# PURGE AND VENT VALVE OPERABILITY

# QUALIFICATION ANALYSIS

Prepared for:

Jersey Central Power and Light Co.
General Public Utilities
Oyster Creek Nuclear Generating Station

80-8170-01, 02, 03, 04 and 05

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REVIEWED BY [illegible] 4/23/82

TECHNICAL DIRECTOR DATE

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Report No. 4-01-82

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JERSEY CENTRAL POWER AND LIGHT CO.
GENERAL PUBLIC UTILITIES
OYSTER CREEK NUCLEAR GENERATING STATION

by

James E. Krueger

Robert C. Sansone

March 1982

Work performed under GPU Purchase Order Number 72016

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This report covers Valve Mark Nos: V-23-13, V-23-14, V-23-15, V-23-16, V-28-17, V-28-18, V-27-1, V-27-2, V-27-3, V-27-4, V-26-16, and V-26-18

# REVISIONS

CLOW CORPORATION

ENGINEERED PRODUCTS DIVISION

REPORT NO. 4-01-82

DATE: June 7, 1982

| REV. NO | DATE | PAGES AFFECTED | BY | APPL. | DESCRIPTION OF CHANGES |
|---|---|---|---|---|---|
| A | 6/7/82 | 36 | JEK | | 18" Failure Mode should be closed |
| A | 6/7/82 | 36 | JEK | | 20" Failure Mode should be open |
| A | 6/7/82 | 66 | JEK | | Line 13 should be margins |

CERTIFICATION

This is to certify that all valves (Mark Nos. V-23-13 thru V-23-16, V-28-17, V-28-18, V-27-1 thru V-27-4, V-26-16 and V-26-18) have been evaluated for operability under the installed conditions indicated in Jersey Central Power and Light Company's Procurement Specification 492-7, Rev. 3 dated 7/29/81. 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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TABLE OF CONTENTS

| | Page |
|---|---|
| LIST OF TABLES | iv |
| LIST OF FIGURES | vi |
| 1.0 INTRODUCTION | 1 |
| 2.0 DESIGN OF VALVE AND ACTUATOR ASSEMBLY | 3 |
| 2.1 Valve Design | |
| 2.1.1 Geometry | 3 |
| 2.1.2 Materials | 6 |
| 2.1.3 Operation | 13 |
| 2.2 Actuator Design | |
| 2.2.1 Geometry | 17 |
| 2.2.2 Actuator Design Materials | 22 |
| 2.2.3 Actuator and Valve Operation | 23 |
| 2.2.3.1 Actuators and Accessories Supplied | 23 |
| 2.2.3.2 Actuator Output Torques | 26 |
| 2.2.3.3. Operating Time | 33 |
| 3.0 VALVE OPERATING AND INSTALLATION REQUIREMENTS | 34 |
| 3.1 Valve Operating Conditions | 34 |
| 3.2 Valve Installation Configurations | 37 |
| 4.0 VALVE STRUCTURAL INTEGRITY UNDER SEISMIC AND OPERATIONAL LOADINGS | 42 |
| 4.1 Valve Frequency and Stress Analysis | 42 |
| 4.2 Clow Tricentric Valve Assembly Resonant Frequency Test | 43 |

TABLE OF CONTENTS


| | Page |
|---|---|
| 4.3 Asco Solenoid Valve Resonant Frequency Test | 43 |
| 4.4 Static Load Test During Simulated LOCA Flow | 44 |
| 4.5 Fatigue Analysis | 55 |
| 5.0 VALVE AERODYNAMIC TORQUES | 58 |
| 5.1 Model Tests | 59 |
| 5.1.2 Tests With an Upstream Elbow | 65 |
| 5.1.3 Downstream Piping Effects | 72 |
| 5.2 Model Data Verification | 73 |
| 5.3.0 Application of Model Aerodynamic Test to Full Size Valve Operability | 74 |
| 5.3.1 Valve Operating Times Expected in Service | 74 |
| 5.3.2 Aerodynamic Torques for Valves as Installed | 75 |
| 5.3.3 Conclusions Concerning Valve Operability | 93 |
| 6.0 VALVE SEALING CHARACTERISTICS | 95 |
| 6.1 Normal Sealing | 95 |
| 6.2 Long Term Sealing | 97 |
| 6.3 Debris Effects on Sealing | 98 |
| 6.4 Sealing Under Temperature Variations | 99 |
| 7.0 REFERENCES | 100 |
| APPENDIX A | |
| APPENDIX B | |
| APPENDIX C | |

## LIST OF TABLES


| TABLE | TITLE | PAGE |
|---|---|---|
| 1 | ACTUATOR ACCESSORIES | 25 |
| 2 | GUARANTEED TORQUE RATIOS | 26 |
| 3 | VALVE BENCH TEST OPERATING TIMES | 33 |
| 4 | PLANT OPERATING AND SEISMIC LOADINGS FOR VALVES V-28-17 and V-28-18 | 35 |
| 5 | SEISMIC LOADINGS FOR ALL VALVES EXCEPT V-28-17 and V-28- 18 | 35 |
| 6 | PRESSURE DIFFERENTIALS APPLIED TO VALVES | 35 |
| 7 | ALLOWED SEAT LEAKAGE RATES | 36 |
| 8 | LOWEST VALVE RESONANT FREQUENCIES | 45 |
| 9 | CONDITION APPLIED FOR STRESS ANALYSIS | 45 |
| 10 | ALLOWED STRESS | 45 |
| 11 | MAXIMUM STRESS RATIO UPSET CONDITION 8" VALVE | 46 |
| 12 | MAXIMUM STRESS RATIO FAULTED CONDITION 8" VALVE | 47 |
| 13 | MAXIMUM STRESS RATIO UPSET CONDITION 12" VALVE | 48 |
| 14 | MAXIMUM STRESS RATIO EMERGENCY CONDITION 12" VALVE | 49 |
| 15 | MAXIMUM STRESS RATIO FAULTED CONDITION 12" VALVE | 50 |
| 16 | MAXIMUM STRESS RATIO UPSET CONDITION 18" VALVE | 51 |
| 17 | MAXIMUM STRESS RATIO FAULTED CONDITION 18" VALVE | 52 |
| 18 | MAXIMUM STRESS RATIO UPSET CONDITION 20" VALVE | 53 |
| 19 | MAXIMUM STRESS RATIO FAULTED CONDITION 20" VALVE | 54 |
| 20 | TEST VALVE SCALED SIZES (CRITICAL ELEMENTS) | 63 |

LIST OF TABLES (con't)


| TABLE | TITLE | PAGE |
|---|---|---|
| 21 | COMPARISON OF PRODUCTION VALVES TO VALVE MODEL SIZES (CRITICAL ELEMENTS) | 64 |
| 22 | FLOW DATA V-23-13 & 14 NORMAL OPERATING PRESSURE | 76 |
| 23 | FLOW DATA V-23-13 & 14 MAX. OPERATING PRESSURE | 77 |
| 24 | FLOW DATA V-23-15 & 16 NORMAL OPERATING PRESSURE | 78 |
| 25 | FLOW DATA V-23-15 & 16 MAX. OPERATING PRESSURE | 79 |
| 26 | FLOW DATA V-28-17 & 18 NORMAL OPERATING PRESSURE | 80 |
| 27 | FLOW DATA V-28-17 & 18 MAX. OPERATING PRESSURE | 81 |
| 28 | FLOW DATA V-27-1,2,3,4 NORMAL OPERATING PRESSURE | 82 |
| 29 | FLOW DATA V-27-1,2,3,4 MAX. OPERATING PRESSURE | 83 |
| 30 | FLOW DATA V-26-16 & 18 NORMAL OPERATING PRESSURE | 84 |
| 31 | FLOW DATA V-26-16 & 18 MAX. OPERATING PRESSURE | 85 |
| 32 | VALVE NO. V-23-16 (8") PREDICTED TORQUE | 88 |
| 33 | VALVE NO. V-23-15 (8") PREDICTED TORQUE | 88 |
| 34 | VALVE NO. V-26-16 & 18 (20") PREDICTED TORQUE | 89 |
| 35 | VALVE NO. V-23-14 (8") PREDICTED TORQUE | 89 |
| 36 | VALVE NO. V-23-13 (8") PREDICTED TORQUE | 90 |
| 37 | VALVE NO. V-27-3 (18") PREDICTED TORQUE | 90 |
| 38 | VALVE NO. V-27-1,2, 4 (18")PREDICTED TORQUE | 91 |
| 39 | VALVE NO. V-28-17 (12") PREDICTED TORQUE | 91 |
| 40 | VALVE NO. V-28-18 (12") PREDICTED TORQUE | 92 |
| 41 | VALVE SEALING CHARACTERISTICS | 96 |

# LIST OF FIGURES


| FIGURE | TITLE | PAGE |
|---|---|---|
| 1 | TRICENTRIC VALVE OFFSETS | 4 |
| 2 | 8" VALVE ASSEMBLY AND MATERIALS | 8 |
| 3 | 8" VALVE ASSEMBLY AND MATERIALS | 9 |
| 4 | 12" VALVE ASSEMBLY AND MATERIALS | 10 |
| 5 | 18" VALVE ASSEMBLY AND MATERIALS | 11 |
| 6 | 20" VALVE ASSEMBLY AND MATERIALS | 12 |
| 7 | DISC WITH CLOSING FORCES APPLIED | 15 |
| 8 | ACTUATOR SCOTCH YOKE DESIGN | 18 |
| 9 | TYPICAL TORQUE OUTPUT FOR DOUBLE ACTING SCOTCH YOKE ACTUATOR | 20 |
| 10 | FAIL SAFE, SPRING RETURN ACTUATOR DESIGN | 20 |
| 11 | TYPICAL TORQUE OUTPUT CURVES FOR A SPRING RETURN ACTUATOR | 21 |
| 12A | CALCULATED TORQUE DATA HD-732SR80 | 27 |
| 12B | CALCULATED TORQUE PLOT HD-732-SR80 | 28 |
| 13A | CALCULATED TORQUE DATA T-316 SR2 | 29 |
| 13B | CALCULATED TORQUE PLOT T-316-SR2 | 30 |
| 14A | CALCULATED TORQUE DATA T-430-SR2 | 31 |
| 14B | CALCULATED TORQUE PLOT T-420-SR2 | 32 |
| 15 | INSTALLED ORIENTATION OF 8" VALVES V-23-15 & 16 | 38 |
| 16 | INSTALLED ORIENTATION OF 20" VALVE V-26-16 | 38 |

LIST OF FIGURES (con't)


| FIGURE | TITLE | PAGE |
|---|---|---|
| 17 | INSTALLED ORIENTATION OF 8" VALVES V-23-13 & 14 | 39 |
| 18 | INSTALLED ORIENTATION OF 20" VALVE V-26-18 | 40 |
| 19 | INSTALLED ORIENTATION OF 18" VALVES V-27-3 & 4 | 40 |
| 20 | INSTALLED ORIENTATION OF 18" VALVES V-27-1 & 2 | 41 |
| 21 | INSTALLED ORIENTATION OF 12" VALVES V-28-17 & 18 | 41 |
| 22 | WATER TABLE STUDY OF CHOKED FLOW PATTERN WITH DISC FULL OPEN ($90^{o}$) | 67 |
| 23 | WATER TABLE STUDY OF CHOKED FLOW PATTERN WITH DISC PARTIALLY OPEN ($60^{o}$) | 68 |
| 24 | WATER TABLE STUDY OF CHOKED FLOW PATTERN WITH DISC PARTIALLY OPEN ($40^{o}$) | 69 |
| 25A | TEE WITH FLOW FROM TWO SIDES | 70 |
| 25B | TEE WITH FLOW FROM ONE SIDE | 70 |
| 26 | VALVE ORIENTATIONS RELATIVE TO UPSTREAM ELBOW | 71 |

# 1. INTRODUCTION

The Nuclear Regulatory Commission has, since 1979, been highly concerned about the operability of purge and vent valves during certain postulated occurrences. Their study in this area has shown that many valves were designed only to operate under normal flow requirements. For a postulated loss of coolant accident, such valves may fail to close in the time required to prevent discharge of radioactive gases to the outside environment. Such a failure could exceed 10 CFR 100 guidelines and present a significant hazard to the health of persons in the area. NRC Branch Technical Position CSB 6-4 gives some background on operations of purge and vent systems and basic requirements for their design. For the valves used in such systems, further guidelines are provided in "Guidelines for Demonstration of Operability of Purge and Vent Valves", which was provided to nuclear plant operators by an NRC letter of September 27, 1979. This set of guidelines covers twenty-one points (less two) which are to be addressed by the plant operator. This paper addresses those items which may be answered by the valve manufacturer based on the conditions provided by the plant operator for the postulated loss of coolant accident.

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

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

## 2.0 DESIGN OF VALVE AND ACTUATOR ASSEMBLY

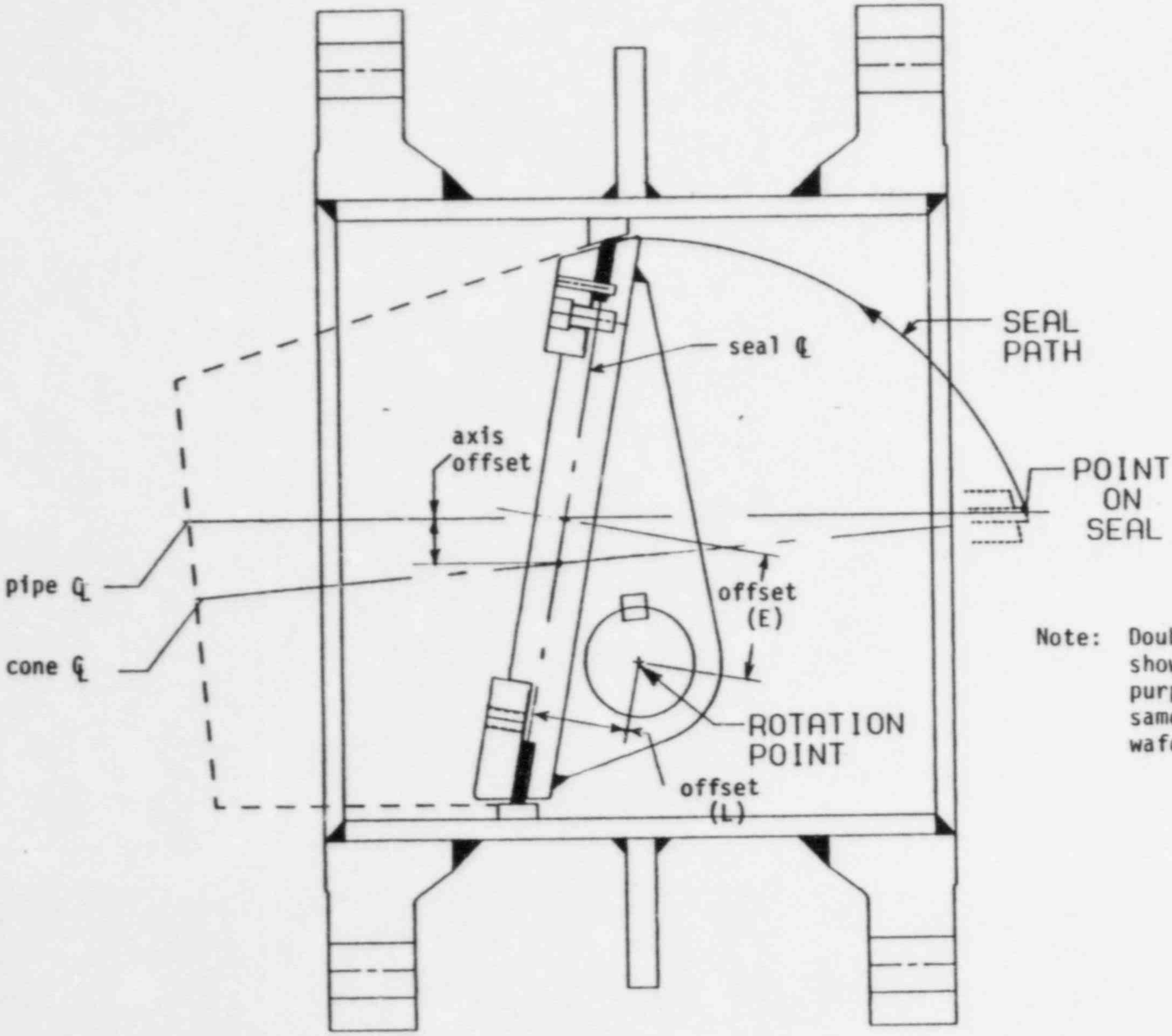
### 2.1 Valve Design

#### 2.1.1 Geometry

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

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

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




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

FIGURE 1 - TRICENTRIC VALVE OFFSETS

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

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

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

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

2.1.2 Materials

A complete list of valve component materials used on G.P.U.* Purchase Order Number 72016 may be found on the General Arrangement Drawings (D-0650 thru D-0654) which follow this section.

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

* General Public Utilities

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 (SA515 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 material is 1/16 inch thick as is the 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 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.

