.1034
70.0	.5427	.9444	.0020	.9992	.7520	.9891	.1360
65.0	.5125	.8918	.0021	.9992	.7474	.9864	.1546
60.0	.4749	.8264	.0022	.9993	.7405	.9844	.1627
55.0	.4316	.7510	.0023	.9995	.7306	.9832	.1629
50.0	.3840	.6682	.0025	.9996	.7177	.9829	.1580
45.0	.3338	.5808	.0026	.9997	.7018	.9833	.1499
40.0	.2826	.4917	.0027	.9998	.6831	.9845	.1405
35.0	.2320	.4036	.0027	.9998	.6620	.9865	.1310
30.0	.1835	.3193	.00[illegible]	.9999	.6394	.9893	.1224
25.0	.1387	.2414	.0028	.9999	.6163	.9929	.1152
20.0	.0993	.1729	.0029	1.0000	.5940	.9973	.1095
15.0	.0669	.1164	.0029	1.0000	.5741	1.0000	.1051
10.0	.0429	.0747	.0029	1.0000	.5586	1.0000	.1014
5.0	.0290	.0505	.0029	1.0000	.5491	1.0000	.0972

ALPHA (DEG)	YCV (...)	W (LBM/HR)	TQ (IN-LBF)
80.0	6789.81	19723.82	9.41
75.0	6606.13	19356.80	18.62
70.0	6248.45	18627.55	25.35
65.0	5771.90	17590.59	30.12
60.0	5220.21	16300.50	33.26
55.0	4627.45	14811.78	35.00
50.0	4022.69	13178.93	35.55
45.0	3425.04	11456.55	35.16
40.0	2849.53	9699.13	34.12
35.0	2306.50	7961.22	32.71
30.0	1805.31	6297.34	31.21
25.0	1355.14	4762.02	29.81
20.0	965.68	3409.78	28.61
15.0	648.21	2295.12	27.62
10.0	415.34	1472.56	26.70
5.0	280.97	996.57	25.62

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

N

TABLE 22
NORMAL FLOW CALCULATIONS 24" VALVE

CASE: BECHTEL,LIMERICK,82-2053(N)
DATE: 4-7-83 UNITS SYSTEM: ES
PATM: 14.70(PSIA) SHAFT: US
PSU = 15.70(PSIA) TSU = 609.67(R)
MEDIUM: GAS = A GAMMA = 1.40 MW = 29.0
FLOW = UF OPTION = 1
WBO = 6600.00(SCFM)
DV = 24.000(IN)

OUTPUT DATA

SOLUTION: DPBO = .02(PSIG)

PSD/POU = .9981

ALPHA	CF	WR	DPS/PSU	PSU/POU	PSC/POU	POD/POU	TQR1
80.0	.5970	1.0000	.0012	.9993	.7578	.9973	.0237
75.0	.5859	.9814	.0012	.9993	.7569	.9931	.0800
70.0	.5638	.9444	.0013	.9994	.7546	.9896	.1148
65.0	.5324	.8918	.0014	.9995	.7506	.9869	.1331
60.0	.4934	.8264	.0014	.9995	.7434	.9850	.1392
55.0	.4483	.7510	.0015	.9996	.7346	.9839	.1371
50.0	.3989	.6682	.0016	.9997	.7220	.9836	.1300
45.0	.3467	.5808	.0017	.9998	.7062	.9840	.1205
40.0	.2936	.4917	.0017	.9998	.6873	.9853	.1105
35.0	.2410	.4036	.0018	.9999	.6660	.9874	.1014
30.0	.1906	.3193	.0018	.9999	.6429	.9902	.0941
25.0	.1441	.2414	.0019	1.0000	.6192	.9939	.0885
20.0	.1032	.1729	.0019	1.0000	.5963	.9983	.0843
15.0	.0695	.1164	.0019	1.0000	.5757	1.0000	.0804
10.0	.0446	.0747	.0019	1.0000	.5597	1.0000	.0749
5.0	.0302	.0505	.0019	1.0000	.5499	1.0000	.0657

ALPHA (DEG)	YCV (...)	W (LBM/HR)	TQ (IN-LBF)
80.0	12800.11	29585.73	6.29
75.0	12435.19	29035.20	21.70
70.0	11738.80	27941.31	32.34
65.0	10818.82	26385.90	39.34
60.0	9758.69	24450.74	43.44
55.0	8629.82	22217.65	45.18
50.0	7483.72	19768.39	45.09
45.0	6359.47	17184.81	43.72
40.0	5283.71	14548.70	41.63
35.0	4271.33	11941.82	39.40
30.0	3340.71	9446.01	37.37
25.0	2505.86	7143.03	35.74
20.0	1785.03	5114.67	34.40
15.0	1197.93	3442.68	32.97
10.0	767.38	2208.84	30.84
5.0	519.04	1494.85	27.07

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

TABLE 23

EMERGENCY FLOW CALCULATIONS 4" VALVE

CASE: BECHTEL,LIMERICK,82-2053(NO
DATE: 4-7-83
PATM: 14.70(PSIA)
PSU = 69.70(PSIA) TSU = 799.67(R)
MEDIUM: GAS = A GAMMA = 1.40
FLOW = CF OPTION = 2
DV = 4.000(IN)

UNITS SYSTEM: ES
SHAFT: US
MW = 29.0

OUTPUT DATA

CHOKING PRESSURE RATIOS: PSC/POU = .740 DPS/PSU = .217
SOLUTION: W80 = 8.15(LBM/S)

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

PSD/POU = .7397

ALPHA	CF	WR	DPS/PSU	PSU/POU	PSC/POU	POD/POU	TQR1
80.0	.4710	1.0000	.2171	.9447	.7397	.8491	.1046
75.0	.4623	.9814	.2188	.9469	.7378	.8381	.1404
70.0	.4448	.9444	.2223	.9510	.7338	.8265	.1676
65.0	.4201	.8918	.2268	.9566	.7277	.8148	.1868
60.0	.3893	.8264	.2319	.9630	.7193	.8032	.1987
55.0	.3537	.7510	.2372	.9697	.7084	.7919	.2041
50.0	.3 47	.6682	.2423	.9762	.6951	.7814	.2038
45.0	.2736	.5808	.2469	.9821	.6795	.7718	.1989
40.0	.2316	.4917	.2508	.9872	.6619	.7634	.1904
35.0	.1901	.4036	.2540	.9914	.6427	.7562	.1796
30.0	.1504	.3193	.2564	.9947	.6225	.7503	.1679
25.0	.1137	.2414	.2581	.9970	.6024	.7458	.1567
20.0	.0814	.1729	.2592	.9984	.5832	.7426	.1475
15.0	.0548	.1164	.2598	.9993	.5664	.7406	.1421
10.0	.0352	.0747	.2601	.9997	.5532	.7397	.1423
5.0	.0238	.0505	.2602	.9999	.5456	.7398	.1499

ALPHA (DEG)	YCV (...)	W (LBM/HR)	TQ (IN-LBF)	TQA (IN-LBF)
80.0	247.86	29341.05	101.25	368.11
75.0	241.70	28795.05	137.32	495.17
70.0	229.79	27710.23	167.25	593.83
65.0	213.57	26167.67	191.36	665.86
60.0	194.40	24248.53	209.56	713.03
55.0	173.47	22033.90	221.64	737.34
50.0	151.70	19604.90	227.58	741.23
45.0	129.86	17042.70	227.64	727.70
40.0	108.51	14428.37	222.58	700.39
35.0	88.13	11843.06	213.57	663.63
30.0	69.16	9367.89	202.19	622.36
25.0	52.00	7083.95	190.37	582.11
20.0	37.10	5072.37	180.28	548.90
15.0	24.92	3414.21	174.24	529.22
10.0	15.97	2190.57	174.69	529.96
5.0	10.81	1482.49	184.15	558.41

TABLE 24
EMERGENCY FLOW CALCULATIONS 6" VALVE

CASE: BECHTEL,LIMERICK,82-2053(N)
DATE: 4-7-83 UNITS SYSTEM: ES
PATM: 14.70(PSIA) SHAFT: US
PSU = 69.70(PSIA) TSU = 799.67(R)
MEDIUM: GAS = A GAMMA = 1.40 MW = 29.0
FLOW = CF OPTION = 2
DV = 6.000(IN)

OUTPUT DATA

CHOKING PRESSURE RATIOS: PSC/POU = .745 DPS/PSU = .206
SOLUTION: W80 = 19.47(LBM/S)

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

PSD/POU = .7446

ALPHA	CF	WR	DPS/PSU	PSU/POU	PSC/POU	POD/POU	TQR1
80.0	.4970	1.0000	.2062	.9380	.7446	.8564	.0956
75.0	.4877	.9814	.2083	.9404	.7430	.8455	.1334
70.0	.4693	.9444	.2122	.9451	.7393	.8339	.1613
65.0	.4432	.8918	.2174	.9514	.7334	.8219	.1803
60.0	.4107	.8264	.2233	.9586	.7252	.8099	.1916
55.0	.3732	.7510	.2293	.9661	.7145	.7983	.1961
50.0	.3321	.6682	.2351	.9734	.7012	.7874	.1950
45.0	.2887	.5808	.2403	.9800	.6854	.7774	.1895
40.0	.2444	.4917	.2447	.9858	.6674	.7687	.1808
35.0	.2006	.4036	.2483	.9905	.6477	.7613	.1702
30.0	.1587	.3193	.2510	.9941	.6269	.7553	.1589
25.0	.1200	.2414	.2529	.9966	.6059	.7508	.1482
20.0	.0859	.1729	.2541	.9983	.5860	.7476	.1395
15.0	.0578	.1164	.2549	.9992	.5683	.7456	.1342
10.0	.0371	.0747	.2552	.9997	.5547	.7448	.1338
5.0	.0251	.0505	.2553	.9999	.5464	.7450	.1397

ALPHA (DEG)	YCV (...)	W (LBM/HR)	TQ (IN-LBF)	TQA (IN-LBF)
80.0	608.02	70082.34	296.73	1135.55
75.0	592.18	68778.28	419.32	1588.72
70.0	561.76	66187.12	519.23	1930.79
65.0	520.63	62502.62	598.72	2173.08
60.0	472.48	57918.69	658.14	2325.86
55.0	420.36	52628.94	697.25	2399.28
50.0	366.65	46827.16	716.21	2404.18
45.0	313.13	40707.25	716.31	2352.59
40.0	261.16	34462.80	700.17	2257.96
35.0	211.81	28287.68	671.75	2135.15
30.0	166.03	22375.62	636.20	2000.28
25.0	124.76	16920.34	599.48	1870.53
20.0	88.96	12115.57	568.05	1763.84
15.0	59.74	8154.99	548.61	1698.72
10.0	38.29	5232.27	547.87	1694.14
5.0	25.90	3540.99	572.52	1769.48

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

TABLE 25
EMERGENCY FLOW CALCULATIONS 18" VALVE

CASE: BECHTEL,LIMERICK,82-2053(N)
DATE: 4-7-83
PATM: 14.70(PSIA)
PSU = 69.70(PSIA) TSU = 799.67(R)
MEDIUM: GAS = A GAMMA = 1.40
FLOW = CF OPTION = 2
DV = 18.000(IN)

UNITS SYSTEM: ES
SHAFT: US

MW = 29.0

OUTPUT DATA

CHOKING PRESSURE RATIOS: PSC/POU = .756 DPS/PSU = .174
SOLUTION: WBO = 207.00(LBM/S)

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

PSD/POU = .7558

ALPHA	CF	WR	DPS/PSU	PSU/POU	PSC/POU	POD/POU	TQR1
80.0	.5747	1.0000	.1738	.9148	.7558	.8785	.0390
75.0	.5640	.9814	.1769	.9183	.7547	.8676	.0890
70.0	.5427	.9444	.1828	.9249	.7520	.8553	.1211
65.0	.5125	.8918	.1905	.9337	.7474	.8422	.1391
60.0	.4749	.8264	.1991	.9437	.7405	.8288	.1464
55.0	.4316	.7510	.2078	.9541	.7306	.8156	.1460
50.0	.3840	.6682	.2160	.9640	.7177	.8032	.1404
45.0	.3338	.5808	.2233	.9731	.7018	.7920	.1317
40.0	.2826	.4917	.2295	.9809	.6831	.7821	.1218
35.0	.2320	.4036	.2344	.9872	.6620	.7739	.1119
30.0	.1835	.3193	.2381	.9920	.6394	.7673	.1030
25.0	.1387	.2414	.2407	.9955	.6163	.7624	.0956
20.0	.0993	.1729	.2424	.9977	.5940	.7591	.0898
15.0	.0669	.1164	.2434	.9989	.5741	.7572	.0853
10.0	.0429	.0747	.2439	.9996	.5586	.7566	.0815
5.0	.0290	.0505	.2440	.9998	.5491	.7570	.0773

ALPHA (DEG)	YCV (...)	W (LBM/HR)	TQ (IN-LBF)	TQA (IN-LBF)
80.0	7067.39	745186.25	2754.78	12508.30
75.0	6848.29	731319.75	6423.24	28649.73
70.0	6436.71	703767.75	9097.80	39270.39
65.0	5897.96	664591.00	10996.10	45543.58
60.0	5289.47	615849.75	12225.10	48449.50
55.0	4653.79	559604.00	12860.17	48835.41
50.0	4019.45	497913.62	12987.55	47448.19
45.0	3404.63	432840.25	12717.72	44944.75
40.0	2820.83	366443.25	12179.94	41888.17
35.0	2276.20	300783.00	11505.42	38735.11
30.0	1777.64	237919.94	10809.42	35822.37
25.0	1332.31	179914.06	10175.66	33355.00
20.0	948.55	128824.97	9645.87	31398.53
15.0	636.39	86712.09	9214.48	29875.41
10.0	407.67	55634.77	8826.97	28564.29
5.0	275.73	37651.42	8381.03	27101.45