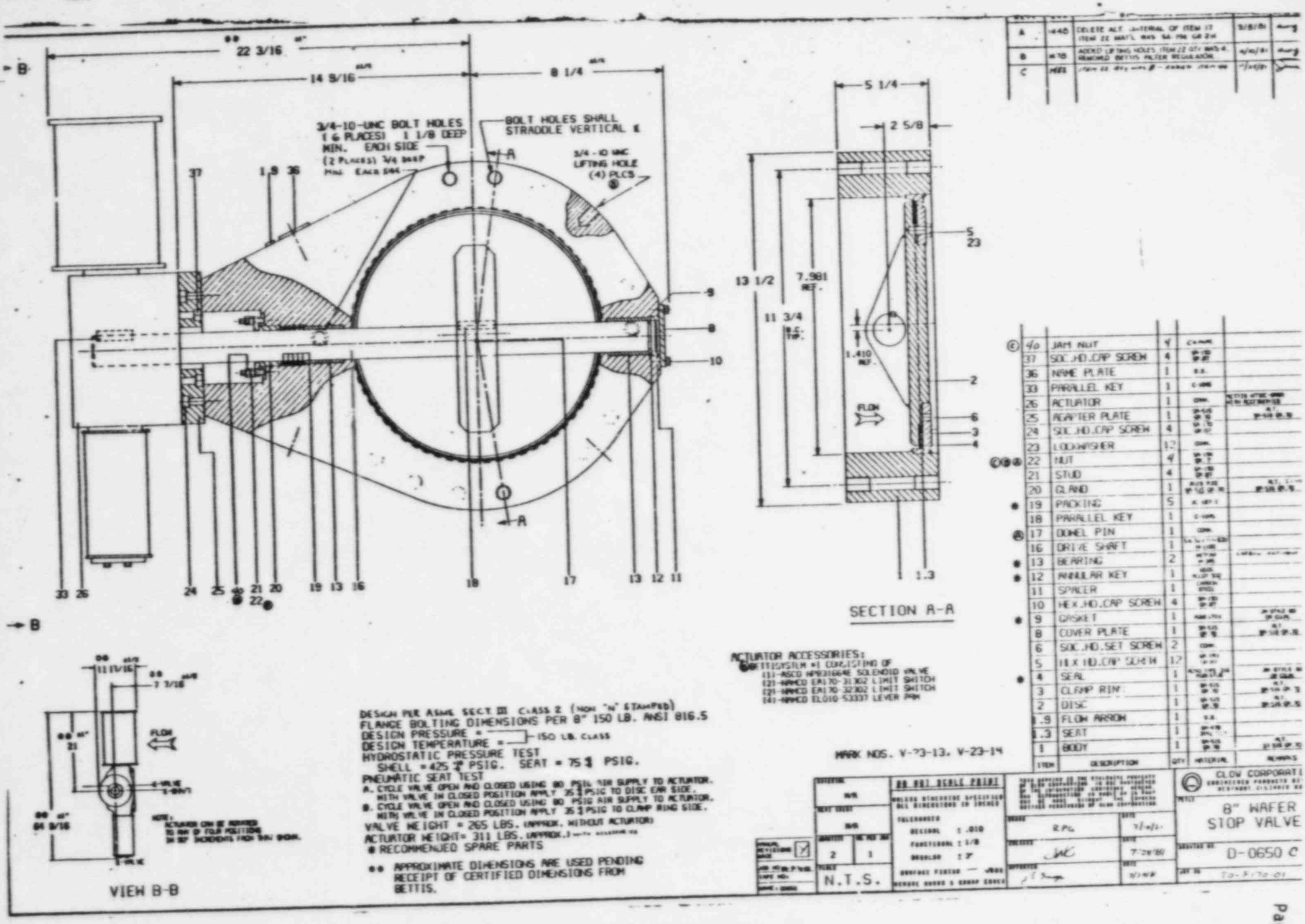



FIGURE 2 - 8" VALVE ASSEMBLY AND MATERIALS

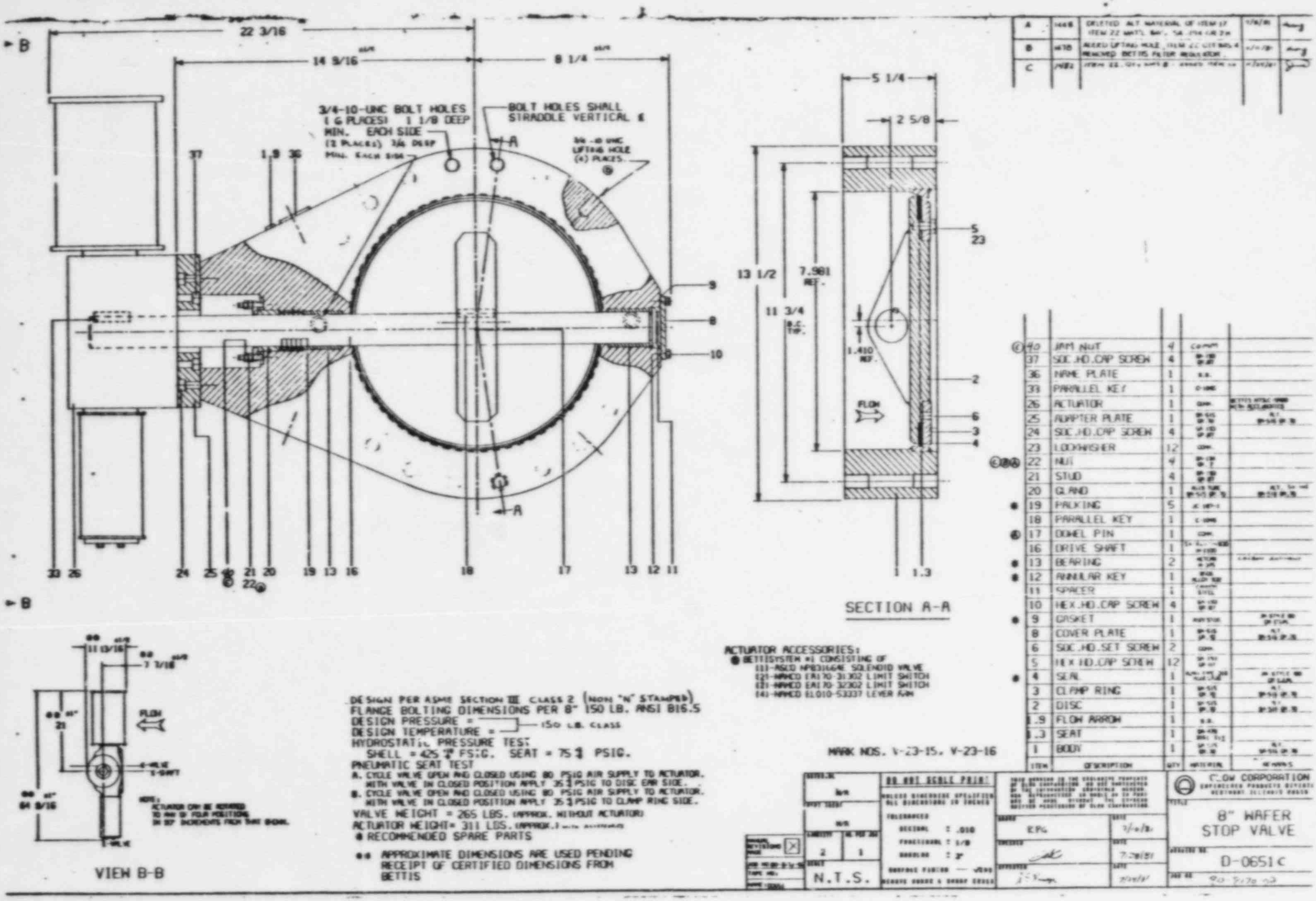
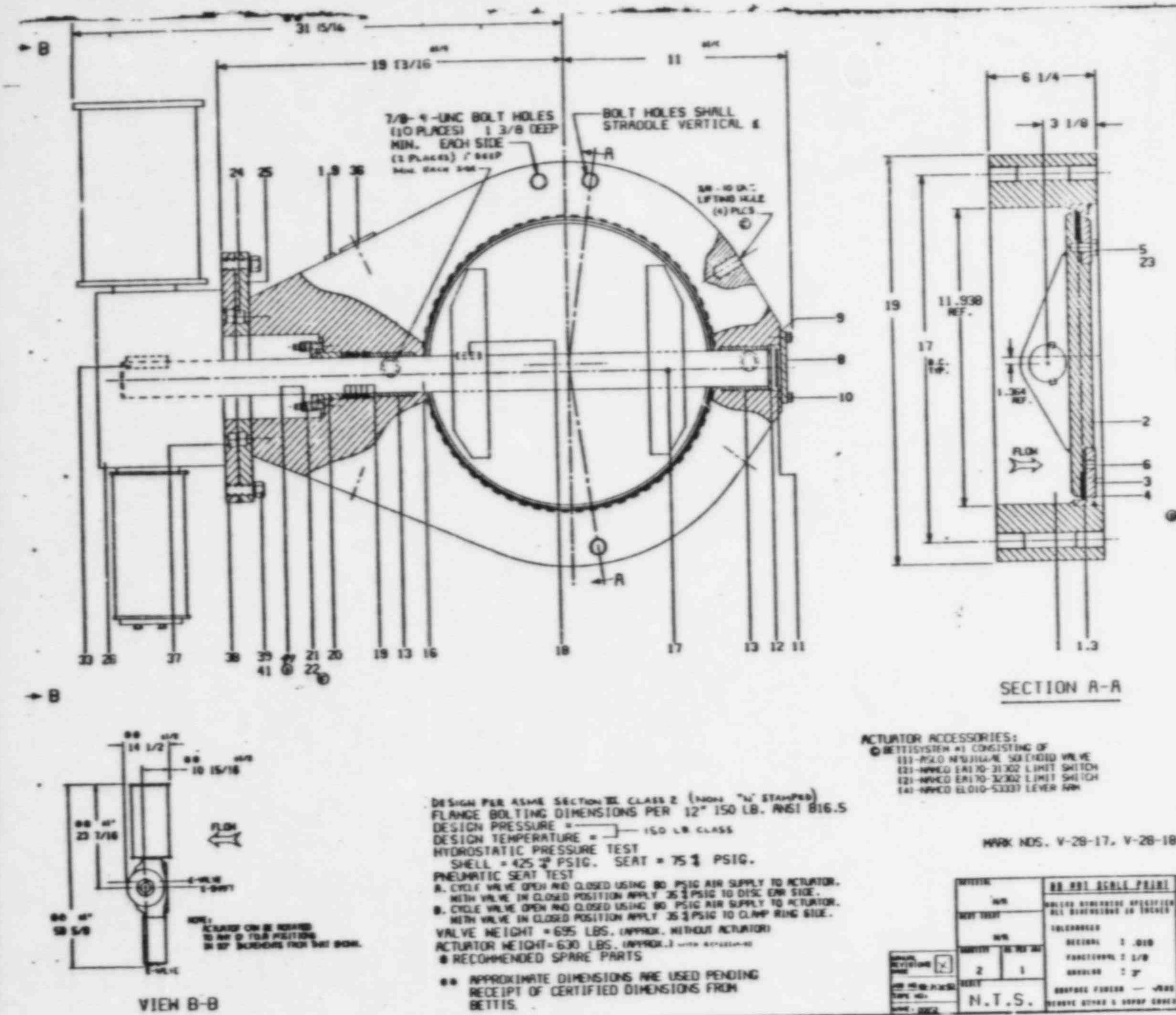



FIGURE 3 - 8" VALVE ASSEMBLY AND MATERIALS




| ITEM | DESCRIPTION | QTY | MATERIAL | REMARKS |
|---|---|---|---|---|
| 49 | JAM NUT | 4 | COMM. | |
| 41 | LOCKWIRE | A/R | S.S. | |
| 40 | | | | |
| 39 | HEX HD. CAP SCREW | 4 | [illegible] | LOCKWIRED TOGETHER |
| 38 | ADAPTER PLATE | 1 | [illegible] | [illegible] |
| 37 | SOC. HD. CAP SCREW | 4 | [illegible] | |
| 36 | NAME PLATE | 1 | S.S. | |
| 33 | PARALLEL KEY | 1 | [illegible] | |
| 26 | ACTUATOR | 1 | COM. | [illegible] |
| 25 | ADAPTER PLATE | 1 | [illegible] | [illegible] |
| 24 | SOC. HD. CAP SCREW | 4 | [illegible] | |
| 23 | LOCKWASHER | 16 | COM. | |
| 22 | NUT | 4 | [illegible] | |
| 21 | STUD | 4 | [illegible] | |
| 20 | GLAND | 1 | [illegible] | [illegible] |
| 19 | PACKING | 5 | [illegible] | |
| 18 | PARALLEL KEY | 2 | [illegible] | |
| 17 | DOWEL PIN | 1 | COM. | |
| 16 | DRIVE SHAFT | 1 | [illegible] | |
| 13 | BEARING | 2 | [illegible] | [illegible] |
| 12 | ANNULAR KEY | 1 | [illegible] | |
| 11 | SPACER | 1 | [illegible] | |
| 10 | HEX. HD. CAP SCREW | 4 | [illegible] | |
| 9 | GASKET | 1 | [illegible] | [illegible] |
| 8 | COVER PLATE | 1 | [illegible] | [illegible] |
| 6 | SOC. HD. SET SCREW | 2 | COM. | |
| 5 | HEX HD. CAP SCREW | 16 | [illegible] | |
| 4 | SEAL | 1 | [illegible] | [illegible] |
| 3 | CLAMP RING | 1 | [illegible] | [illegible] |
| 2 | DISC | 1 | [illegible] | [illegible] |
| 1.9 | FLOW ARROW | 1 | S.S. | |
| 1.3 | SEAT | 1 | [illegible] | |
| 1 | BODY | 1 | [illegible] | [illegible] |