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

TABLE 26

EMERGENCY FLOW CALCULATIONS 24" VALVE

CASE: BECHTEL,LIMERICK,82-2053(N)
DATE: 4-7-83
PATM: 14.70(PSIA)
PSU = 69.70(PSIA)
MEDIUM: GAS = A
FLOW = CF
DV = 24.000(IN)

TSU = 799.67(R)
GAMMA = 1.40
OPTION = 2

UNITS SYSTEM: ES
SHAFT: US

MW = 29.0

OUTPUT DATA

CHOKING PRESSURE RATIOS: PSC/POU = .758 DPS/PSU = .165
SOLUTION: W80 = 384.99(LBM/S)

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

PSD/POU = .7578

ALPHA	CF	WR	DPS/PSU	PSU/POU	PSC/POU	POD/POU	TQR1
80.0	.5970	1.0000	.1647	.9072	.7578	.8846	.0101
75.0	.5859	.9814	.1683	.9111	.7569	.8737	.0662
70.0	.5638	.9444	.1748	.9183	.7546	.8612	.1005
65.0	.5324	.8918	.1834	.9280	.7506	.8476	.1180
60.0	.4934	.8264	.1930	.9389	.7434	.8336	.1234
55.0	.4483	.7510	.2025	.9502	.7346	.8199	.1205
50.0	.3989	.6682	.2115	.9611	.7220	.8069	.1126
45.0	.3467	.5808	.2195	.9709	.7062	.7952	.1024
40.0	.2936	.4917	.2262	.9793	.6873	.7850	.0919
35.0	.2410	.4036	.2316	.9862	.6660	.7764	.0824
30.0	.1906	.3193	.2357	.9914	.6429	.7696	.0747
25.0	.1441	.2414	.2385	.9951	.6192	.7646	.0689
20.0	.1032	.1729	.2403	.9975	.5963	.7612	.0646
15.0	.0695	.1164	.2414	.9989	.5757	.7592	.0605
10.0	.0446	.0747	.2419	.9995	.5597	.7586	.0551
5.0	.0302	.0505	.2421	.9998	.5499	.7591	.0458

ALPHA (DEG)	YCV (...)	W (LBM/HR)	TQ (IN-LBF)	TQA (IN-LBF)
80.0	13516.89	1385976.00	1607.16	7698.29
75.0	13071.36	1360185.50	10779.35	50557.17
70.0	12241.69	1308941.50	17128.80	77308.28
65.0	11169.06	1236076.00	21328.98	91760.06
60.0	9973.56	1145422.00	23735.84	97074.44
55.0	8740.61	1040810.50	24626.73	95952.09
50.0	7523.94	926072.00	24318.80	90721.25
45.0	6355.61	805042.00	23186.74	83352.94
40.0	5254.50	681549.50	21626.09	75433.00
35.0	4233.11	559427.75	19991.84	68115.81
30.0	3302.06	442508.62	18538.34	62079.27
25.0	2472.85	334623.00	17374.76	57489.56
20.0	1759.68	239602.19	16440.28	53984.20
15.0	1180.26	161276.25	15497.00	50667.03
10.0	755.96	103475.44	14133.72	46113.09
5.0	511.27	70028.09	11771.11	38374.39

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

For Figures 26, 28, 29, and 32 (Section 3.2), the configurations for MO-57-163(4"), MO-57-109(6"), AO-57-121(6"), and AO-57-114(24") indicate upstream piping is at a sufficient distance so that all the valves will respond as if under fully developed straight run pipe flow. Thus, for all these valves the torque modification factor comparing straight line flow to actual flow is 1.0 as indicated in Tables 29, 30, and 25. For Figure 25 (MO-57-161) an upstream radius elbow at 1.9 diameters is indicated. This is reasonably represented in the calculations for the case of an upstream mitered elbow at 2 diameters. A worst case orientation (flow tends to minimize torque aiding closure) was used in performing calculations even though the valve may be installed in a different orientation. For Figure 27, 30, and 31 an upstream tee is the major element of concern (reducer ahead of MO-57-162 tends to straighten flow into valve). These valves (MO-57-162, 164, AO-57-131) are best represented by a mitered elbow upstream at 4 diameters. In reality, use of a mitered elbow for calculation here is more severe than would occur in actual service. Also a worst case orientation was assumed. For Figure 33, valve AO-57-104 and upstream mitered elbow at 4 diameters with worst case valve orientation is considered. For Figures 34 and 35, valves AO-57-123 and 124, an upstream mitered elbow at 2 diameters is considered. For Figures 32, 33, 35, and 34 valves MO-57-112, 115, 135, and 147 a worst case failure is assumed. Valve orientations relative to one another are assumed worst case and it is assumed the upstream valve has failed and stopped at the 60^{o} position

(a worst case for the downstream valve). This will result in an aerodynamic torque on the downstream valve which tends to resist closure. Such a torque must be overcome by the actuator for the valve to close. For all the subject cases, the resultant torques are summarized in Tables 32 and 35.

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

Table 27

Valve No. MO-57-161 (4")

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

Model Test Valve Angle	Actual Valve Angle	Torque for Straight Flow Normal	Maximum	Torque Modification Factor	Torque for Installed Condition Normal	Maximum
80	90	1	370	.45	1	167
70	80	1	595	.84	1	500
60	70	1	715	.94	1	672
50	60	1	740	1.0	1	740
40	50	1	700	1.0	1	700
30	40	1	625	1.0	1	625
20	30	1	550	1.0	1	550
10	20	1	530	1.0	1	530

Table 28

Valve No. MO-57-163 (4")

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

Model Test Valve Angle	Actual Valve Angle	Torque for Straight Flow Normal	Maximum	Torque Modification Factor	Torque for Installed Condition Normal	Maximum
80	90	1	370	1.0	1	320
70	80	1	595	1.0	1	595
60	70	1	715	1.0	1	715
50	60	1	740	1.0	1	740
40	50	1	700	1.0	1	700
30	40	1	625	1.0	1	625
20	30	1	550	1.0	1	550
10	20	1	530	1.0	1	530

Table 29

Valve No. MO-57-109 & AO-57-121 (6")

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

Valves under straight line flow conditions

Model Test Valve Angle	Actual Valve Angle	Torque for Straight Flow Normal	Maximum	Torque Modification Factor	Torque for Installed Condition Normal	Maximum
80	90	1	1135	1.0	1	1135
70	80	1	1930	1.0	1	1930
60	70	1	2325	1.0	1	2325
50	60	1	2400	1.0	1	2400
40	50	1	2260	1.0	1	2260
30	40	1	2000	1.0	1	2000
20	30	1	1765	1.0	1	1765
10	20	1	1695	1.0	1	1695

Table 30

Valve No. MO-57-162, MO-57-164, AO-57-131 (6")

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

Mitered elbow 4 diameter upstream Geometry 1

Model Test Valve Angle	Actual Valve Angle	Torque for Straight Flow Normal	Maximum	Torque Modification Factor	Torque for Installed Condition Normal	Maximum
80	90	1	1135	.75	1	850
70	80	1	1930	.95	1	1835
60	70	1	2325	1.0	1	2325
50	60	1	2400	1.0	1	2400
40	50	1	2260	1.0	1	2260
30	40	1	2000	1.0	1	2000
20	30	1	1765	1.0	1	1765
10	20	1	1695	1.0	1	1695

Valve No. AO-57-104 (18")

Model Data For Torque Modification: Mitered elbow 4 diameter upstream Geometry 2
All torques in In-lbs.

Model Test Valve Angle	Actual Valve Angle	Torque for Straight Flow Normal	Maximum	Torque Modification Factor	Torque for Installed Condition Normal	Maximum
80	90	9	12,510	-.36	-3	-4,503
70	80	25	39,270	.84	+21	32,990
60	70	33	48,450	1.0	33	48,450
50	60	35	47,450	1.0	35	47,450
40	50	34	41,890	1.0	34	41,890
30	40	31	35,820	1.0	31	35,820
20	30	28	31,400	1.0	28	31,400
10	20	27	28,565	1.0	27	28,565

Table 32

Valve No. MO-57-112 (18")

Model Data For Torque Modification: 2 valves in series upstream valve assumed failed at 60° open position
All torques in In-lbs.

Model Test Valve Angle	Actual Valve Angle	Torque for Straight Flow Normal	Maximum	Torque Modification Factor	Torque for Installed Condition Normal	Maximum
80	90	9	12,510	-.17	-2	-2,127
70	80	25	39,270	+.67	+17	+26,310
60	70	33	48,450	1.0	33	48,450
50	60	35	47,450	1.0	35	47,450
40	50	34	41,890	1.0	34	41,890
30	40	31	35,820	1.0	31	35,820
20	30	28	31,400	1.0	28	31,400
10	20	27	28,565	1.0	27	28,565

Table 33

Valve No. AO-57-114 (24")

Model Data For Torque Modification: Straight pipe run
All torques in In-lbs. (Upstream elbow at more than 8 dia.)

Model Test Valve Angle	Actual Valve Angle	Torque for Straight Flow Normal	Maximum	Torque Modification Factor	Torque for Installed Condition Normal	Maximum
80	90	7	7,698	1.0	7	7,698
70	80	32	77,310	1.0	32	77,310
60	70	43	97,075	1.0	43	97,075
50	60	45	90,720	1.0	45	90,720
40	50	42	75,430	1.0	42	75,430
30	40	37	62,080	1.0	37	62,080
20	30	34	53,980	1.0	34	53,980
10	20	30	46,100	1.0	30	46,100

Table 34

Valve No. AO-57-123 and 124 (24")

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

Model Test Valve Angle	Actual Valve Angle	Torque for Straight Flow Normal	Maximum	Torque Modification Factor	Torque for Installed Condition Normal	Maximum
80	90	7	7,698	-6.0	-42	-46,188
70	80	32	77,310	1.52	+48	115,970
60	70	43	97,075	1.18	51	114,550
50	60	45	90,720	1.20	54	108,860
40	50	42	75,430	1.16	49	87,500
30	40	37	62,080	1.0	37	62,080
20	30	34	53,980	1.0	34	53,980
10	20	30	46,100	1.0	30	- 46,100

Table 35

Valve No. MO-57-115, 135, and 147 (24")

Model Data For Torque Modification: 2 valves in series upstream valve assumed failed at 60° open position
All torques in In-lbs.

Model Test Valve Angle	Actual Valve Angle	Torque for Straight Flow Normal	Torque for Straight Flow Maximum	Torque Modification Factor	Torque for Installed Condition Normal	Torque for Installed Condition Maximum
80	90	7	7,698	-1.5	-11	-11,550
70	80	32	77,310	+ .48	15	+37,110
60	70	43	97,075	1.0	43	97,075
50	60	45	90,720	1.0	45	90,720
40	50	42	75,430	1.0	42	75,430
30	40	37	62,080	1.0	37	62,080
20	30	34	53,980	1.0	34	53,980
10	20	30	46,100	1.0	·30	46,100

5.3.3 Conclusions Concerning Valve Operability

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

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

The following additional criteria apply for electric actuated valves only:

1. The worst case torque resisting valve closure must be less by a sufficient margin than the closed torque switch trip point.

For LOCA flow conditions it can be seen in Tables for AO-57-121, 131, 114 that all aerodynamic torques tend to aid valve closure. For valves AO-57-104, 123, 124 aerodynamic torques for the first 5 degrees from full open resist closure. Pertinent torques for air operated valves are listed in Table 36.

TABLE 36

Pneumatic Actuated Valve Torques
Torques (in-lb)

Valve Size	Valve No.	Valve Design Torque	1	2	3	4	5
6"	AO-57-121	7,800	none	none req'd	2,400	27,585	26,668
6"	AO-57-131	7,800	none	none req'd	2,400	27,585	26,668
24"	AO-57-114	135,000	none	none req'd	97,075	362,496	211,000
18"	AO-57-104	63,300	4,503	76,096	48,450	207,563	174,000
24"	AO-57-123	135,000	46,188	124,695	115,970	362,496	211,000
24"	AO-57-124	135,000	46,188	124,695	115,970	362,496	211,000

1 Worst case closure resisting aerodynamic torque

2 Actuator torque use to overcome aerodynamic torque

3 Maximum aerodynamic torque

4 Torque to yield actuator key

5 Actuator safe structural torque

For valves MO-57-109, 161, 162, 163, 164 for LOCA flow conditions, it can be seen in Tables 29, 27, 28, 30 that all aerodynamic torques are in the closed direction. Due to the use of an electric actuator, this torque will not help the valve close. For valves MO-57-112, 115, 135, 147, aerodynamic torques for the first 5^{0} from full open resist closure. For these units the closed torque switch trip point must be greater than the maximum aerodynamic torque resisting closure.

TABLE 37

Electric Actuated Valve Torques
Torque (in-lb)

Valve Size	Valve No.	Valve Design Torque	1	2	3	4	5
4"	MO-57-161	2,112	none	none req'd	740	7,594	15,600
4"	MO-57-163	2,112	none	none req'd	740	7,594	15,600
6"	MO-57-109	7,800	none	none req'd	2,400	27,585	26,400
6"	MO-57-162	7,800	none	none req'd	2,400	27,585	26,400
6"	MO-57-164	7,800	none	none req'd	2,400	27,585	26,400
18"	MO-57-112	63,300	2,127	34,000	48,450	207,563	235,000
24"	MO-57-115	135,000	11,550		97,075	362,496	235,000
24"	MO-57-135	135,000	11,550		97,075	362,496	235,000
24"	MO-57-147	135,000	11,550		97,075	362,496	235,000

1 Worst case closure resisting aerodynamic torque
2 Torque switch trip torque
(Actuator torque used to overcome aerodynamic torque)
3 Maximum aerodynamic torque
4 Torque to yield actuator key
5 Actuator safe structural torque

From the preceding data it can be seen that the minimum actuator torque margin over that required to overcome worst case aerodynamic torque is better than 2.7. The safety factor is obtained even after full containment pressure has been developed!