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12" WAFER STOP VALVE

D-0652 D

FIGURE 4 - 12" VALVE ASSEMBLY AND MATERIALS

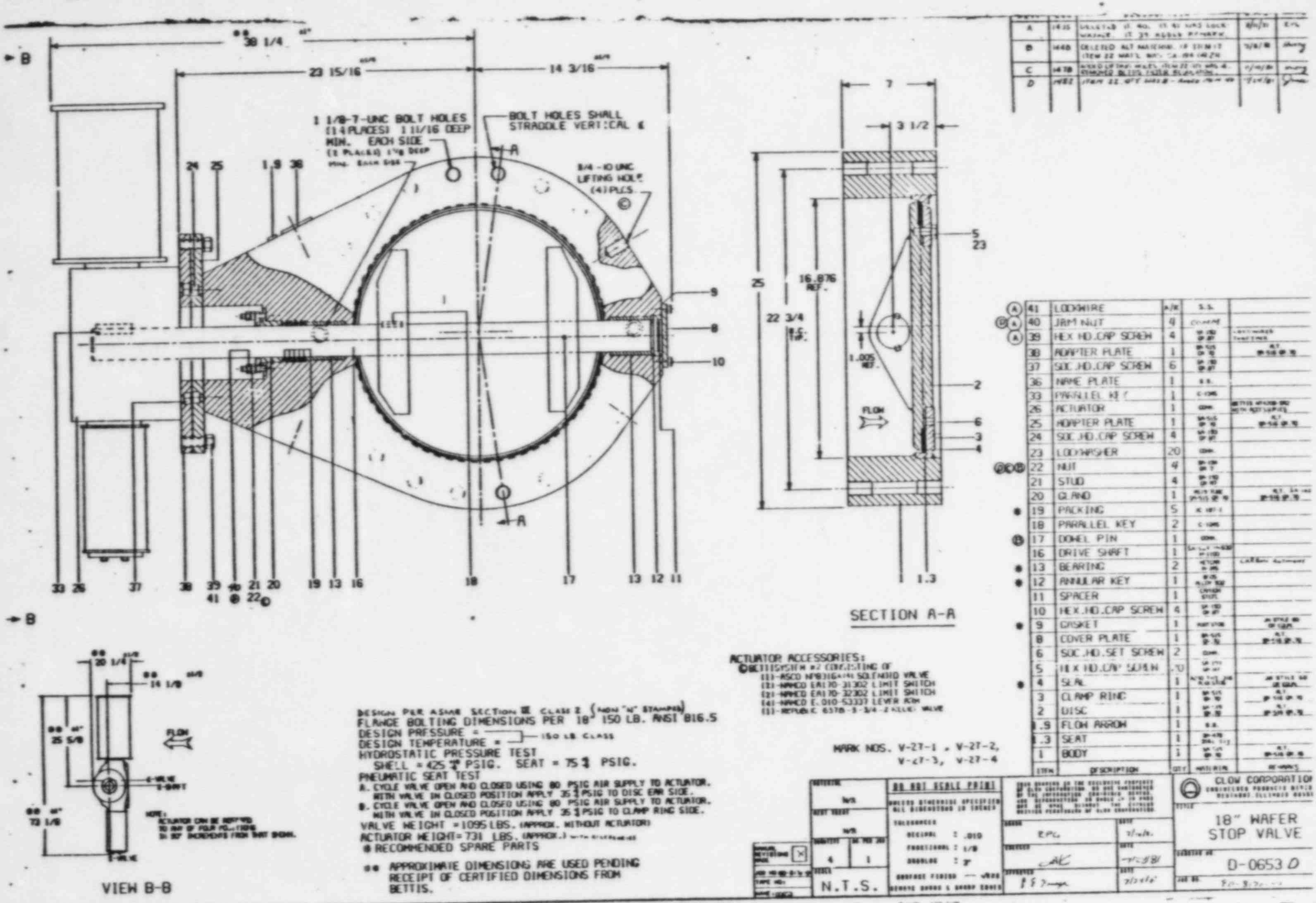



FIGURE 5 - 18" VALVE ASSEMBLY AND MATERIALS

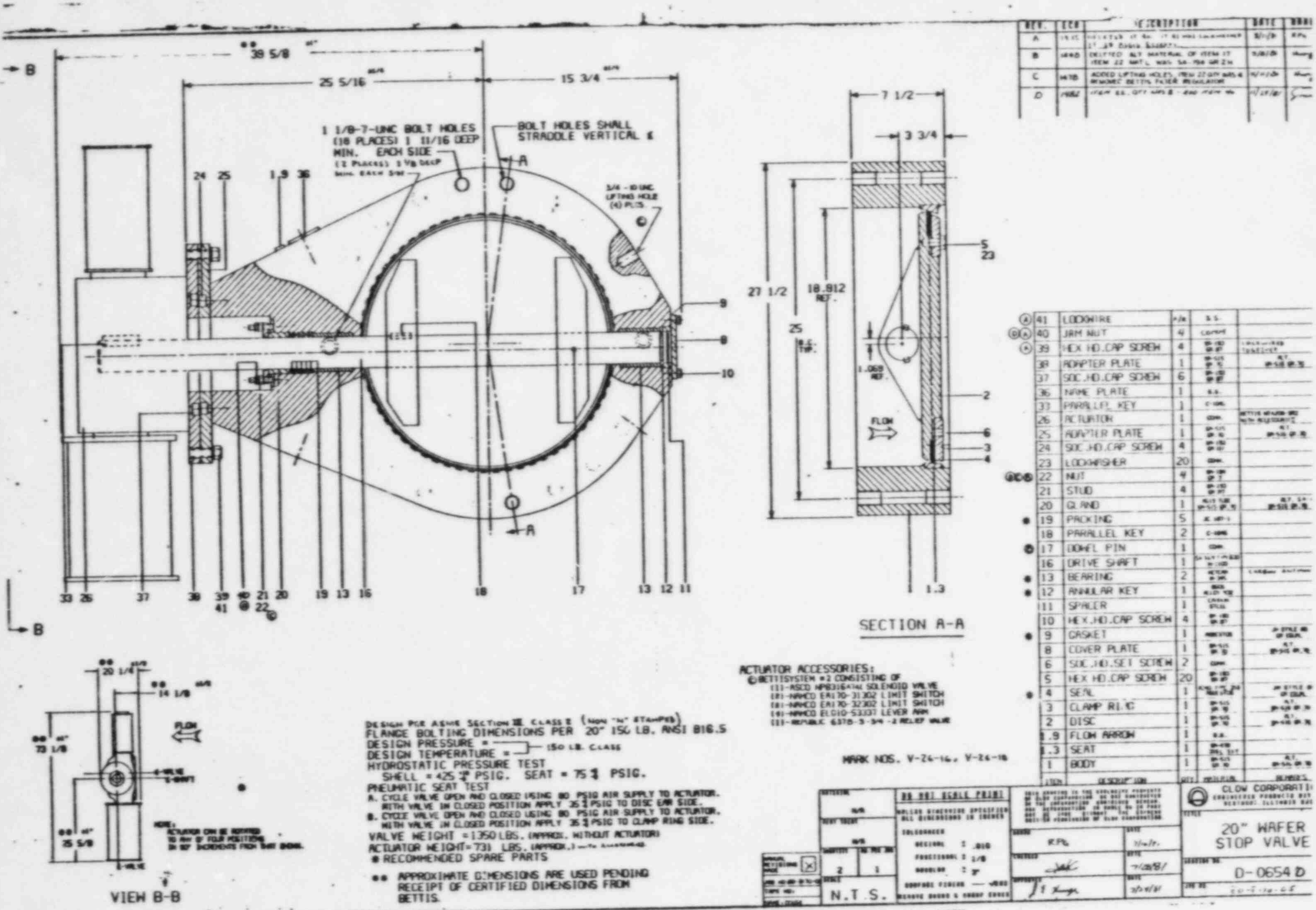



FIGURE 6 - 20" VALVE ASSEMBLY AND MATERIALS

2.1.3 Operation

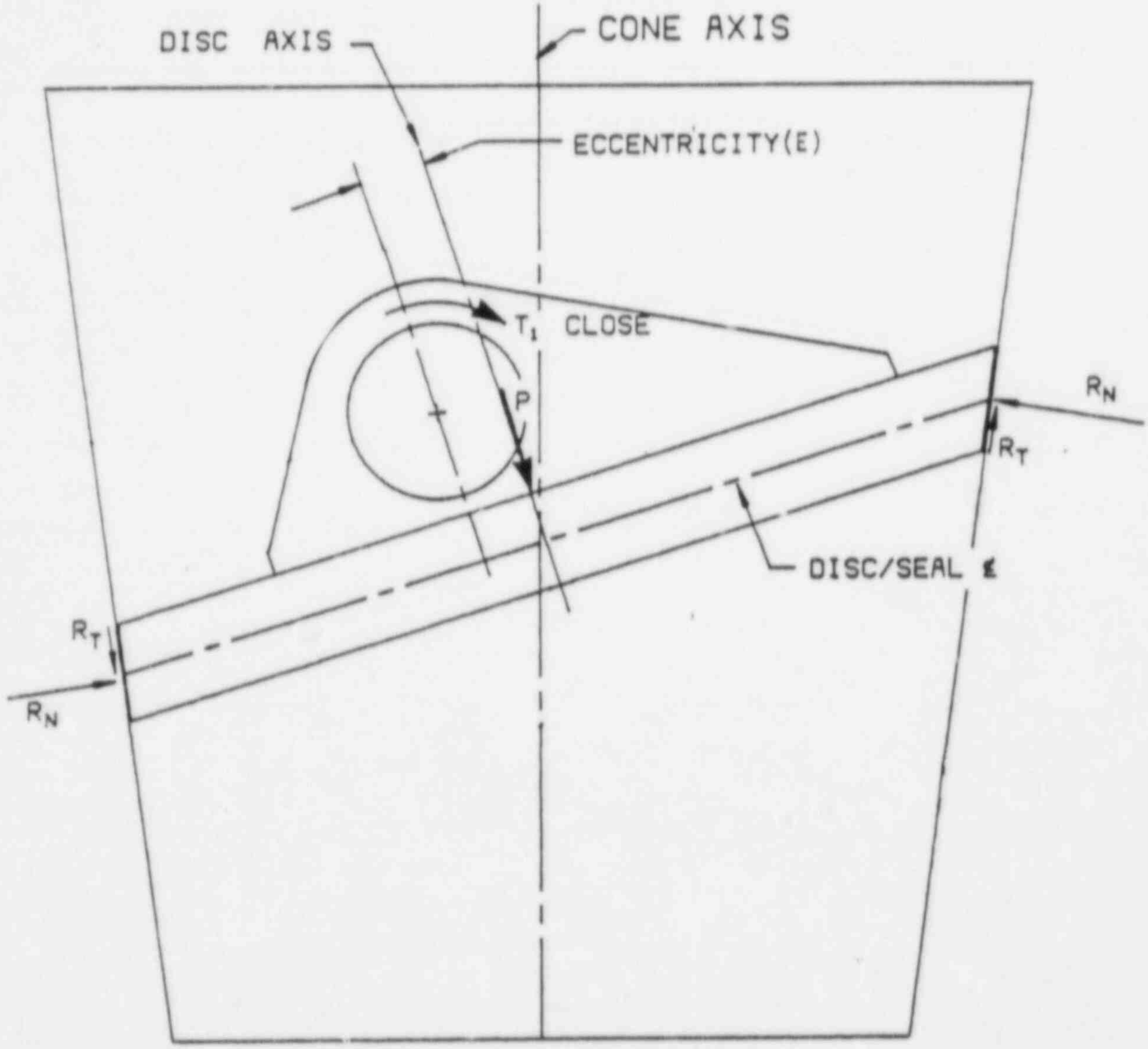
The operation of the Tricentric valve is extremely simple since there are only 2 moving parts, the disc assembly and the shaft. The valve operates by changing the position of the disc relative to the seat. This is accomplished through the application or control of torque on the valve shaft through the 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. 7 ) 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. 7, 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 7

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

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

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

* Center of Gravity

As seen in the Vought Corp. Test Report (Reference 7.0 B3), the running torque was approximately 1000 in.lbs. This is seen in Fig. 8 Run 1 and Fig. 15 Run 8 with no flow through the valve. This running torque is a combination of bearing, packing, and weight offset torque values. The unseating torque may also be seen, which was approximately 1500 in. lbs. when a seating torque of approximately 18,000 in.lbs. was used to close the valve with a 80 psig air supply to the actuator.

## 2.2 Actuator Design

### 2.2.1 Geometry

The basic actuator is a device by which air pressure is converted to thrust through a linear cylinder and then converted to a rotary (90°) motion through the use of a "Scotch-Yoke". This device has a torque output at the beginning and end of its stroke, commonly referred to as breaking torque, that is approximately twice the magnitude of the torque output at the center of its stroke, referred to as running torque. The basic design of the scotch yoke can be seen in Figure 8.

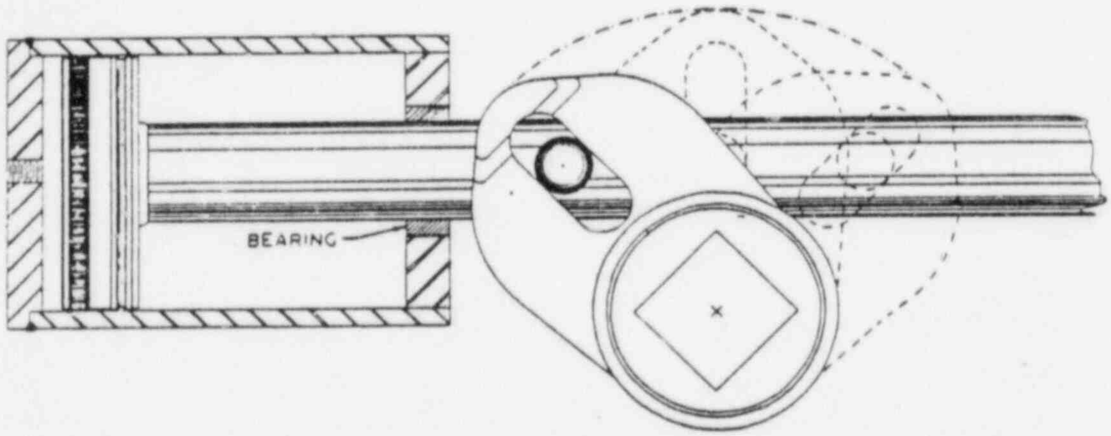



FIGURE 8 - ACTUATOR SCOTCH YOKE DESIGN

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

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

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

TORQUE AT CENTER OF STROKE

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

Where:

T = Torque in in-lb

P = Operating pressure in p.s.i.

MA = Moment arm in inches at center

A = Area of the piston in square inches

TORQUE AT BEGINNING AND END OF STROKE

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

Where:

T = Torque in in-lb

F = Resultant total force in pounds = PXA

$\frac{MA}{\text{Cos. } 45^{o}}$ = 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 9.

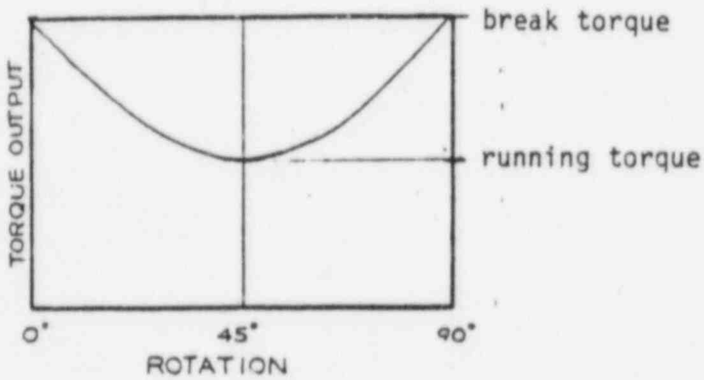



FIGURE 9 - Typical torque output for double acting scotch yoke actuator.

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.

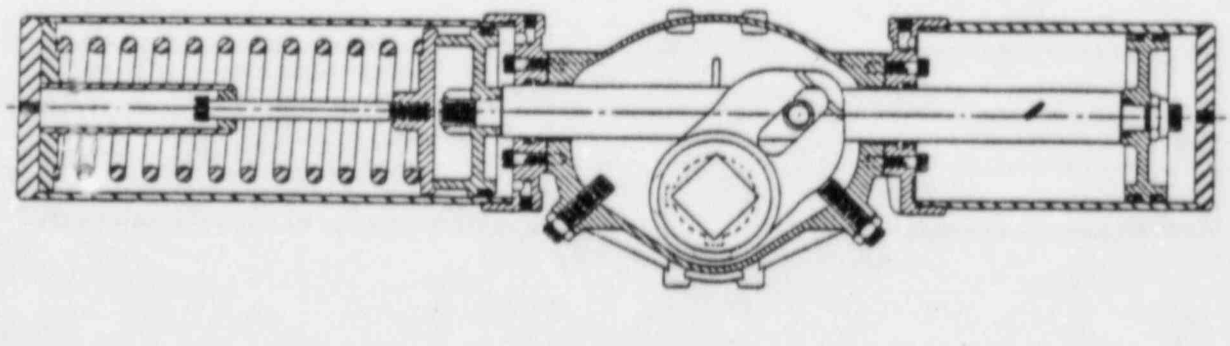


FIGURE 10 - Fail safe, spring return actuator design

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.

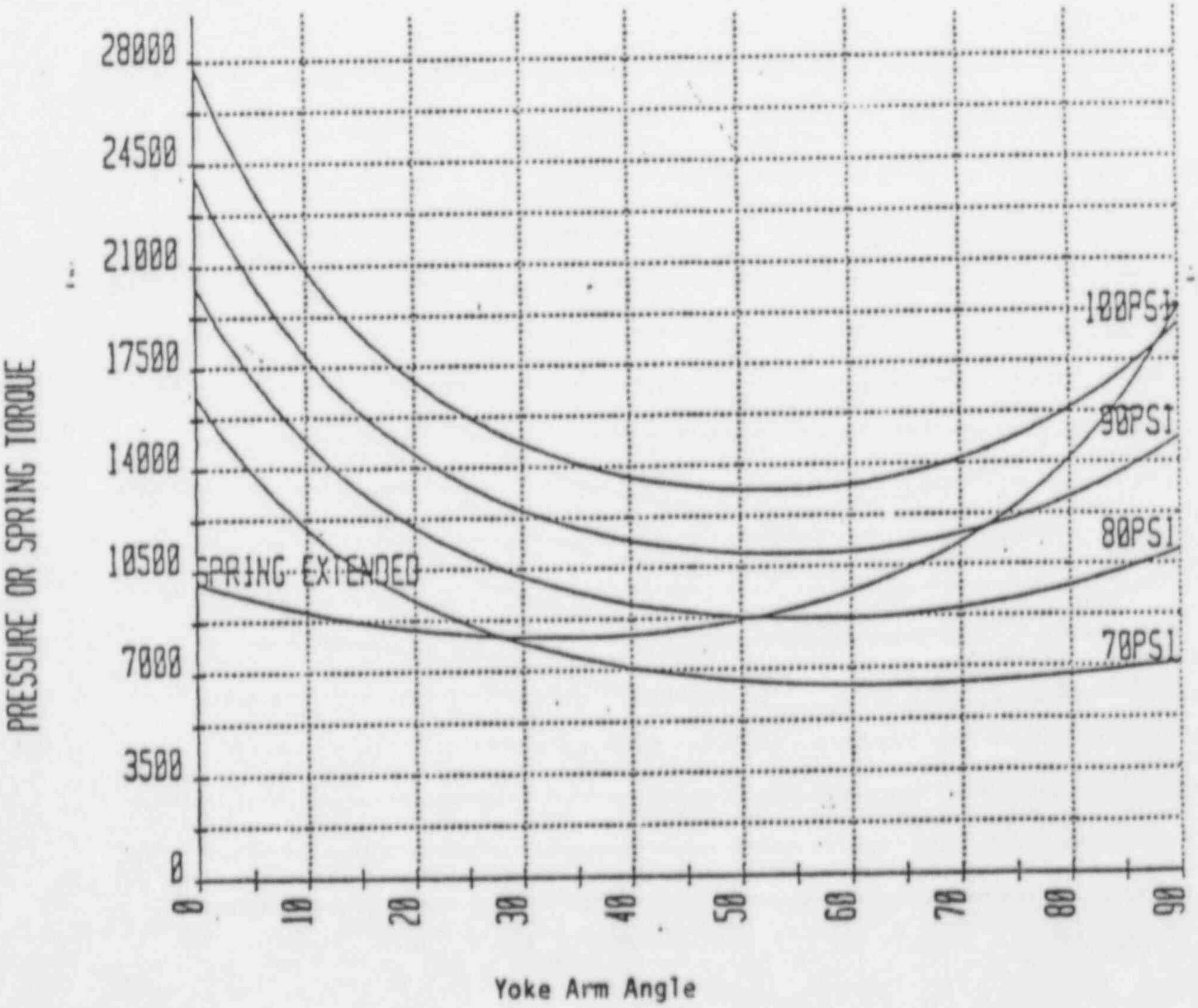



FIGURE 11 - Typical torque output curves for a spring return actuator

2.2.2 Actuator Design Materials

The Bettis actuators used for this job are the HD and 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 $1.4 \times 10^8$ 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

#### 2.2.3.1 Actuators and Accessories Supplied

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

An Asco solenoid valve is used on each actuator to control the air supply to the actuator and, to "dump" the air in the cylinder which allows the valve to open or close as required. The solenoid valves are 3 way, internal piloted diaphragm valves. The solenoid valves are controlled by a 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 ½ 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 NP831 [illegible]47E which is identical to the NP831664E except the port size is 3/4 in. NPT and the orifice is 11/16 in.

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

EA 170-31302 and EA 170-32302 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 010-5337.

Another accessory used on the actuators is a pressure relief valve. The relief valves are only used on the NT 420B-SR2 actuators as seen on the Data Sheet. These were required and supplied by Bettis to prevent over pressurization of the air cylinder. The relief valves are Republic model 637B-3-3/4-2.

TABLE 1

| 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) | Republic Relief Valve Model No. |
|---|---|---|---|---|---|---|---|---|
| 8 | V-23-13<br>V-23-14 | 80-8170-01 | N732C-SR80 | CCW | Close | NP831664E | EA 170-31302 L.S.<br>EA 170-32302 L.S.<br>EL 010-53337 L.A. | N.A. |
| 8 | V-23-15<br>V-23-16 | 80-8170-02 | N732C-SR80 | CCW | Close | NP831664E | EA 170-31302 L.S.<br>EA 170-32302 L.S.<br>EL 010-53337 L.A. | N.A. |
| 12 | V-28-17<br>V-28-18 | 80-8170-03 | NT316B-SR2 | CCW | Close | NP831664E | EA 170-31302 L.S.<br>EA 170-32302 L.S.<br>EL 010-53337 L.A. | N.A. |
| 18 | V-27-1<br>V-27-2<br>V-27-3<br>V-27-4 | 80-8170-04 | NT420B-SR2 | CCW | Close | NP8316A74E | EA 170-31302 L.S.<br>EA 170-32302 L.S.<br>EL 010-53337 L.A. | 637B-3-3/4-2 |
| 20 | V-26-16<br>V-26-18 | 80-8170-05 | NT420B-SR2 | CW | Open | NP8316A74E | EA 170-31302 L.S.<br>EA 170-32302 L.S.<br>EL 010-53337 L.A. | 637B-3-3/4-2 |

2.2.3.2 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 | % |
|---|---|---|
| N732C-SR80 | 9,055/10,060 | .90 |
| NT316B-SR2 | 20,200/21,045 | .96 |
| NT420B-SR2 | 41,300/41,911 | .99 |

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.

FIGURE 12A

# 732 SR80

## DATA INPUT

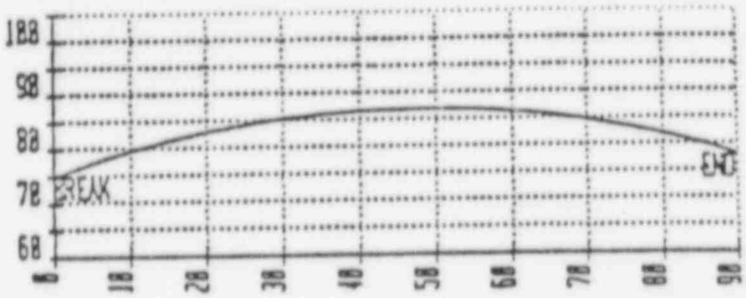
| | | | | |
|---|---|---|---|---|
| CYLINDER DIAMETER (in)= | | | | 7.03 |
| CENTER OR TIE BAR DIAMETER (in)= | | | | 0.000 |
| PISTON ROD DIAMETER (in)= | | | | 2.125 |
| NUMBER OF PISTONS = | | | | 2 |
| MOMENT ARM (in)= | | | | 3.375 |
| SPRING LOAD A (lbs)= | | | | 1737 |
| SPRING LOAD B (lbs)= |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

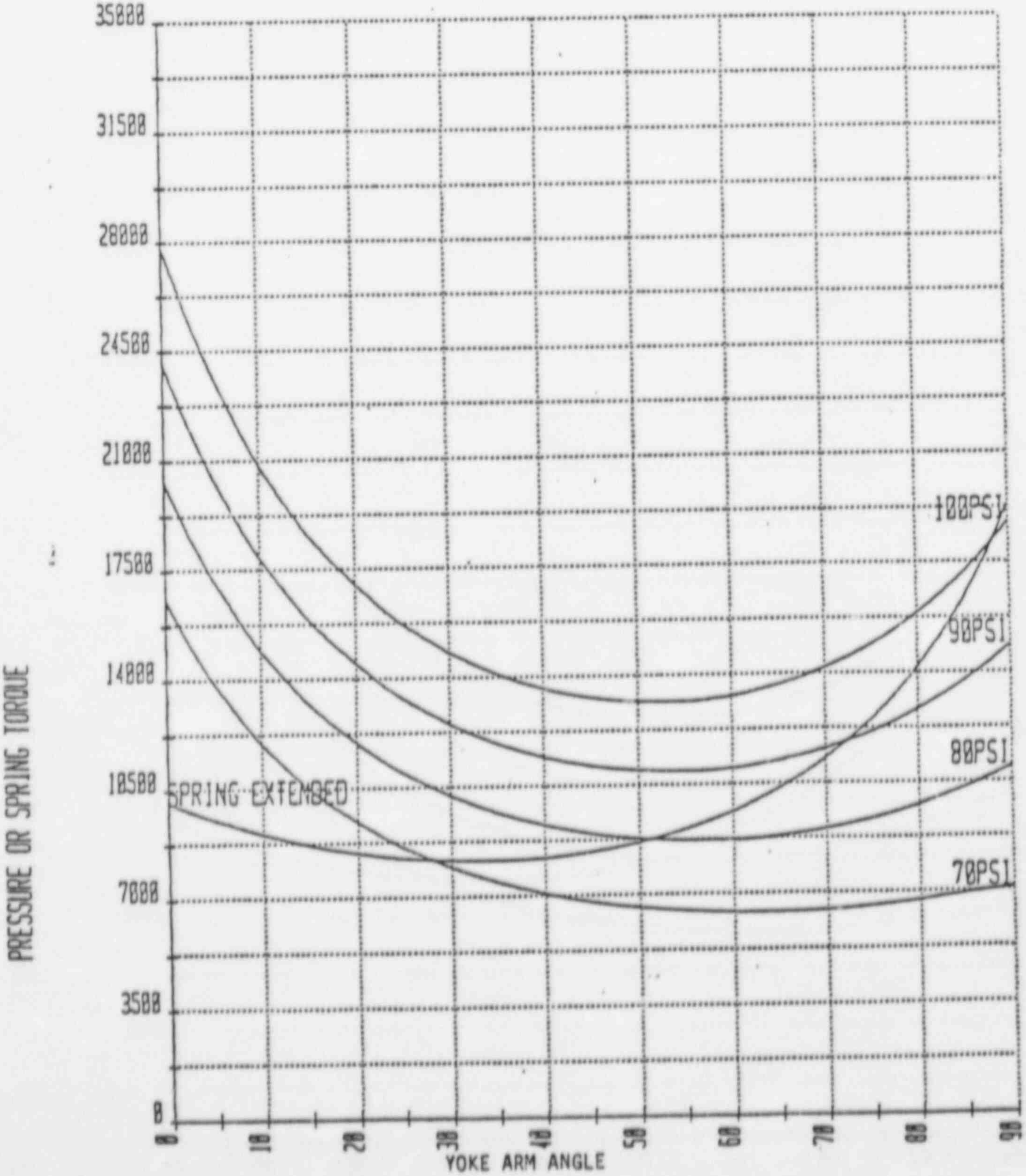
| YOKE ARM ANGLE (degrees) | SPRING TORQUE (in lb) | PRESSURE TORQUE ( 70)psi | PRESSURE TORQUE ( 80)psi | PRESSURE TORQUE ( 90)psi | PRESSURE TORQUE ( 100)psi | EFFICIENCY SPR. % | EFFICIENCY PRES. % |
|---|---|---|---|---|---|---|---|
| 0 | 10060 | 16541 | 20286 | 24031 | 27776 | 78 | 75 |
| 5 | 9511 | 13881 | 17179 | 20477 | 23775 | 80 | 78 |
| 10 | 9051 | 11948 | 14913 | 17878 | 20843 | 82 | 80 |
| 15 | 8687 | 10508 | 13222 | 15937 | 18651 | 83 | 82 |
| 20 | 8418 | 9416 | 11942 | 14468 | 16994 | 85 | 83 |
| 25 | 8240 | 8578 | 10965 | 13351 | 15737 | 86 | 84 |
| 30 | 8149 | 7933 | 10219 | 12504 | 14790 | 86 | 85 |
| 35 | 8146 | 7438 | 9656 | 11874 | 14092 | 87 | 86 |
| 40 | 8232 | 7061 | 9242 | 11423 | 13604 | 87 | 87 |
| 45 | 8412 | 6784 | 8955 | 11126 | 13297 | 87 | 87 |
| 50 | 8697 | 6591 | 8779 | 10967 | 13155 | 87 | 87 |
| 55 | 9101 | 6471 | 8704 | 10938 | 13171 | 86 | 87 |
| 60 | 9648 | 6417 | 8726 | 11036 | 13345 | 85 | 86 |
| 65 | 10368 | 6424 | 8844 | 11264 | 13684 | 84 | 86 |
| 70 | 11311 | 6488 | 9061 | 11633 | 14205 | 83 | 85 |
| 75 | 12544 | 6605 | 9381 | 12157 | 14933 | 82 | 83 |
| 80 | 14172 | 6765 | 9812 | 12860 | 15907 | 80 | 82 |
| 85 | 16355 | 6955 | 10363 | 13771 | 17179 | 78 | 80 |
| 90 | 19350 | 7138 | 11033 | 14928 | 18822 | 75 | 78 |

FIGURE 12B

# EFFICIENCY PLOT

EFFICIENCY vs ANGLE



# T316 SR2

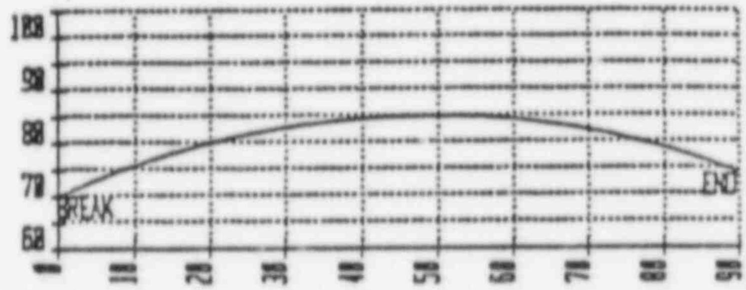
## DATA INPUT

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

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

# EFFICIENCY PLOT

EFFICIENCY vs ANGLE



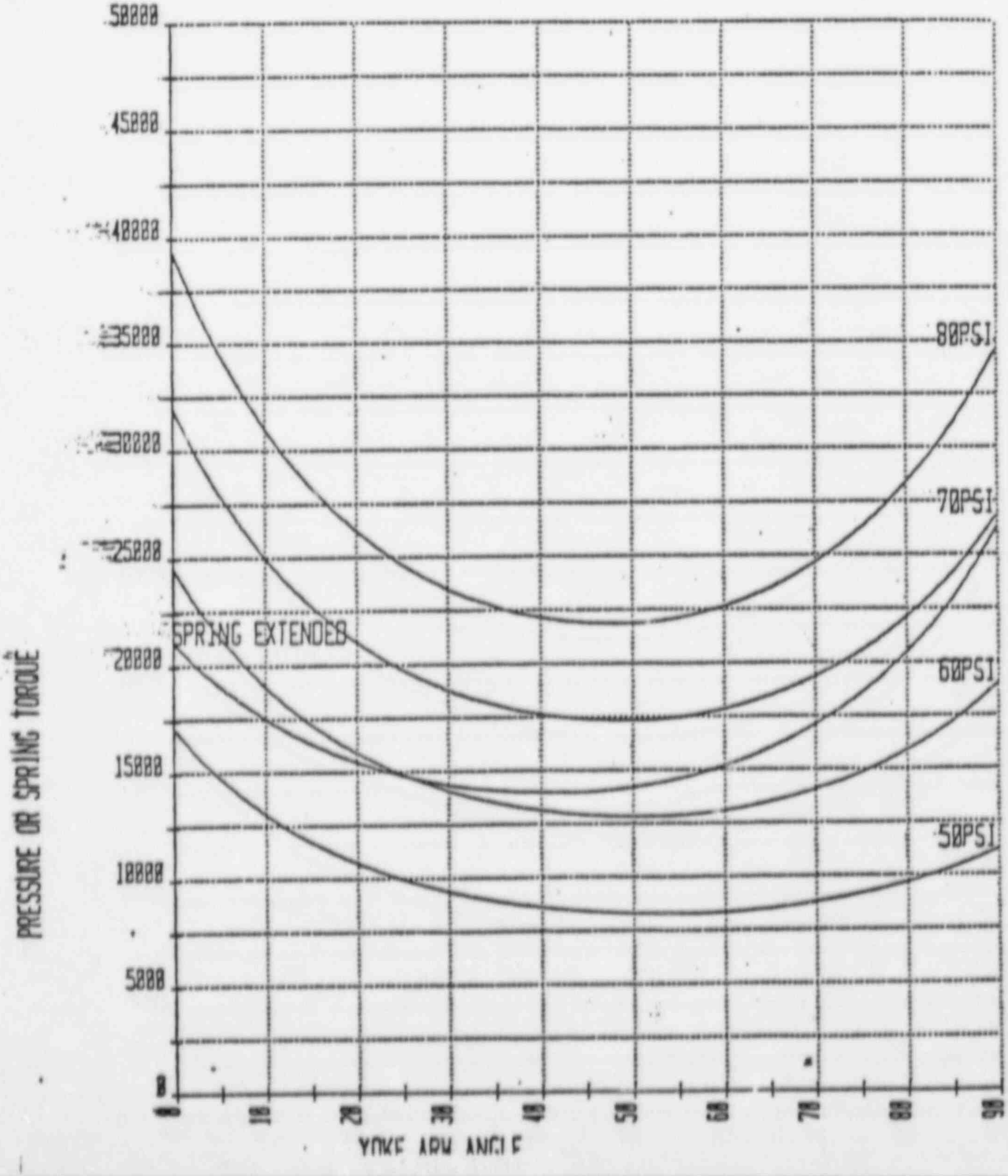



FIGURE 14A

# T420 SR2

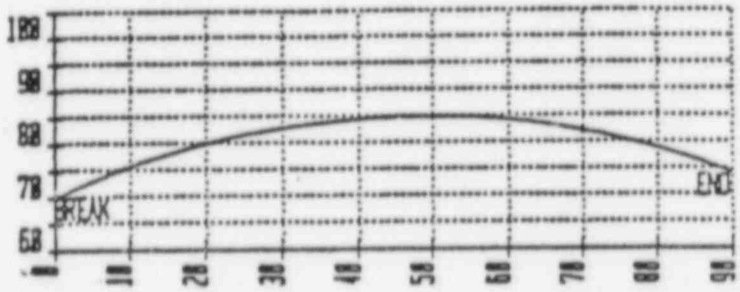
## DATA INPUT

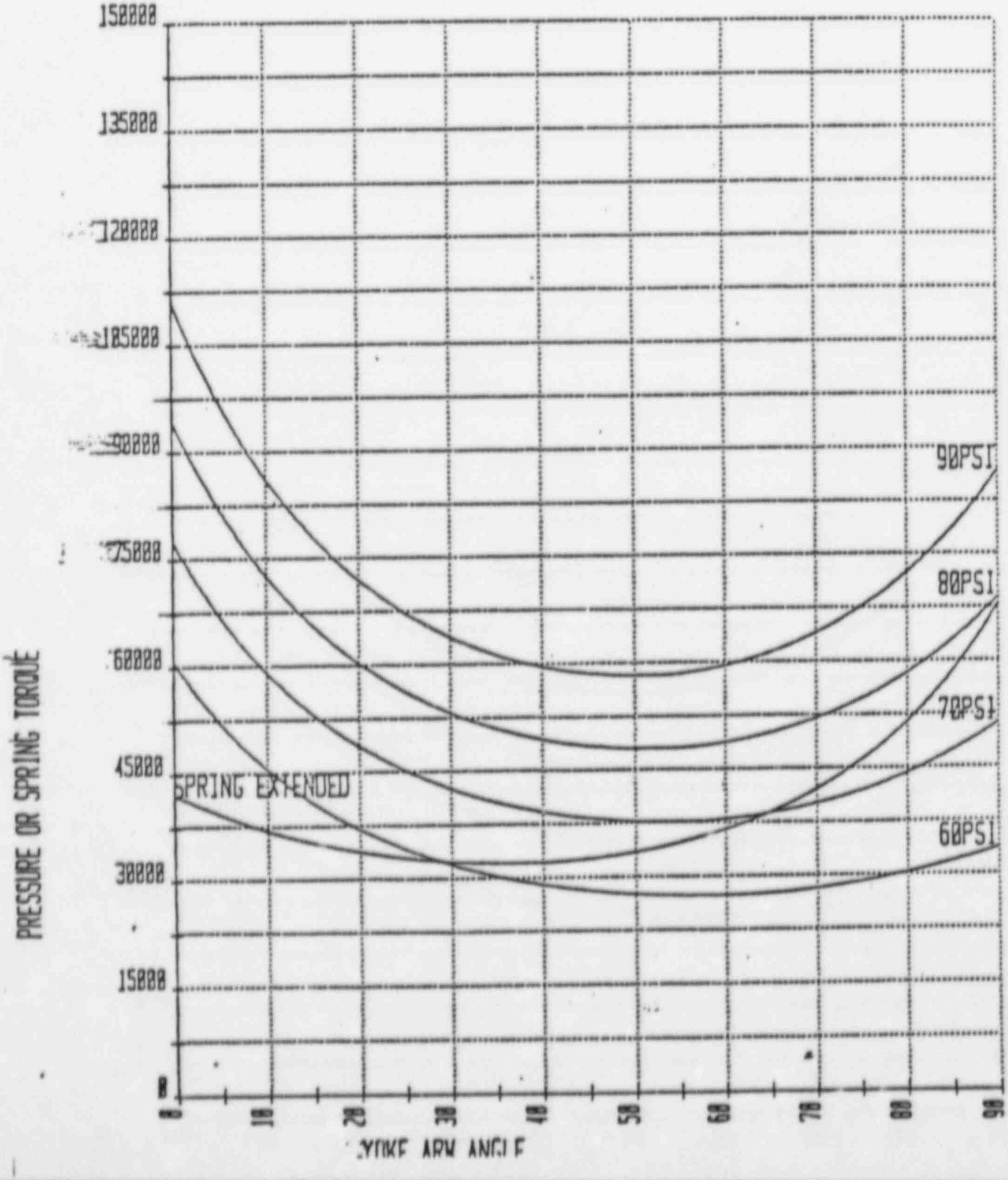
| | | | | |
|---|---|---|---|---|
| CYLINDER DIAMETER (in)= | | | | 19.58 |
| CENTER OR TIE BAR DIAMETER (in)= | | | | 0.875 |
| PISTON ROD DIAMETER (in)= | | | | 1.375 |
| NUMBER OF PISTONS = | | | | 1 |
| MOMENT ARM (in)= | | | | 4.000 |
| SPRING LOAD A (lbs)= | | | | 6436 |
| SPRING LOAD B (lbs)= | | | | 13540 |
| BREAK EFFICIENCY (%)= | | | | 70 |
| RUNNING EFFICIENCY (%) = | | | | 85 |
| ENDING EFFICIENCY (%) = | | | | 74 |
| PRESSURES (psi) = | 60 | 70 | 80 | 90 |
| ACTUATOR TYPE,CB=1,HD=2,T,TR=3, = | | | | 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 | 41911 | 60622 | 77333 | 94044 | 110756 | 74 | 70 |
| 5 | 39116 | 51960 | 66848 | 81737 | 96625 | 77 | 73 |
| 10 | 36861 | 45532 | 59038 | 72544 | 86051 | 79 | 76 |
| 15 | 35098 | 40671 | 53123 | 65576 | 78029 | 81 | 78 |
| 20 | 33775 | 36949 | 48601 | 60254 | 71906 | 82 | 80 |
| 25 | 32850 | 34083 | 45137 | 56191 | 67244 | 83 | 82 |
| 30 | 32294 | 31880 | 42502 | 53125 | 63747 | 84 | 83 |
| 35 | 32088 | 30207 | 40541 | 50876 | 61211 | 85 | 84 |
| 40 | 32230 | 28970 | 39146 | 49322 | 59498 | 85 | 85 |
| 45 | 32728 | 28105 | 38244 | 48383 | 58522 | 85 | 85 |
| 50 | 33610 | 27569 | 37791 | 48012 | 58234 | 85 | 85 |
| 55 | 34919 | 27333 | 37761 | 48189 | 58617 | 84 | 85 |
| 60 | 36724 | 27383 | 38153 | 48922 | 59691 | 83 | 84 |
| 65 | 39123 | 27715 | 38978 | 50241 | 61504 | 82 | 83 |
| 70 | 42262 | 28331 | 40270 | 52209 | 64147 | 80 | 82 |
| 75 | 46349 | 29244 | 42082 | 54919 | 67756 | 78 | 81 |
| 80 | 51690 | 30468 | 44489 | 58511 | 72532 | 76 | 79 |
| 85 | 58748 | 32010 | 47594 | 63178 | 78761 | 73 | 77 |
| 90 | 68242 | 33856 | 51522 | 69189 | 86855 | 70 | 74 |