From the presented data and supplemental test reports, it has been shown that the valves will operate as designed under the prescribed conditions. This has been shown using the conservative assumption of no credit taken for pressure ramp in containment and no credit taken for back pressure due to downstream piping. Further, no credit has been taken for activation of the first valve under back pressure conditions produced on the second valve due to closure and pressure drop across the first valve.

6.0 VALVE SEALING CHARACTERISTICS

6.1 Normal Sealing

The following chart shows the sealing ability of the valves as they were shop tested for record. The tests were performed with pressure on the indicated side of the disc and the opposite side open to atmosphere. The normal recommended flow direction for these valves is with pressure on the shaft side, so when pressure is applied to the clamp ring side, it is considered to be the reverse flow direction. The test performed was an air under water test in which the smallest detectable leakage was a bubble (.15cc).

TABLE 38

VALVE SEALING CHARACTERISTICS

VALVE MARK NO.	VALVE SIZE (IN.)	TEST PRESSURE PSIG	LEAKAGE (Bubbles/min)* Pressurized Side Shaft Side	Clamp Ring Side
MO-57-161	4	5, 25, & 55	0	0
MO-57-163	4	5, 25, & 55	0	0
MO-57-109	6	5, 25, & 55		
MO-57-162	6	5, 25, & 55	0	0
MO-57-164	6	5, 25, & 55	0	0
AO-57-121	6	5, 25, & 55		
AO-57-131	6	5, 25, & 55	0	0
MO-57-112	18	5, 25, & 55		
AO-57-104	18	5, 25, & 55	0	0
MO-57-115	24	5, 25, & 55		
MO-57-135	24	5, 25, & 55		
MO-57-147	24	5, 25, & 55		
AO-57-114	24	5, 25, & 55		
AO-57-123	24	5, 25, & 55	0	0
AO-57-124	24	5, 25, & 55	0	0

*Worst case pressure

6.2 Long Term Sealing

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

This has been verified by experience and is documented in the Shell International Cycling Test (reference 7.0 D3). This test was performed by Gebruder Adams of Bochum, West Germany. Clow's Engineered Products Division produces the Tricentric design under license of Gebruder Adams. The test showed sealing improved continuously up to 41,000 cycles, the limit of the test.

6.3 Debris Effects On Sealing

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

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

6.4 Sealing Under Temperature Variations

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

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

7.0 REFERENCES

A. Seismic Analysis Reports

prepared by: Patel Engineers
Huntsville, Alabama

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

1. Report PEI-TR-83-16 Rev. A for Clow 4" Wafer Stop Valve (April 83)
   Mark Nos. 4"-HBB-BF-MO-57-161 and 163
   Clow Job No. 82-2053-01(N)-01 & 02

2. Report PEI-TR-83-15 Rev. A for Clow 6" Wafer Stop Valve (April 83)
   Mark Nos. 6"-HBB-BF-MO-57-109, 162, and 164
   6"-HBB-BF-AO-57-121 and 131
   Clow Job No. 82-2053-02(N)-01, 02, & 03
   82-2053-03(N)-01 & 02

3. Report PEI-TR-83-14 Rev. A for Clow 18" Wafer Stop Valve (April 83)
   Mark Nos. 18"-HBB-BF-MO-57-104
   18"-HBB-BF-AO-57-112
   Clow Job No. 82-2053-04(N)-01
   82-2053-05(N)-01

4. Report PEI-TR-83-13 Rev. A for Clow 24" Wafer Stop Valve (April 83)
   Mark Nos. 24"-HBB-BF-MO-57-115, 135, and 147
   24"-HBB-BF-AO-57-114, 123, and 124
   Clow Job No. 82-2053-06(N)-01, 02, & 03
   82-2053-07(N)-01, 02, & 03

B. Seismic Qualification Test Plans and Reports

prepared by: National Technical Systems
Testing Division
Saugus, California

1. Nuclear Test Procedure No. 528-0951 Rev. A (Jan. 83)
   "Seismic Qualification Testing of Butterfly Valves and Actuators"

REFERENCES (con't)

2. Nuclear Test Report 528-0951 Rev. A (May 83)
"Seismic Qualification Testing of Butterfly Valves and Actuators"

prepared by: Wyle Laboratories
Huntsville, Alabama

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

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

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

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

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

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

C. Air Flow Tests

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

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

REFERENCES (con't)

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

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

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

D. Other Reports and Information

1. Operating Instructions for Clow Tricentric Wafer Stop Valve covers installation, maintenance, and operating instructions for 82-2053(N) valves.

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

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

E. Other References

1. Bechtel Power Corp. Design Specification 8031-P-144, Rev. 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. "A Parametric Study of a Butterfly Valve Utilizing the Hydraulic Analogy" by Bruce A. Coers; Thesis for fulfillment of Master of Science in Mechanical Engineering requirements, Graduate School of Bradley University, Peoria, Ill., 1983.

APPENDIX A

NUCLEAR REGULATORY

PURGE VALVE

OPERABILITY

GUIDE LINES

BRANCH TECHNICAL POSITION CSB 6-4 *

CONTAINMENT PURGING DURING NORMAL PLANT OPERATIONS

A. BACKGROUND

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

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

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

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

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

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

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

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

Purge system designs that are acceptable for use on 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. BRANCH TECHNICAL POSITION

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

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

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

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

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

   d. The containment isolation provisions for the purge system lines should meet the standards appropriate to engineered safety features; e.g., 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 conseq-uences should be within 10 CFR 100 guideline values.

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

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

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

GUIDELINES FOR DEMONSTRATION OF OPERABILITY OF PURGE AND VENT VALVES

OPERABILITY

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

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

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

DEMONSTRATION

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

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

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

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

Bench Testing

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

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

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

3. Whether the following load and environmental factors were considered

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

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

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

In-Situ Testing

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

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

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

End CSB 6-4

CLARIFICATION OF SEPT. 27 LETTER TO LICENSEES REGARDING *

DEMONSTRATION OF OPERABILITY OF PURGE AND VENT VALVES

1. The ΔP across the valve is in part predicated on the containment pressure and gas density conditions. What were the containment conditions used to determine the ΔP's across the valve at the incremental angle positions during the closure cycle?

2. Were the dynamic torque coefficients used for the determination of torques developed, based on data resulting from actual flow tests conducted on the particular disc shape/design/size? What was the basis used to predict torques developed in valve sizes different (especially larger valves) than the sizes known to have undergone flow tests?

3. Were installation effects accounted for in the determination of dynamic torques developed? Dynamic torques are known to be affected for example, by flow direction through valves with off-set discs, by downstream piping backpressure, by shaft orientation relative to elbows, etc. What was the basis (test data or other) used to predict dynamic torques for the particular valve installation?

4. When comparing the containment pressure response profile against the valve position at a given instant of time, was the valve closure rate vs. time (i.e. contstant or other) taken into account? For air operated valves equipped with spring return operators, has the lag time from the time the

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

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

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

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

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

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

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

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

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

11. For valve assemblies requiring a seal pressurization system (inflatable main seal) describe the air pressurization system configuration and operation including means used to determine that valve closure and seal pressurization have taken place. Discuss active electrical components in this system, and the basis used to determine their qualification for the environmental condition experienced. Is this system seismically designed.