# 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. |
|---|---|---|---|---|
| V-23-13 | 8 | N732C-SR80 | 2 | 3 |
| V-23-14 | 8 | " | 4 | 3 |
| V-23-15 | 8 | " | 3 | 2 |
| V-23-16 | 8 | " | 4 | 3 |
| V-28-17 | 12 | NT316B-SR2 | 4 | 2.5 |
| V-28-18 | 12 | " | 2.5 | 2.5 |
| V-27-1 | 18 | NT420B-SR2 | 7 | 5 |
| V-27-2 | 18 | " | 7 | 4 |
| V-27-3 | 18 | " | 7 | 5 |
| V-27-4 | 18 | " | 7 | 4 |
| V-26-16 | 20 | NT420B-SR2 | 5.5 | 5 |
| V-26-18 | 20 | " | 5.5 | 5 |

For a description of operating time for valve Serial No. 80-8170-03-01 during the LOCA and Seismic Simulation Test refer to the Vought Corp. Peport (reference 7.0 B3). The Vought Test demonstrated when there was flow through the valve, the aerodynamic torque aided closure thus reducing closing time.

# 3.0 VALVE OPERATING AND INSTALLATION REQUIREMENTS

## 3.1 Valve Operating Conditions

The normal and accident operating conditions for the subject valves are taken from Jersey Central Power & Light Companies Procurement Specification No. 492-7 rev.3, paragraphs 4.1.10 a & b, 6.1, 6.2, 6.3. Leakage requirements are per spec. paragraph 10.5. This data is presented in summarized form in Tables 4, 5, 6, and 7.

For Tables 4 and 5 covering seismic loadings, the following definitions apply:

OBE - Operating Basis Earthquake

SSE - Safety Shutdown Earthquake

EMRV - Electromatic Relief Valve Discharge

CO - Condensation Oscillation

CH - Chugging

PS - Pool Swell

DBA - Design Basis Accident

IBA - Intermediate Break Accident

SBA - Small Break Accident

TABLE 4

Plant Operating and Seismic Loadings
For Valves V-28-17 and V-28-18

| Condition | Damping | Loading Combination (Type Accident) | Acceleration Values (g) Horiz. | Vert. | Peak Stress Cycles |
|---|---|---|---|---|---|
| Upset | 1% | EMRV + OBE | 3.4 | 1.7 | 150 |
| Emergency | 2% | OBE + EMRV + CO (IBA) | 10.6 | 10.7 | 50 |
| " | " | EMRV + CO (IBA) | 10.0 | 10.6 | 25,200 |
| " | " | OBE + EMRV + CH (IBA, SBA) | 4.0 | 3.0 | 50 |
| " | " | EMRV + CH (IBA,SBA) | 2.5 | 2.6 | 4,500 |
| Faulted | 2% | SSE + EMRV + CO | 11.0 | 10.7 | 50 |
| " | " | EMRV + CO | 10.0 | 10.6 | 790 |
| " | " | EMRV + PS | 4.2 | 4.5 | 140 |
| " | " | EMRV + CH | 2.4 | 2.6 | 310 |

NOTE: Natural frequency for valve assembly to be greater that 40 Hz.
g = Acceleration as a fraction of the acceleration due to gravity.

TABLE 5

Seismic Loadings For All Valves
Except V-28-17 and V-28- 18

| Condition | Damping | Loading Condition | Acceleration Values (g) Horiz. | Vert. | Peak Stress Cycles |
|---|---|---|---|---|---|
| Upset | 1% | OBE | 2.0 | 2.0 | 150 |
| Faulted | 2% | SSE | 3.0 | 3.0 | 30 |

NOTE: Natural frequency for valve assembly to be greater than 33 Hz.
g = Acceleration as a fraction of the acceleration due to gravity.

TABLE 6

Pressure Differentials Applied to Valves

| VALVE SIZE | VALVE MARK NO. | NORMAL OPERATING PRESSURE (PSIG) | OPER. TEMP. RANGE (°F) | MAXIMUM DIFFERENTIAL PRESSURE (PSIG) | MAXIMUM NORMAL FLOW SCFM | FAILURE MODE |
|---|---|---|---|---|---|---|
| 8 | V-23-15 & 16 | 1 | 40-340 | 35 | 1200 | closed |
| 8 | V-23-13 & 14 | 25 | 40-340 | 62 | 1200 | closed |
| 12 | V-28-17 & 18 | 1 | 40-150 | 35 | 1200 | closed |
| 18 | V-27-1,2,3,&4 | 1 | 40-340 | 62 | 6200 | closed |
| 20 | V-26-16 & 18 | 1 | 40-150 | 35 | 30,200 | open |

TABLE 7

Allowed Seat Leakage Rates

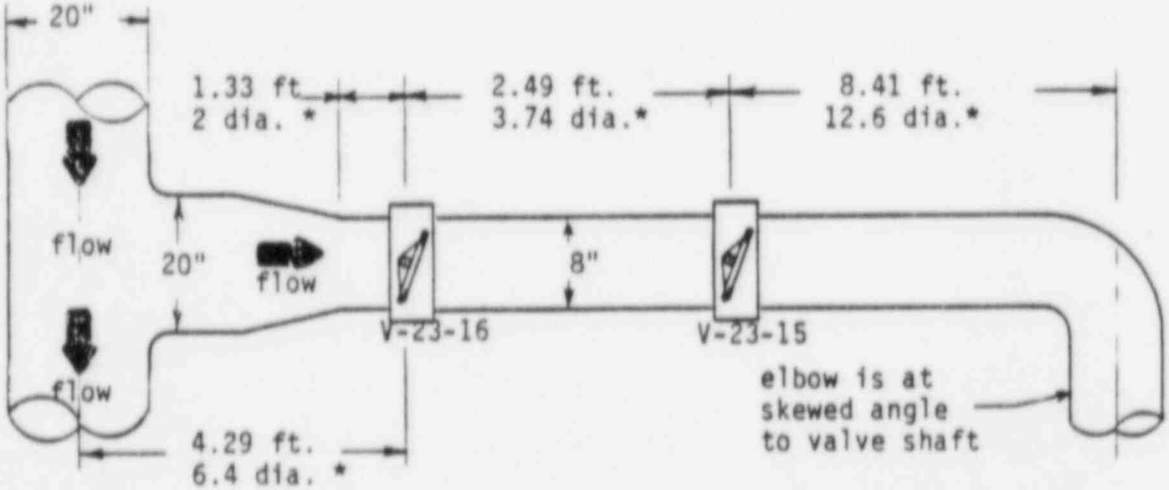
(Per Spec at 35 psig Pneumatic)

| VALVE SIZE | VALVE MARK NO. | ALLOWED LEAKAGE (SCFM) |
|---|---|---|
| 8 | V-23-15 & 16 | 0.013 |
| 8 | V-23-13 & 14 | 0.013 |
| 12 | V-28-17 & 18 | 0.020 |
| 18 | V-27-1,2,3,&4 | 0.030 |
| 20 | V-26-16 & 18 | 0.033 |

## 3.2 Valve Installation Configurations

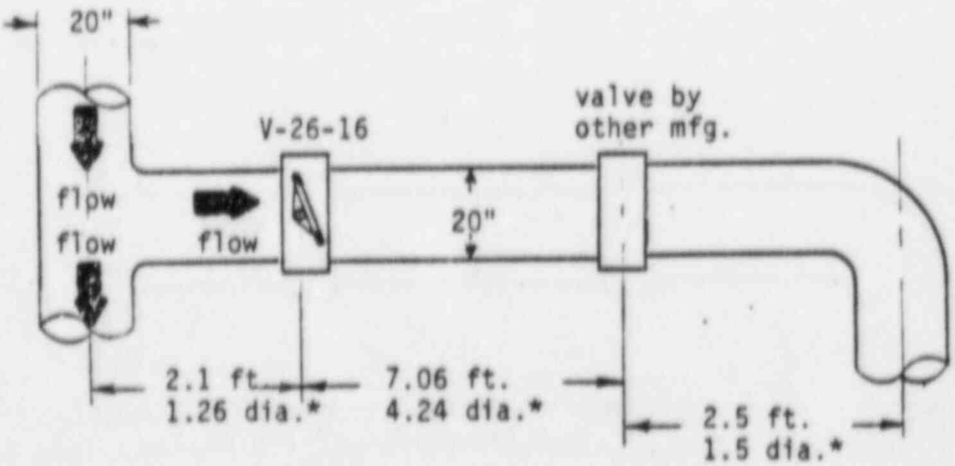
In addition to the pressure and flow conditions specified in 3.0, the valve performance is effected by the as installed orientation. Upstream and downstream, tees, elbows, reducers, and other valves can effect the aerodynamic torque characteristics of butterfly valves. These effects are discussed in Section 5.0. The installed configurations for the subject valves as derived from Stone & Webster prints 13432.19-02 thru -06 and 13432.19 EM-2 page 1 thru 4, are summarized in Figures 15 thru 21.

NOTE: All valve discs in the figures are shown in the partially open (approximately $20^{o}$ off of seat) position.



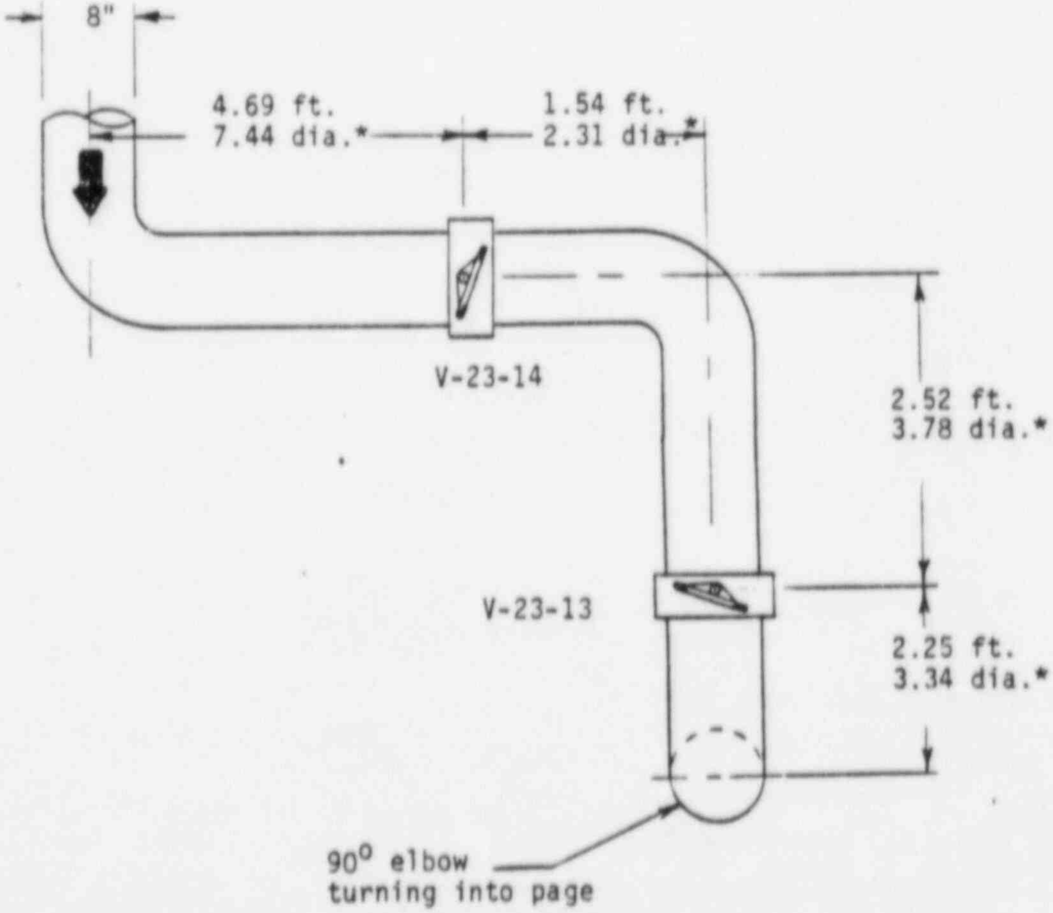

* Expressed in nominal valve diameters (8" = 1.0 dia.)

FIGURE 15 Installed orientation of 8" valves V-23-15 & 16
(per Stone & Webster drwgs. 13432.19-03 & 13432.19-EM-2,
page 2 of 4, rev. 0)



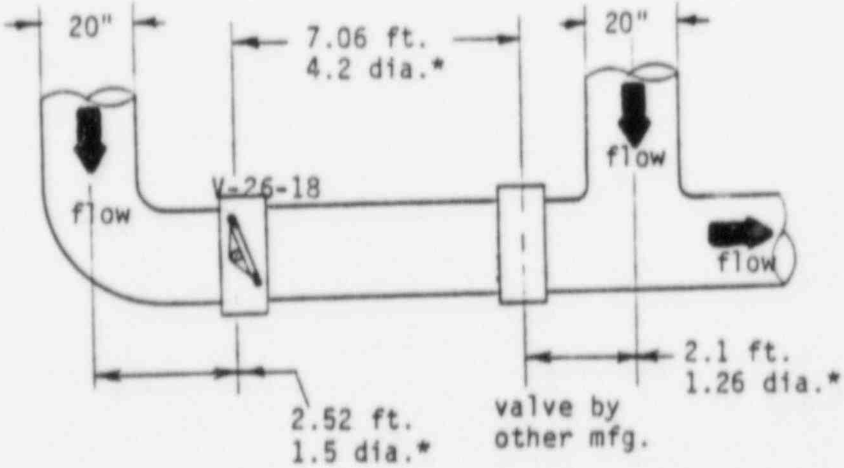

* Expressed in nominal valve diameters (20" = 1.0 dia.)

FIGURE 16 Installed orientation of 20" valve V-26-16
(per S & W drwgs. 13432.19-03 and 13432.19-EM-2,
page 2 of 4, rev.0 with orientations as specified
by Dave Miller of G.P.U.)



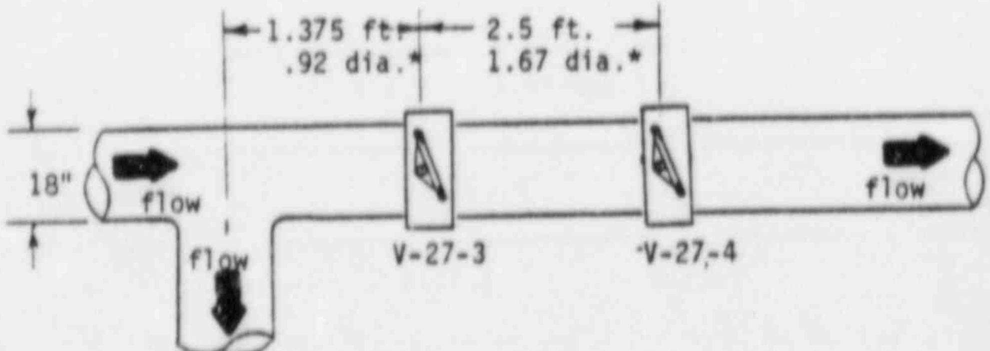

* Expressed in nominal valve diameters

FIGURE 17 Installed orientation of 8" valves V-23-13 & 14 (per S & W drwgs. 13432.19-02 & 13432.19-EM-2, page 1 of 4, rev. 0)



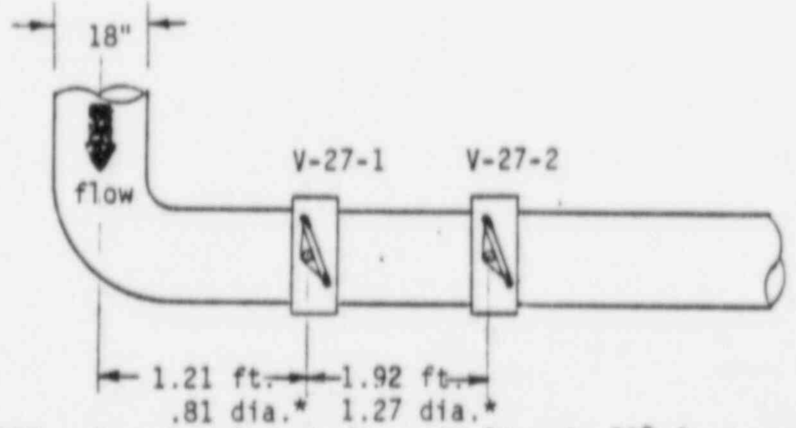

* Expressed in nominal valve diameters (20" = 1.0 dia.)

FIGURE 18 Installed orientation of 20" valve V-26-18 (per S & W drwgs. 13432.19-03 & 13432.19-EM-2, page 2 of 4, rev. 0 with orientations as specified by Dave Miller of G.P.U.)




* Expressed in nominal valve diameters (18" = 1.0 dia.)

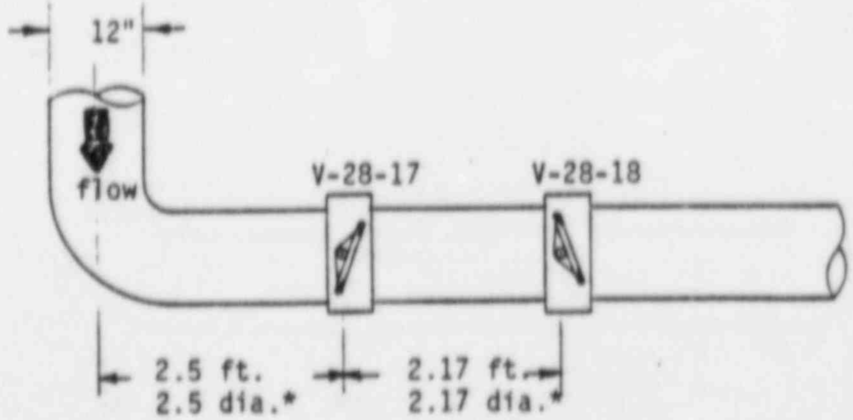
FIGURE 19 Installed orientation of 18" valves V-27-3 and 4 (per S & W dwgs. 13432.19-02 & 13432.19-EM-2, page 1 of 4, rev. 0)




NOTE: Actual valve shaft centerline is $30^{\circ}$ from vertical with vertical being $90^{\circ}$ to the plane of the page

* Expressed in valve diameters (18" = 1.0 dia.)

FIGURE 20 Installed orientation for 18" valves V-27-1 & 2 (per S & W drwgs. 13432.19-05 & 13432.19-EM-2, page 1 of 4, rev. 0)




* Expressed in vlave diameters (12" = 1.0 dia.)

FIGURE 21 Installed orientation for 12" valves V-28-17 & 18 (per S & W drwgs. 13432.19-06 & 13432.19-EM-2, page 4 of 4, rev. 0)

## 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 Jersey Central Power and Light Specification 492-7. 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 Wyle Laboratories, 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 9 (these exceed those required by Spec. 492.7). The obtained lowest resonant frequencies for the valve assembly are presented in Table 8. All meet spec. requirements. For stress analysis, allowable stresses were in accord with ASME Section III requirements and Table 10 of allowed stress values. Table 11 thru 19

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

## 4.2 CLOW TRICENTRIC VALVE ASSEMBLY RESONANT FREQUENCY TEST

A Low-Level Seismic Vibration Test was performed on a Clow 12" valve/actuator assembly to determine resonant frequencies less than 40 Hz. The test was performed at Wyle Labs, Huntsville, Alabama. The test program consisted of biaxial sine sweep testing in each of two testing orientations. The assembly was instrumented with accelerometers to measure input and response accelerations. The test demonstrated that the unit possessed no major structural resonances within the frequency range of 1 to 40 Hz and thus verified the seismic frequency analysis for this range.

## 4.3 ASCO SOLENOID VALVE RESONANT FREQUENCY TEST

A valve actuator solenoid valve, Asco model 831664, was subjected to both a sine sweep test and sine beat test in each of three orthogonal test orientations. 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.

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.

## 4.4 STATIC LOAD TEST DURING SIMULATED LOCA FLOW

As part of the operability test performed at Vought (see reference 7.0 B-3) an 11.0 g load was applied in each of two orthagonal directions through the approximate center of gravity of the actuator. With the load applied and flow through the valve greater than expected in service, the valve operated within the required time period. This aspect of the test demonstrated that the actuator to valve connection was sufficiently rigid to remain fully operable under this load. Further details are included in the subject report.

TABLE 8

Lowest Valve Resonant Frequencies
(Per Analysis)

| VALVE SIZE | FREQUENCY (Hz) |
|---|---|
| 8 | 122 |
| 12 | 108 |
| 18 | 124 |
| 20 | 150 |

TABLE 9

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) |
|---|---|---|---|---|---|
| 8 | V-23-13,14,15,16 | 275 | 65 | 350 | 8,324 |
| 12 | V-28-17 & 18 | 275 | 65 | 350 | 20,600 |
| 18 | V-27-1,2,3 & 4 | 275 | 65 | 350 | 41,000 |
| 20 | V-26-16 & 18 | 275 | 65 | 350 | 68,000 |

TABLE 10

Allowed Stress

| Applicable Valves | Condition | Membrane Stress Limit | Membrane + Bending Stress Limit |
|---|---|---|---|
| V-28-17 & 18 (12" Valves) | Upset | 1.1 S | 1.65 S |
| | Emergency | 1.5 S | 1.8 S |
| | Faulted | 2.0 S | 2.40 S |
| All Others | Upset | 1.1 S | 1.65 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).

Table 11
Maximum Stress Ratio
Upset Condition
8" Valve

| Location | Upset | | Maximum Stress Ratio | |
|---|---|---|---|---|
| | Node or Element | Max Stress | Allowable Stress | Stress Ratio |
| Valve Body | 292 | 772 | 19250 | 0.04 |
| Disc | 2 | 1990 | 19250 | 0.10 |
| Drive Shaft | 201 | 13301 | 57008 | 0.23 |
| Operator Adapter Plate | 322 | 1710 | 19250 | 0.09 |
| Operator/Adapter Bolts | N/A | 7634 | 27500 | 0.28 |
| Body/Adapter Bolts | N/A | 7634 | 27500 | 0.28 |

Table 12
Maximum Stress Ratio
Faulted Condition
8" Valve

| Location | Faulted | | Maximum Stress Ratio | |
|---|---|---|---|---|
| | Node or Element | Max Stress | Allowable Stress | Stress Ratio |
| Valve Body | 292 | 1007 | 35000 | 0.03 |
| Disc | 2 | 1995 | 35000 | 0.06 |
| Drive Shaft | 201 | 13217 | 82920 | 0.16 |
| Operator Adapter Plate | 321 | 2175 | 35000 | 0.06 |
| Operator/Adapter Bolts | N/A | 12370 | 50000 | 0.25 |
| Body/Adapter Bolts | N/A | 12370 | 50000 | 0.25 |
| Cover Plate | N/A | 5663 | "S" = 17500 | 0.32 |
| Cover Plate Bolts | N/A | 6054 | "S" = 25000 | 0.24 |

Table 13
Maximum Stress Ratio
Upset Condition
12" Valve

| UPSET | | MAXIMUM STRESS RATIO | | |
|---|---|---|---|---|
| Location | Node or Element | Max Stress | Allowable Stress | Stress Ratio |
| Valve Body | 295 | 1454 | 19250 | 0.08 |
| Disc | 7 | 2343 | 19250 | 0.12 |
| Drive Shaft | 223 | 14032 | 57008 | 0.25 |
| Body Adapter Plate | 295 | 1128 | 19250 | 0.06 |
| Operator Adapter Plate | 264 | 867 | 19250 | 0.05 |
| Adapter Plate Bolts | Beam 317 | 10056 | 27500 | 0.37 |
| Operator/Adapter Bolts | N/A | 13380 | 27500 | 0.49 |
| Body/Adapter Bolts | N/A | 13380 | 27500 | 0.49 |

Table 14
Maximum Stress Ratio
Emergency Condition
12" Valve

| EMERGENCY | | | MAXIMUM STRESS RATIO | |
|---|---|---|---|---|
| Location | Node or Element | Max Stress | Allowable Stress | Stress Ratio |
| Valve Body | 295 | 4187 | 26250 | 0.16 |
| Disc | 7 | 2382 | 26250 | 0.09 |
| Drive Shaft | 222 | 15591 | 62190 | 0.25 |
| Body Adapter Plate | 295 | 3469 | 26250 | 0.13 |
| Operator Adapter Plate | 264 | 2622 | 26250 | 0.10 |
| Adapter Plate Bolts | Beam 317 | 30953 | 37500 | 0.83 |
| Operator/Adapter Bolts | N/A | 46642 | ASME, Section III, Appendix XVII, 2460 | 0.72 |
| Body/Adapter Bolts | N/A | 46642 | ASME, Section III, Appendix XVII, 2460 | 0.72 |

Table 15
Maximum Stress Ratio
Faulted Condition
12" Valve

| FAULTED | | | MAXIMUM STRESS RATIO | |
|---|---|---|---|---|
| Location | Node or Element | Max Stress | Allowable Stress | Stress Ratio |
| Valve Body | 295 | 4271 | 35000 | 0.12 |
| Disc | 7 | 2385 | 35000 | 0.07 |
| Drive Shaft | 222 | 15637 | 82920 | 0.19 |
| Body Adapter Plate | 295 | 3516 | 35000 | 0.10 |
| Operator Adapter Plate | 264 | 2658 | 35000 | 0.08 |
| Adapter Plate Bolts | Beam | 31395 | 50000 | 0.63 |
| Operator/Adapter Bolts | | 47650 | 50000 | 0.95 |
| Body/Adapter Bolts | | 47650 | 50000 | 0.95 |
| Cover Plate | | 8836 | "S" = 17500 | 0.50 |
| Cover Plate Bolts | | 6970 | "S" = 25000 | 0.28 |

A

Table 16
Maximum Stress Ratio
Upset Condition

18" Valve

| Upset Maximum Stress Ratio | | | | |
|---|---|---|---|---|
| Location | Node or Element | Max Stress | Allowable Stress | Stress Ratio |
| Valve Body | 32 | 1356 | 19250 | 0.07 |
| Disc | 6 | 4519 | 19250 | 0.23 |
| Drive Shaft | 222 | 14403 | 57008 | 0.25 |
| Body Adapter Plate | 295 | 801 | 19250 | 0.04 |
| Operator Adapter Plate | 459 | 940 | 19250 | 0.05 |
| Adapter Plate Bolts | Beam 314 | 10818 | 27500 | 0.39 |
| Operator/Adapter Bolts | N/A | 5952 | 27500 | 0.22 |
| Body/Adapter Bolts | N/A | 5952 | 27500 | 0.22 |

Table 17
Maximum Stress Ratio
Faulted Condition
18" Valve

| Faulted Maximum Stress Ratio | | | | |
|---|---|---|---|---|
| Location | Node or Element | Max Stress | Allowable Stress | Stress Ratio |
| Valve Body | 32 | 1429 | 35000 | 0.04 |
| Disc | 6 | 4527 | 35000 | 0.13 |
| Drive Shaft | 222 | 14546 | 82920 | 0.18 |
| Body Adapter Plate | 295 | 991 | 35000 | 0.03 |
| Operator Adapter Plate | 449 | 1187 | 35000 | 0.03 |
| Adapter Plate Bolts | Beam 314 | 13513 | 50000 | 0.27 |
| Operator/Adapter Bolts | N/A | 8663 | 50000 | 0.18 |
| Body/Adapter Bolts | N/A | 8663 | 50000 | 0.18 |
| Cover Plate | N/A | 10862 | "S" = 17500 | 0.62 |
| Cover Plate Bolts | N/A | 8713 | "S" = 25000 | 0.35 |

Table 18
Maximum Stress Ratio
Upset Condition

20" Valve

| Location | Upset | | Maximum Stress Ratio | |
|---|---|---|---|---|
| | Node or Element | Max Stress | Allowable Stress | Stress Ratio |
| Valve Body | 32 | 1304 | 19250 | 0.07 |
| Disc | 6 | 4629 | 19250 | 0.24 |
| Drive Shaft | 226 | 18501 | 57008 | 0.32 |
| Body Adapter Plate | 295 | 1027 | 19250 | 0.05 |
| Operator Adapter Plate | 464 | 902 | 19250 | 0.05 |
| Adapter Plate Bolts | Beam 317 | 10972 | 27500 | 0.40 |
| Operator/Adapter Bolts | N/A | 6495 | 27500 | 0.24 |
| Body/Adapter Bolts | N/A | 6495 | 27500 | 0.24 |

Table 19
Maximum Stress Ratio
Faulted Condition
20" Valve

| Location | Faulted Maximum Stress Ratio | | | |
|---|---|---|---|---|
| | Node or Element | Max Stress | Allowable Stress | Stress Ratio |
| Valve Body | 32 | 1346 | 35000 | 0.04 |
| Disc | 6 | 4644 | 35000 | 0.13 |
| Drive Shaft | 226 | 18884 | 82920 | 0.23 |
| Body Adapter Plate | 295 | 1232 | 35000 | 0.04 |
| Operator Adapter Plate | 464 | 1087 | 35000 | 0.03 |
| Adapter Plate Bolts | Beam 317 | 13209 | 50000 | 0.26 |
| Operator/Adapter Bolts | N/A | 8880 | 50000 | 0.18 |
| Body/Adapter Bolts | N/A | 8880 | 50000 | 0.18 |
| Cover Plate | N/A | 9322 | "S" = 17500 | 0.53 |
| Cover Plate Bolts | N/A | 6705 | "S" = 25000 | 0.27 |

4.5 Fatigue Analysis

JC P & L Specification 492-7, Rev. 3, Section 6.2.2, item 5, states "The number of cycles of maximum stress due to upset, emergency, and faulted loading combinations occurring from earthquake, EMRV, CO and CH vibrations shall be considered for fatigue analysis when fatigue effects are a design consideration". For all valves except V-28-17 and V-28-18, the number of cycles of maximum stress is 150. For these valves, fatigue effects are not a design consideration. For valves V-28-17 and V-28-18, the emergency condition would be a worst case in regard to fatigue with 25,200 peak stress cycles at 10.0 g horizontal and 10.6 vertical for the loading combination of EMRV plus CO.

For all components, except the body to adaptor bolting under the emergency condition, stress levels are sufficiently low so that demonstration of adequacy of body to adaptor bolts would provide assurance that all other components are acceptable. The evaluation of bolt stresses for the emergency condition is given in Wyle Analysis Report WR 81-54. The bolt stresses given are for the worst case loading (not worst fatigue case of 25,200 cycles) thus, these are conservative in regard to a fatigue analysis.

In accord with ASME, Section III, Appendix XIV, paragraph 1322 (High Strength Bolts), design fatigue analysis is to be accomplished by use of Figure I-9.4. The upper curve (max. nominal stress 2.7 $S_m$) may be used since:

1. Maximum tensile stress (for higher than required loading) is 46,642 psi which is less than 2.7 Sm (Sm = 33.8 ksi @ 150°F per Table I-1.3 of Appendix I, ASME Section III) neglecting stress concentrations.
2. Direct tensile stress is 43,404 psi which is less than 2.0 Sm.

In accord with Table I-9.1 allowed cycles may be interpolated between given values in accord with the formula

$$N/N_i = (N_j/N_i)^{(\log S_i/S)/(\log S_i/S_j)}$$

where:

$N_j$ = no. of cycles above required cycles (from table)

$N_i$ = no. of cycles below required cycles (from table)

$S_j$ = peak stress above fatigue analysis design stress

$S_i$ = peak stress below fatigue analysis design stress

$S$ = actual fatigue analysis stress

$N$ = allowed no. of cycles

Since torquing of bolts (as specified in maintenance and operation manual) provides a prestress at least equal to the external load stress, the bolt peak tensile stress will vary as a maximum from zero to a peak value of 46,642 (a conservative assumption since bolt will always maintain some prestress to prevent joint separation). Thus in accord with XIV-1221.3 $S = .5\ (46{,}642-0) = 23{,}321$ psi

Thus, $N/20{,}000\ (50{,}000/20{,}000)^{(\log 27/23.3)/(\log 27/22)}$

$\therefore N = 38{,}673$

NOTE: See Appendix C for copies of Table I-9.1 and Figure I-9.4 from ASME Section III.

Since $N > 25{,}200$ bolts are acceptable for fatigue conditions specified in JC P & L Spec 492-7.

## 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 20 ) 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 $80^{\circ}$ 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 $90^{\circ}$ (¼ turn) valve design. Therefore, at small opening angles ($0^{\circ}$ to $20^{\circ}$) 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

$\Delta P$ = pressure differential across the valve (lb/in$^2$)

$D_V$ = nominal valve diameter (in.)

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

Table 20 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 21 shows a comparison between the provided size valves and the interpolated sizes.

Linear interpolation was used to predict torque characteristics in Clow Program CVAP, thus a similar interpolation of sizes is applicable for size comparison purposes. It can be seen in the table that very good (less than 9% deviation) correlation was obtained for torque critical items. Thus torque data from the program is valid for this application.