    For this type valve, has it been determined that the "valve travel stops" (closed position) are capable of withstanding the loads imposed at closure during the DBA-LOCA conditions.

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

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

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

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

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

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

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

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

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

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

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

APPENDIX B

SUMMARY OF 12" CLOW TRICENTRIC CHOKED FLOW/STATIC SEISMIC OPERABILITY TEST

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

APPENDIX B

DESCRIPTION OF OPERATIONAL TESTS

OF A 12 INCH CLOW TRICENTRIC VALVE

FOR

NUCLEAR PURGE SYSTEM SERVICE

BY

J. E. KRUEGER
NUCLEAR VALVE DESIGN ENGINEER

NOVEMBER 30, 1981

INTRODUCTION -

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

OBJECTIVE -

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

TEST SET-UP -

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

VALVE AND ACTUATOR DESIGN PARAMETERS -

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

CONDUCT OF TEST -

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

RESULTS OF TESTS -

The tests demonstrated the following:

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

CONCLUSION -

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

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

APPENDIX C

BULLETIN FC-77

FAST CLOSING VALVE OPERATORS

An Industry Problem... Speed Versus Torque/Thrust

Prepared by

LIMITORQUE CORPORATION

PREFACE

Since the advent of nuclear power, there has been an ever increasing demand on valve operator manufacturers to build electric actuators with faster operating times. This demand has, for the most part, been satisfied with very slight modifications to existing, popular models, such as the Limitorque SMB or SMC.

The use of existing and proven equipment incorporating these minor modifications has given all of the valve companies, engineering consultants and end users the confidence necessary to continue to place ever increasing demands on Limitorque for research and development to find the limits to which present designs might be utilized to obtain the faster closing speeds of tomorrow.

APPLICATION

Most valve applications are either torque seated or at least protected in the closing direction by the use of a torque sensing device called the torque switch. This switch (shown in Fig. 1) operates by measuring the amount of compression of a fixed pre-calibrated belleville spring (the torque spring). As a load developes (either a seating load or an obstruction in the valve) and the output worm gear is prevented from rotating, a thrust is generated on the worm causing it to compress the torque spring. This action takes place in either direction of rotation of the output worm gear.

The amount of compression of the torque spring is an accurate measurement of output torque as we have reduced the forces to a linear spring force (F) acting on a torque arm (TA) to produce a torque (T) at the valve stem (Fig. 1). The magnitude of the compression of the torque spring is empirically calibrated in pounds foot and a nameplate is affixed to each switch compartment denoting the calculated torque output at that setting. The maximum setting allowed by Limitorque for the particular application is also marked on the torque switch calibration tag.

Figure 1

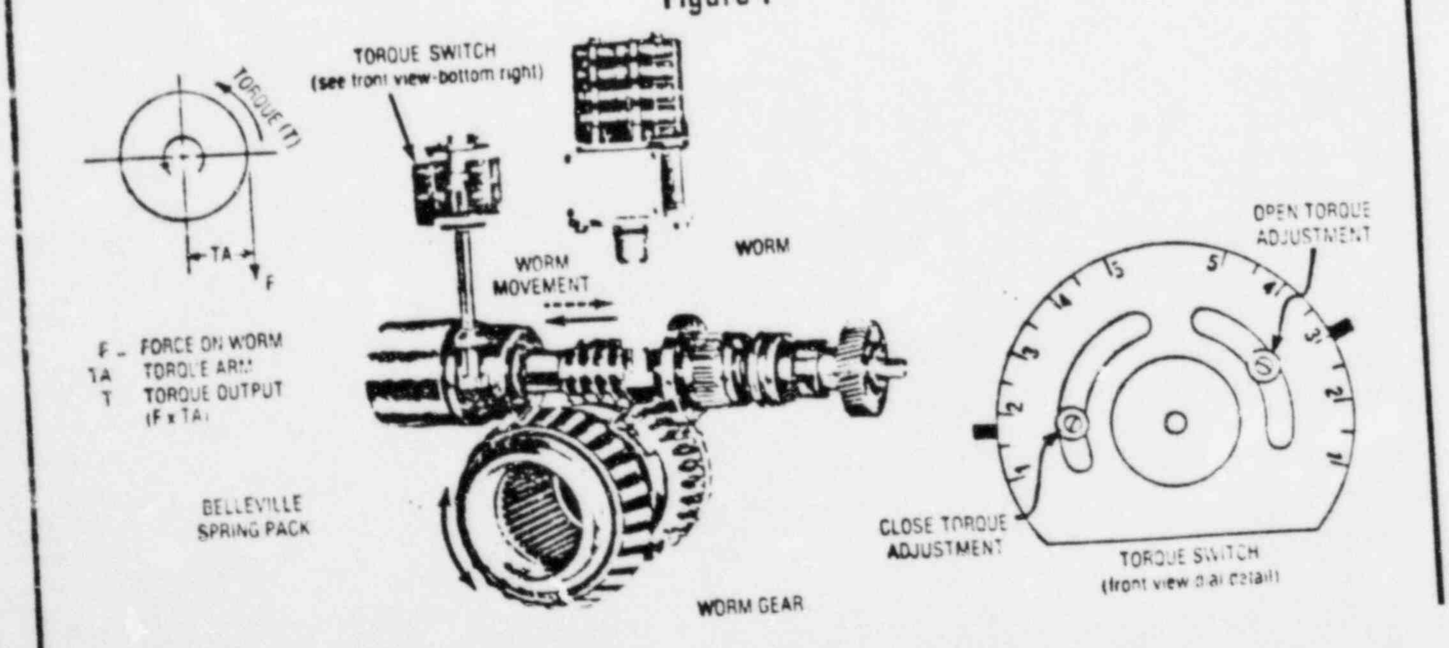
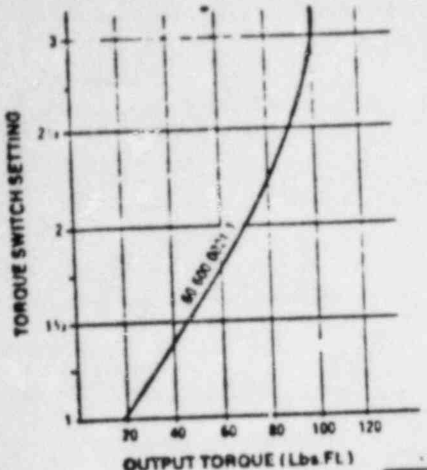
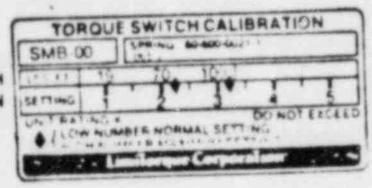


Figure 2





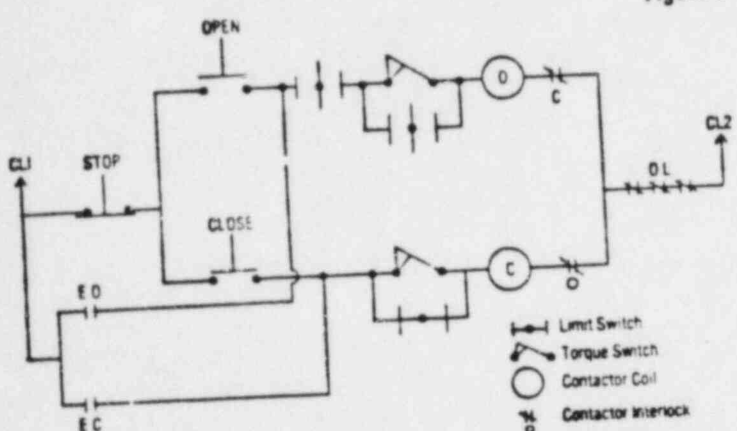
TYPICAL TORQUE SWITCH CALIBRATION

Fig. 2 shows an example of a torque switch calibration tag for a valve operator designed to deliver 75 lbs.-ft. of torque with a maximum safe operator capability of 105 lbs.-ft. Limitorque also installs a torque limiter plate on the torque switch to physically prevent someone from setting the torque switch to a setting higher than the maximum computed capability of the unit.

EFFECTS OF STARTER DROPOUT

The torque switch, although a mechanical device, is an electrical control element. It is wired in series with either a pushbutton or relay and the reversing starter coils (Fig. 3). To stop the motor in the "close" direction, the torque switch contacts must open and the motor starter coil de-energizes. This breaks the starter contacts and disconnects the electrical power to the motor. Tests have indicated that the starter dropout time (the time from the torque switch trip point to zero voltage on the motor) averages between 25 to 50 milliseconds depending on the size of the reversing starter and its physical construction.

Figure 3



In the time between the torque switch trip and the motor starter drop-out (25 to 50 milliseconds), the motor develops torque proportional to the resistance of the valve. This means that the more rigid the valve system, the faster the motor approaches its pullout torque rating. At normal stem speeds of 12"/min. on gate valves and 4"/min. on globe valves or less and an output of 50 rpm or less of the valve operator, the 25 to 50 milliseconds drop-out time of the motor starter has a negligible effect on the total output torque of the valve operator. As the speed of the operator increases, the starter drop-out time becomes a very appreciable factor in the final output torque at any given torque switch setting as the unit travels further in this time interval.

Figure 4

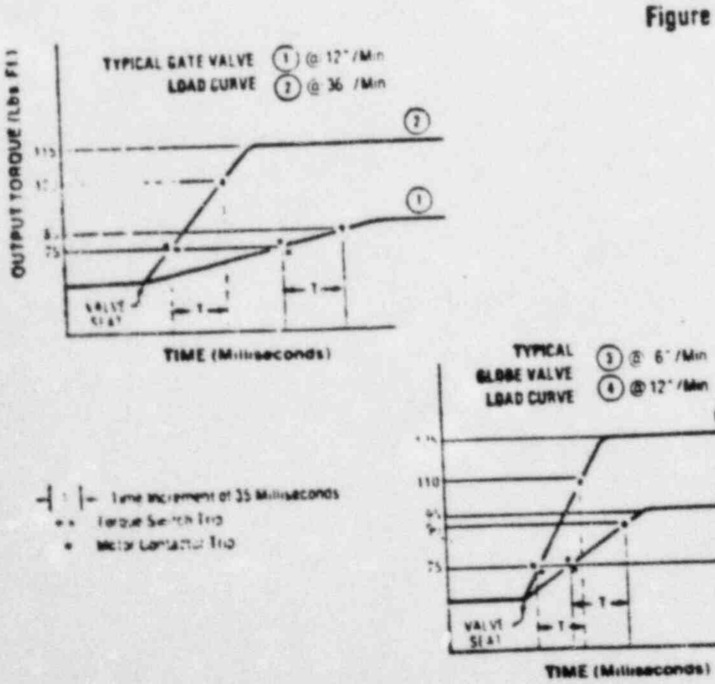


Fig. 4 illustrates a gate valve (Curve 1) at 12"/min. and 36 rpm output of the valve operator against Curve 2 for the same unit at 36"/min. and 108 rpm output of the valve operator. Similar curves for globe valve operators (Curves 3 & 4) at stem speeds of 6" and 12"/min. respectively are shown. The torque switch in all four cases trips at the predetermined output torque (75 lbs.-ft.) shown as (**). The motor starter in each case requires 35 milliseconds to break the motor voltage. The output torque plotted at the motor starter trip point is indicated with an (*).

EFFECTS OF EFFICIENCY

The efficiency variation of valve actuators depends upon two primary factors, gear ratio and worm

speed. The lower the gear ratio, the higher the efficiency and also the higher the worm speed, the higher the efficiency. Fig. 5 demonstrates the efficiency variation of valve actuator gearing illustrating the difference between self-locking (high ratios) and non-locking (low ratios). It can be seen from Fig. 5 that the efficiency of actuators at higher worm speeds is much greater than at lower worm speeds.

The rigidity of the system determines the slope of the load curve (torque) as plotted in Fig. 4. It can be seen that the more rigid the system, the steeper the slope, as illustrated by the difference between gate (Curves 1 & 2) and globe (Curves 3 & 4) valve curves where the average globe valve is substantially more rigid than the comparable gate valve. This rigidity determines the speed of the worm existing at the time the torque switch activates (the steeper the curve, the higher the speed of the worm). For example, if a valve operator having a self-locking worm gear set was used to compress a set of springs until the torque switch tripped (at 75 lbs.-ft.), the load curve would look like Curve A in Fig. 6. If the same unit was operated against a rigid structure instead of the springs, the load curve would look like Curve B in Fig. 6. The difference in the motor current draw is attributable to the fact that the motor on Curve A had begun to be loaded gradually, approaching its torque rating, causing it to decelerate and thereby decrease the worm speed, thus lowering the unit efficiency. On the other hand, the operator in Curve B had its motor instantly loaded and before the motor speed dropped any appreciable amount, the unit tripped the torque switch, hence, the higher efficiency due to higher worm speeds.

The standard Acme thread utilized by valve companies today also varies in efficiency as speed increases. Acme thread efficiency is greatly affected by surface finish, cleanliness and lubrication. It is not uncommon to find an increase in stem efficiency of 10-15% at yoke nut speeds greater than 100 rpm.

EFFECTS OF INERTIA

Fast operating times necessarily bring with them inertial loads which are always additive to the increased load due to the motor starter drop-out lag. The magnitude of this inertial load, like operator efficiency, depends on the operator speed and the rigidity of the system. It is possible to have a valve operator traveling at 120 to 144"/min. in a rigid system to experience a 1-1/2 to 2-1/2 load magnification factor due to inertia alone. Unfortunately, it is impossible to determine this load magnification factor empirically before the operator is installed on the valve.