TABLE 20

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 offset from body centerline

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

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

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

All dimensions in inches

TABLE 21

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

| | VALVE SIZES | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| ELEMENTS | 8" | | 12" | | 18" | | 20" | |
| | Size | Ratio | Size | Ratio | Size | Ratio | Size | Ratio |
| *I.D. | 7.98 | 1.05 | 11.94 | 1.00 | 16.88 | 1.02 | 18.81 | 1.01 |
| *$A_2$ | 7.24 | 1.06 | 11.15 | 1.00 | 16.08 | 1.02 | 17.96 | 1.01 |
| *$K_2$ | 7.07 | 1.05 | 10.88 | 1.99 | 15.70 | 1.01 | 17.53 | 1.00 |
| Shaft Dia. | 1.5 | 1.28 | 2.0 | 1.13 | 2.5 | 1.10 | 2.75 | 1.06 |
| Shaft ℄ to Seal ℄, L | 1.5 | 1.18 | 1.88 | 1.06 | 2.19 | 1.07 | 2.38 | 1.03 |
| *Disc Thickness | 1.25 | 1.09 | 1.50 | 1.00 | 1.63 | 1.04 | 1.75 | 1.00 |
| *Shaft Offset E | 1.38 | 1.01 | 1.31 | .95 | .95 | 1.08 | 1.00 | .96 |
| Shaft Offset LC | 1.41 | NA | 1.36 | NA | 1.0 | NA | 1.07 | NA |
| Ear Width | 2.0 | .96 | 2.0 | 1.13 | 2.5 | 1.10 | 2.75 | 1.06 |
| Ear Height | 2.25 | .78 | 2.75 | 1.23 | 3.25 | 1.27 | 3.50 | 1.25 |

*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 22 ).[1] Further, the mitered elbow produces flow patterns more severe than expected for tee flow (see Figures 22 and 25 ). 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 26) and aid closure for geometry 2.

Based on these considerations, models of a 12", 24", and 48" valve (per Table 20) 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.

FIGURE 22 - Water Table Study of Choked Flow Pattern With Disc Full Open ($90^0$)

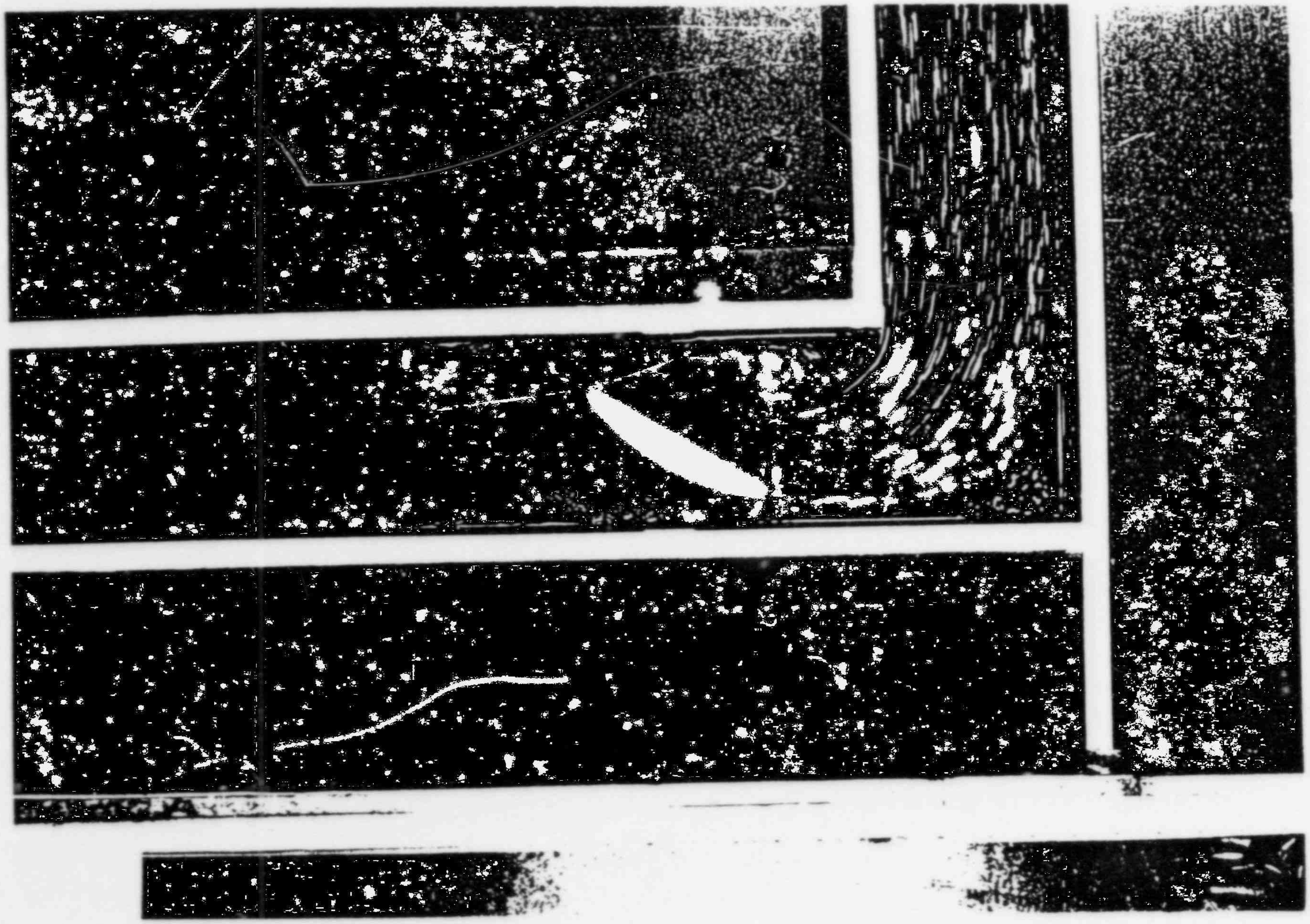


FIGURE 23 - Water Table Study of Choked Flow Pattern With Disc Partially Open ($60^{0}$)

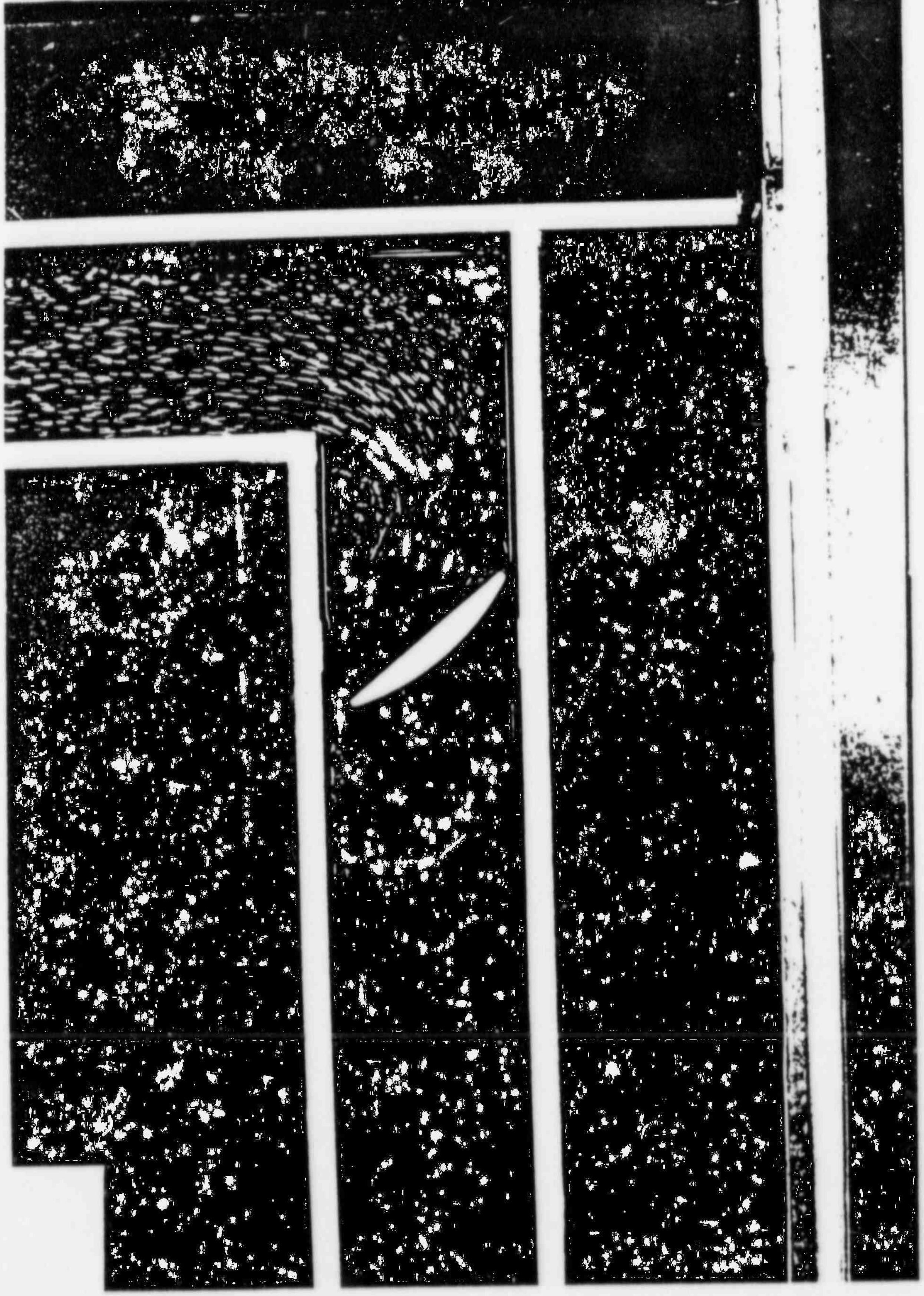


FIGURE 24 - Water Table Study of Choked Flow Pattern With Disc Partially Open ($40^0$)

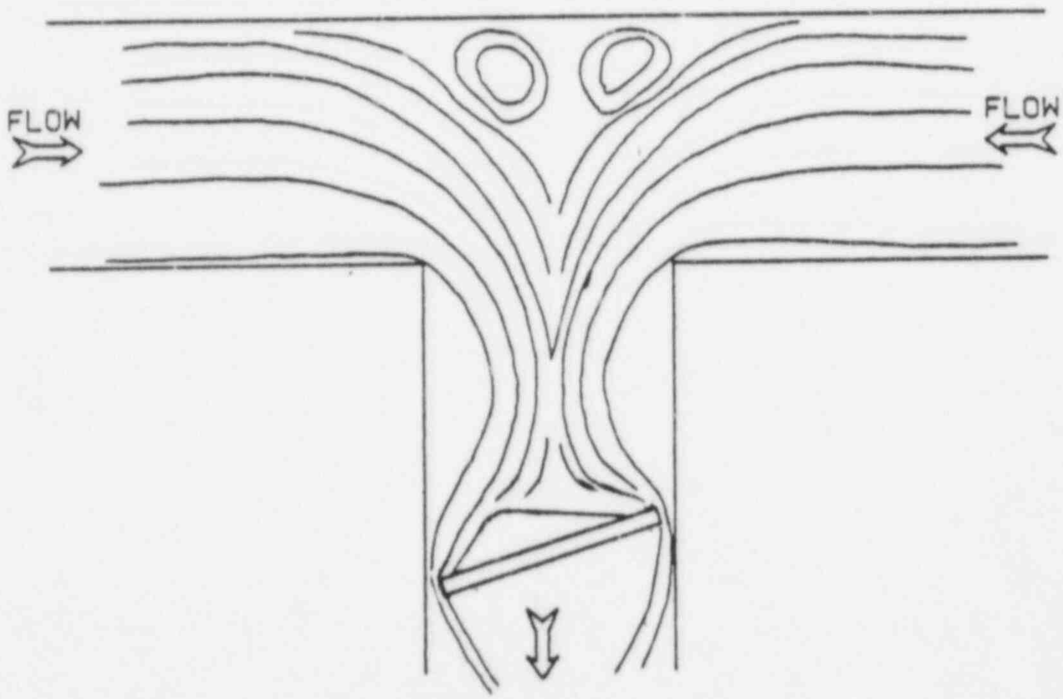



FIGURE 25A
TEE WITH FLOW FROM TWO SIDES

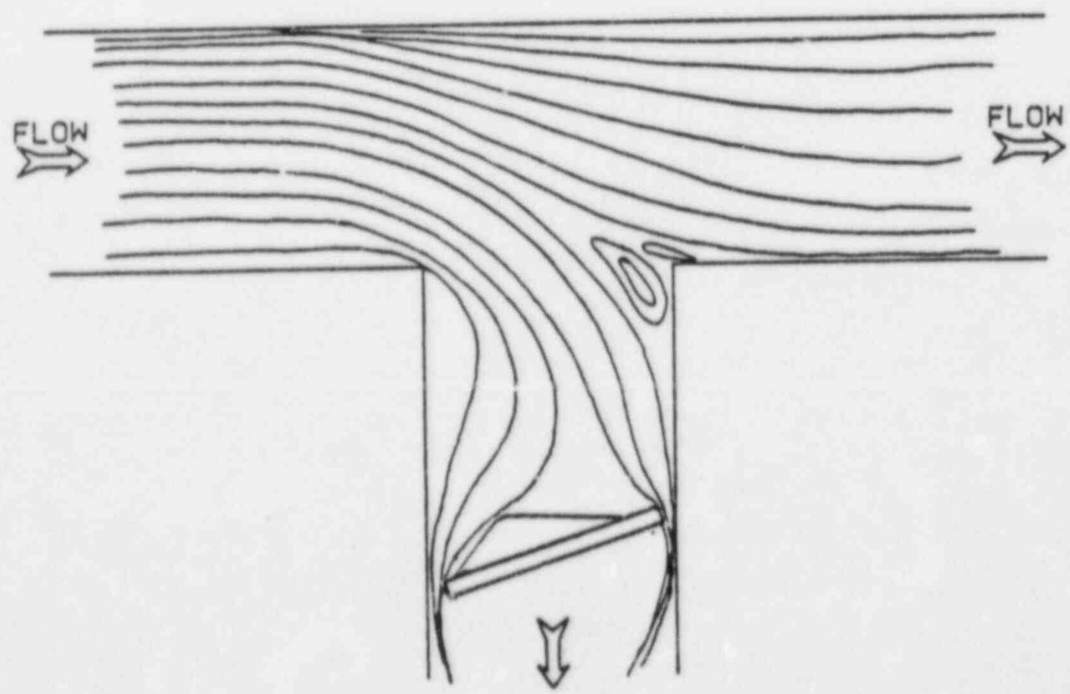



FIGURE 25B
TEE WITH FLOW FROM ONE SIDE

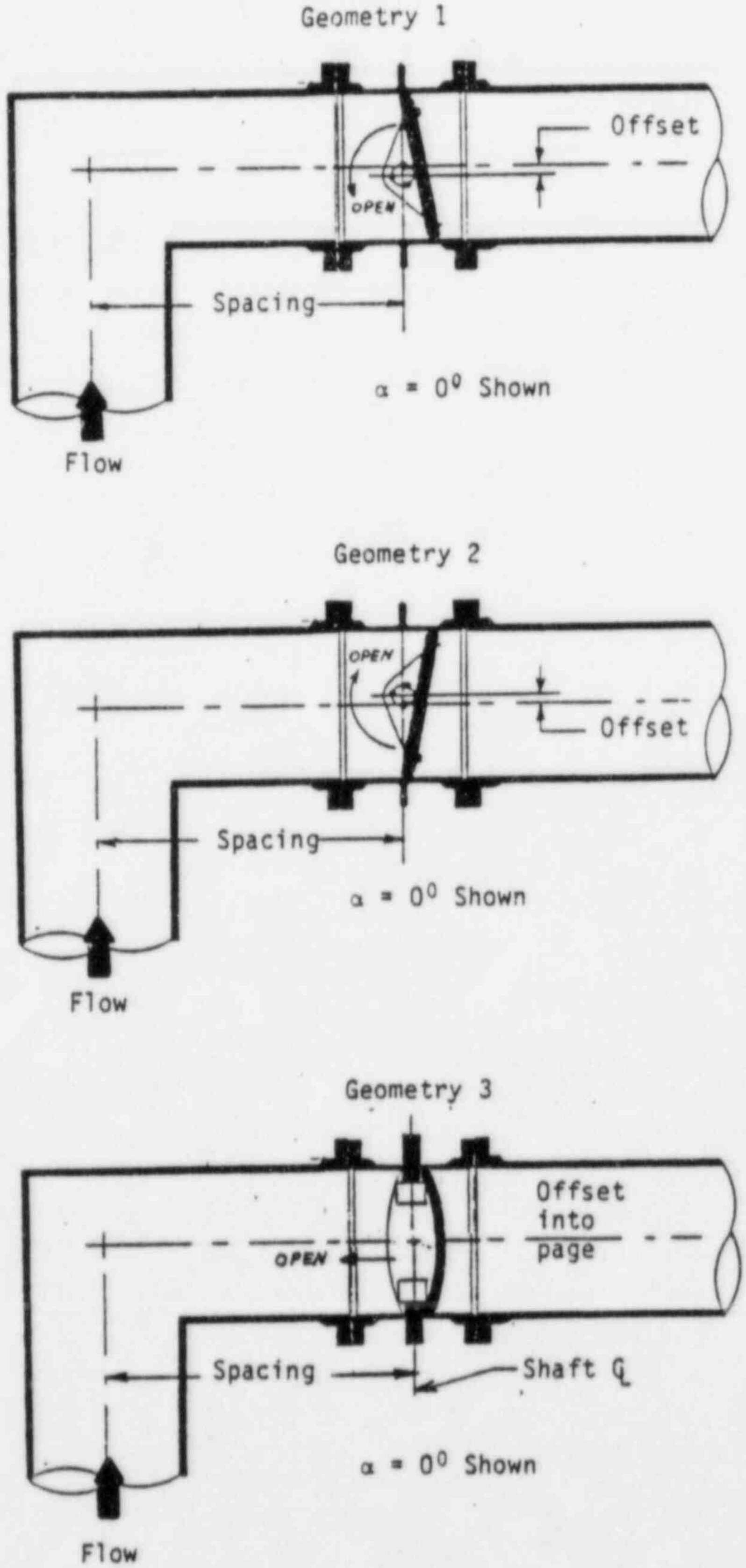



FIGURE 26 - Valve Orientations Relative to Upstream Elbow

### 5.1.3 DOWNSTREAM PIPING EFFECTS

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

## 5.3.0 APPLICATION OF MODEL AERODYNAMIC TEST TO FULL SIZE VALVE OPERABILITY

### 5.3.1 VALVE OPERATING TIMES EXPECTED IN SERVICE

All valves are designed to close within 5 seconds for flow conditions produced by the maximum differential pressure (see 3.0 Table 6). These are the maximum conditions expected in the event of a LOCA. The valves are designed to open within 5 seconds for conditions of normal flow even though most are capable of opening within this time for maximum pressure differential. All except for the 20 inch valves are of fail closed design through use of a return spring in the actuator. The 20 inch valves are of fail open design through use of a return spring. These will close in 5 seconds or less if the air supply to the actuator is adequate. The required air supply for this fucntion has been specified in Clow's "Operating Instructions". Meeting this requirement is the responsibility of the buyer. Opening times measured during in-house bench tests for the 18" valves of 7 seconds were obtained with approximately a 70 SCFM air supply. These valves will open within 5 seconds if a 100 SCFM or better air supply is provided to the actuator.