Suggestions have been received through the years as to the best method of handling the increases in efficiency and inertia on high speed actuators. A tempting solution is to reduce the size of the actuator and utilize some of the higher efficiencies and inertial loads during seating. This solution is, from experience, a fruitless venture as it has been painfully proven that the additive factors of efficiency and inertia being relied upon so heavily are not always present. The best, and applications proven method to handle high speed actuators is to control these magnification factors to the point of predictability.

Figure 5
TYPICAL UNIT EFFICIENCIES

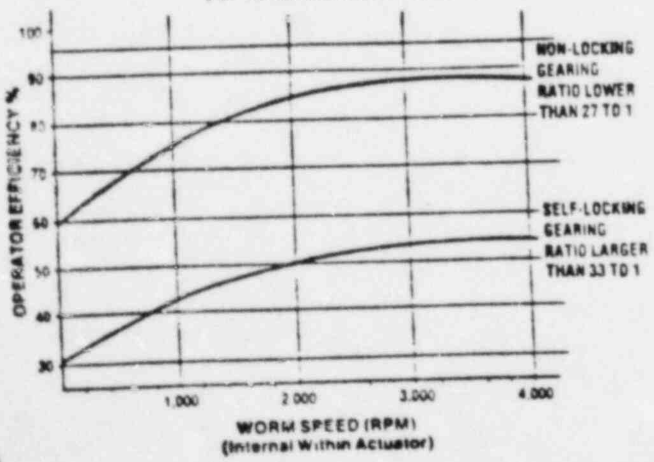
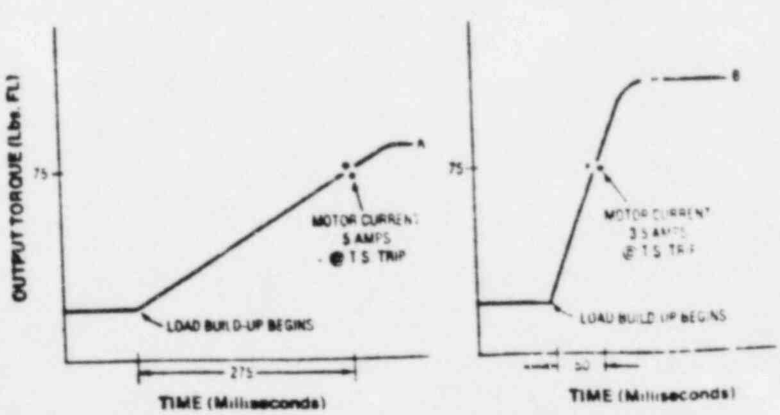


Figure 6
EFFECT OF SYSTEM RIGIDITY ON OPERATOR OUTPUT TORQUE



Since 1956, Limitorque has studied ways in which torque spring calibration and inertial loading for high speed operators could be handled in a manner in which they would be similar in application to slow speed operators. The SA type unit, which was a modified SMA incorporating inertia absorbing belleville springs in the drive sleeve assembly, was the culmination of these studies. The SA unit along with its SB and SC counterparts allows the stem nut to float in one or both directions to absorb the impact of seating (or backseating) by reducing rigidity in the system. If a valve system was defined as Curve B, Fig. 6, with the use of the standard SMB or SMC actuator, it would become Curve A with the use of an SB or SC type unit.

The SB or SC actuators provide a way to increase the controlability and predictability of high speed applications. This is accomplished by minimizing, if not eliminating all of the adverse elements which have been discussed previously:

(1) Motor starter drop-out time has minimal effect with an SB or SC unit as the load generated by the overtravel of the actuator during this time period is primarily absorbed by the springs rather than the valve (because of its flexibility as compared to the valve flexibility).

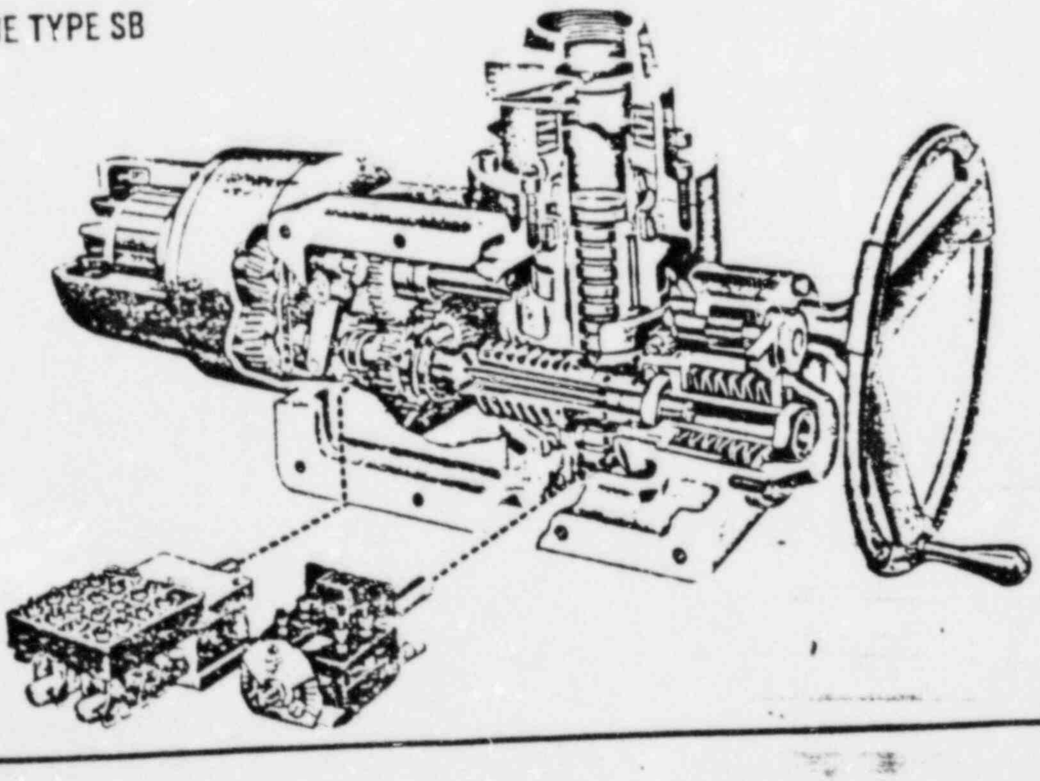
(2) Increased operator and stem efficiencies are greatly reduced as the loads are gradually applied because of the spring compression and begin to approach the minimal efficiencies used in selections.

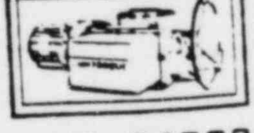
(3) Inertia is greatly minimized in most applications and completely eliminated in the remainder.

Research is continuing in an effort to improve the predictability of the many variables in high speed actuator applications. Many valve companies are contributing to this program to increase the compatibility of the valve actuator to the valve. A notable advance which has developed from the research on high speed is that the use of the energy absorbing SB or SC type unit even in standard speed applications increases the valve life substantially. This life increase is due to the soft, even application of seating thrust when closing the valve.

Information concerning the Limitorque SB or SC units may be obtained from any of the local Limitorque sales offices or by writing to Limitorque Corporation, Lynchburg, VA 24502.

LIMITORQUE TYPE SB





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