In the Vought test (Reference 7.0 B-3) closing times were shown to improve slightly with flow through the valve. Opening times were retarded on the order of ½ 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 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.

### 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 22 thru 31 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. For Figure 15 (Section 3.2) the configuration for V-23-16 is most closely related to straight line flow since the distance from the tee is greater than 4 diameters, the reducer will have an additional straightening effect, and some flow is directed through the other

Table 22

```
CASE: V-23-13 & 14  NORMAL OPERATING PRESSURE
DATE: 3-3-82                                             UNITS SYSTEM:  ES
PATM:     14.70(PSIA)                                    SHAFT:  US
PSU =     39.70(PSIA)    TSU =  559.67(R)
MEDIUM:  GAS = A         GAMMA =  1.40                   MW =  29.0
FLOW = UF                OPTION = 1
W80 =     1200.00(SCFM )
DU =   8.000(IN)
```

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OUTPUT DATA

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SOLUTION:  DP80 =      .03(PSIG)

PSD/P0U =    .9990

| ALPHA | CF | WR | DPS/PSU | PSU/P0U | PSC/P0U | P0D/P0U | TQR1 |
|---|---|---|---|---|---|---|---|
| 80.0 | .5162 | 1.0000 | .0007 | .9997 | .7481 | .9977 | .1025 |
| 75.0 | .5066 | .9814 | .0007 | .9997 | .7464 | .9935 | .1425 |
| 70.0 | .4875 | .9444 | .0007 | .9998 | .7431 | .9900 | .1715 |
| 65.0 | .4604 | .8918 | .0008 | .9998 | .7374 | .9874 | .1908 |
| 60.0 | .4266 | .8264 | .0008 | .9998 | .7294 | .9856 | .2019 |
| 55.0 | .3877 | .7510 | .0008 | .9999 | .7188 | .9845 | .2061 |
| 50.0 | .3449 | .6682 | .0009 | .9999 | .7056 | .9842 | .2048 |
| 45.0 | .2999 | .5808 | .0009 | .9999 | .6897 | .9847 | .1993 |
| 40.0 | .2539 | .4917 | .0009 | .9999 | .6715 | .9860 | .1908 |
| 35.0 | .2084 | .4036 | .0009 | 1.0000 | .6514 | .9882 | .1806 |
| 30.0 | .1648 | .3193 | .0009 | 1.0000 | .6301 | .9911 | .1699 |
| 25.0 | .1246 | .2414 | .0010 | 1.0000 | .6085 | .9947 | .1599 |
| 20.0 | .0892 | .1729 | .0010 | 1.0000 | .5879 | .9992 | .1518 |
| 15.0 | .0601 | .1164 | .0010 | 1.0000 | .5698 | 1.0000 | .1467 |
| 10.0 | .0385 | .0747 | .0010 | 1.0000 | .5556 | 1.0000 | .1458 |
| 5.0 | .0261 | .0505 | .0010 | 1.0000 | .5470 | 1.0000 | .1500 |

| ALPHA (DEG) | VCU (....) | W (LBM/HR) | TQ (IN-LBF) |
|---|---|---|---|
| 80.0 | 1151.88 | 5379.22 | 1.49 |
| 75.0 | 1123.18 | 5279.12 | 2.10 |
| 70.0 | 1067.10 | 5080.24 | 2.60 |
| 65.0 | 991.79 | 4797.43 | 2.98 |
| 60.0 | 901.73 | 4445.59 | 3.28 |
| 55.0 | 804.27 | 4039.57 | 3.48 |
| 50.0 | 702.78 | 3594.25 | 3.58 |
| 45.0 | 600.84 | 3124.51 | 3.60 |
| 40.0 | 501.72 | 2645.22 | 3.54 |
| 35.0 | 407.43 | 2171.24 | 3.43 |
| 30.0 | 319.59 | 1717.46 | 3.28 |
| 25.0 | 240.23 | 1298.73 | 3.12 |
| 20.0 | 171.33 | 929.94 | 2.99 |
| 15.0 | 115.01 | 625.94 | 2.90 |
| 10.0 | 73.71 | 401.61 | 2.89 |
| 5.0 | 49.88 | 271.79 | 2.98 |

Table 23

```
CASE: V-23-13 & 14  MAX. OPERATING PRESSURE           UNITS SYSTEM:  ES
DATE: 3-3-82                                          SHAFT:  US
PATM:     14.70(PSIA)
PSU =     76.70(PSIA)      TSU =  799.67(R)           MW =  29.0
MEDIUM:  GAS - A           GAMMA =  1.40
FLOW = CF                  OPTION = 2
DU =    8.000(IN)
```

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

CHOKING PRESSURE RATIOS:  PSC/P0U =    .748     DPS/PSU =    .198
SOLUTION: W80 =      39.75(LBM/S)

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

PSD/P0U =     .7481

| ALPHA | CF | WR | DPS/PSU | PSU/P0U | PSC/P0U | P0D/P0U | TQR1 |
|---|---|---|---|---|---|---|---|
| 80.0 | .5162 | 1.0000 | .1979 | .9327 | .7481 | .8620 | .0864 |
| 75.0 | .5066 | .9814 | .2002 | .9353 | .7464 | .8512 | .1262 |
| 70.0 | .4875 | .9444 | .2046 | .9405 | .7431 | .8394 | .1548 |
| 65.0 | .4604 | .8918 | .2103 | .9473 | .7374 | .8272 | .1737 |
| 60.0 | .4266 | .8264 | .2168 | .9552 | .7294 | .8149 | .1842 |
| 55.0 | .3877 | .7510 | .2235 | .9633 | .7188 | .8030 | .1879 |
| 50.0 | .3449 | .6682 | .2298 | .9712 | .7056 | .7918 | .1861 |
| 45.0 | .2999 | .5808 | .2354 | .9784 | .6897 | .7816 | .1801 |
| 40.0 | .2539 | .4917 | .2403 | .9846 | .6715 | .7726 | .1712 |
| 35.0 | .2084 | .4036 | .2442 | .9897 | .6514 | .7651 | .1606 |
| 30.0 | .1648 | .3193 | .2471 | .9936 | .6301 | .7590 | .1497 |
| 25.0 | .1246 | .2414 | .2492 | .9963 | .6085 | .7543 | .1396 |
| 20.0 | .0892 | .1729 | .2505 | .9981 | .5879 | .7511 | .1314 |
| 15.0 | .0601 | .1164 | .2513 | .9992 | .5698 | .7492 | .1262 |
| 10.0 | .0385 | .0747 | .2517 | .9996 | .5556 | .7485 | .1252 |
| 5.0 | .0261 | .0505 | .2518 | .9998 | .5470 | .7487 | .1295 |

| ALPHA (DEG) | YCU (....) | W (LBM/HR) | TQ (IN-LBF) | TQA (IN-LBF) |
|---|---|---|---|---|
| 80.0 | 1152.62 | 143116.25 | 671.42 | 2742.26 |
| 75.0 | 1121.44 | 140453.06 | 995.19 | 4017.93 |
| 70.0 | 1061.79 | 135161.62 | 1254.19 | 4955.61 |
| 65.0 | 981.71 | 127637.56 | 1457.11 | 5599.84 |
| 60.0 | 888.66 | 118276.56 | 1606.54 | 5989.57 |
| 55.0 | 788.71 | 107474.37 | 1703.08 | 6161.22 |
| 50.0 | 686.43 | 95626.44 | 1748.28 | 6150.98 |
| 45.0 | 585.15 | 83128.81 | 1746.35 | 5996.23 |
| 40.0 | 487.29 | 70376.97 | 1704.82 | 5736.04 |
| 35.0 | 394.75 | 57766.67 | 1634.24 | 5410.96 |
| 30.0 | 309.16 | 45693.58 | 1547.50 | 5062.52 |
| 25.0 | 232.16 | 34553.27 | 1458.81 | 4732.50 |
| 20.0 | 165.49 | 24741.39 | 1382.90 | 4462.18 |
| 15.0 | 111.11 | 16653.43 | 1334.19 | 4291.85 |
| 10.0 | 71.20 | 10684.90 | 1326.39 | 4260.45 |
| 5.0 | 48.16 | 7231.12 | 1372.32 | 4405.52 |

Table 24

CASE: V-23-15 & 16  NORMAL OPERATING PRESSURE
DATE: 3-3-82
PATM: 14.70(PSIA)
PSU = 15.70(PSIA)    TSU = 559.67(R)
MEDIUM: GAS - A    GAMMA = 1.40
FLOW - UF    OPTION = 1
W80 = 1200.00(SCFM )
DU = 8.000(IN)

UNITS SYSTEM: ES
SHAFT: US

MU = 29.0

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

SOLUTION: DP80 = .07(PSIG)

PSD/P0U = .9937

| ALPHA | CF | WR | DPS/PSU | PSU/P0U | PSC/P0U | P0D/P0U | TQR1 |
|---|---|---|---|---|---|---|---|
| 80.0 | .5162 | 1.0000 | .0046 | .9983 | .7481 | .9950 | .1022 |
| 75.0 | .5066 | .9814 | .0047 | .9984 | .7464 | .9906 | .1422 |
| 70.0 | .4875 | .9444 | .0048 | .9985 | .7431 | .9870 | .1712 |
| 65.0 | .4604 | .8918 | .0049 | .9987 | .7374 | .9842 | .1905 |
| 60.0 | .4266 | .8264 | .0051 | .9969 | .7294 | .9821 | .2016 |
| 55.0 | .3877 | .7510 | .0053 | .9991 | .7188 | .9808 | .2058 |
| 50.0 | .3449 | .6682 | .0055 | .9993 | .7056 | .9802 | .2044 |
| 45.0 | .2999 | .5808 | .0057 | .9994 | .6897 | .9805 | .1989 |
| 40.0 | .2539 | .4917 | .0059 | .9996 | .6715 | .9816 | .1904 |
| 35.0 | .2084 | .4036 | .0060 | .9997 | .6514 | .9835 | .1801 |
| 30.0 | .1648 | .3193 | .0061 | .9998 | .6301 | .9861 | .1694 |
| 25.0 | .1246 | .2414 | .0062 | .9999 | .6085 | .9896 | .1595 |
| 20.0 | .0892 | .1729 | .0062 | .9999 | .5879 | .9940 | .1514 |
| 15.0 | .0601 | .1164 | .0062 | 1.0000 | .5698 | .9991 | .1463 |
| 10.0 | .0385 | .0747 | .0063 | 1.0000 | .5556 | 1.0000 | .1453 |
| 5.0 | .0261 | .0505 | .0063 | 1.0000 | .5470 | 1.0000 | .1496 |

| ALPHA (DEG) | YCU (....) | W (LBM/HR) | TQ (IN-LBF) |
|---|---|---|---|
| 80.0 | 1149.37 | 5379.22 | 3.79 |
| 75.0 | 1120.60 | 5279.12 | 5.34 |
| 70.0 | 1064.79 | 5080.24 | 6.59 |
| 65.0 | 988.97 | 4797.44 | 7.58 |
| 60.0 | 899.48 | 4445.59 | 8.32 |
| 55.0 | 802.01 | 4039.57 | 8.82 |
| 50.0 | 700.75 | 3594.25 | 9.09 |
| 45.0 | 599.37 | 3124.51 | 9.13 |
| 40.0 | 500.46 | 2645.22 | 8.98 |
| 35.0 | 406.31 | 2171.24 | 8.69 |
| 30.0 | 318.67 | 1717.46 | 8.31 |
| 25.0 | 239.54 | 1298.73 | 7.91 |
| 20.0 | 170.87 | 929.94 | 7.57 |
| 15.0 | 114.76 | 625.94 | 7.35 |
| 10.0 | 73.55 | 401.61 | 7.31 |
| 5.0 | 49.76 | 271.79 | 7.53 |

Table 25

```
CASE: V-23-15 & 16  MAX. OPERATING PRESSURE
DATE: 3-3-82                                                   UNITS SYSTEM:  ES
PATM:     14.70(PSIA)                                          SHAFT:  US
PSU =     49.70(PSIA)      TSU =  799.67(R)
MEDIUM:  GAS = A           GAMMA =  1.40                       MW =, 29.0
FLOW = CF                  OPTION = 2
DV =   8.000(IN)
```

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

CHOKING PRESSURE RATIOS:  PSC/P0U =  .748     DPS/PSU =  .198
SOLUTION: W80 =     25.76(LBM/S)

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

PSD/P0U =    .7481

| ALPHA | CF | WR | DPS/PSU | PSU/P0U | PSC/P0U | P0D/P0U | TQR1 |
|---|---|---|---|---|---|---|---|
| 80.0 | .5162 | 1.0000 | .1979 | .9327 | .7481 | .8620 | .0864 |
| 75.0 | .5066 | .9814 | .2002 | .9353 | .7464 | .8512 | .1262 |
| 70.0 | .4875 | .9444 | .2046 | .9405 | .7431 | .8394 | .1548 |
| 65.0 | .4604 | .8918 | .2103 | .9473 | .7374 | .8272 | .1737 |
| 60.0 | .4266 | .8264 | .2168 | .9552 | .7294 | .8149 | .1842 |
| 55.0 | .3877 | .7510 | .2235 | .9633 | .7188 | .8030 | .1879 |
| 50.0 | .3449 | .6682 | .2298 | .9712 | .7056 | .7918 | .1861 |
| 45.0 | .2999 | .5808 | .2354 | .9784 | .6897 | .7816 | .1801 |
| 40.0 | .2539 | .4917 | .2403 | .9846 | .6715 | .7726 | .1712 |
| 35.0 | .2084 | .4036 | .2442 | .9897 | .6514 | .7651 | .1606 |
| 30.0 | .1648 | .3193 | .2471 | .9936 | .6301 | .7590 | .1497 |
| 25.0 | .1246 | .2414 | .2492 | .9963 | .6085 | .7543 | .1396 |
| 20.0 | .0892 | .1729 | .2505 | .9981 | .5879 | .7511 | .1314 |
| 15.0 | .0601 | .1164 | .2513 | .9992 | .5698 | .7492 | .1262 |
| 10.0 | .0385 | .0747 | .2517 | .9996 | .5556 | .7485 | .1252 |
| 5.0 | .0261 | .0505 | .2518 | .9998 | .5470 | .7487 | .1295 |

| ALPHA (DEG) | YCU (....) | W (LBM/HR) | TQ (IN-LBF) | TQA (IN-LBF) |
|---|---|---|---|---|
| 80.0 | 1152.62 | 92733.69 | 435.05 | 1548.05 |
| 75.0 | 1121.44 | 91008.09 | 644.84 | 2268.19 |
| 70.0 | 1061.79 | 87579.44 | 812.67 | 2797.52 |
| 65.0 | 981.71 | 82704.12 | 944.15 | 3161.20 |
| 60.0 | 888.66 | 76638.56 | 1040.97 | 3381.21 |
| 55.0 | 788.71 | 69639.19 | 1103.53 | 3478.11 |
| 50.0 | 686.43 | 61962.17 | 1132.82 | 3472.33 |
| 45.0 | 585.16 | 53864.19 | 1131.57 | 3384.98 |
| 40.0 | 487.29 | 45601.52 | 1104.66 | 3238.09 |
| 35.0 | 394.75 | 37430.52 | 1058.93 | 3054.58 |
| 30.0 | 309.16 | 29607.65 | 1002.72 | 2857.88 |
| 25.0 | 232.16 | 22389.16 | 945.25 | 2671.57 |
| 20.0 | 165.49 | 16031.45 | 896.06 | 2518.98 |
| 15.0 | 111.11 | 10790.77 | 864.50 | 2422.82 |
| 10.0 | 71.20 | 6923.39 | 859.45 | 2405.09 |
| 5.0 | 48.16 | 4685.48 | 889.21 | 2486.99 |

## Table 26

CASE: V-28-17 & 18 NORMAL OPERATING PRESSURE
DATE: 3-3-82
PATM: 14.70(PSIA)
PSU - 15.70(PSIA)
MEDIUM: GAS - A
FLOW - UF
WS0 - 1200.00(SCFM )
DV - 12.000(IN)

TSU - 559.67(R)
GAMMA - 1.40
OPTION - 1

UNITS SYSTEM: ES
SHAFT: US

MU - 29.0

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

SOLUTION: DP80 - .01(PSIG)

PSD/P0U - .9989

| ALPHA | CF | WR | DPS/PSU | PSU/P0U | P>C/P6J | P0D/P0U | TQR1 |
|---|---|---|---|---|---|---|---|
| 80.0 | .5447 | 1.0000 | .0008 | .9997 | .7523 | .9976 | .0828 |
| 75.0 | .5345 | .9814 | .0008 | .9997 | .7509 | .9934 | .1269 |
| 70.0 | .5144 | .9444 | .0008 | .9997 | .7477 | .9900 | .1573 |
| 65.0 | .4858 | .8918 | .0008 | .9997 | .7427 | .9873 | .1764 |
| 60.0 | .4501 | .8264 | .0009 | .9998 | .7351 | .9855 | .1863 |
| 55.0 | .4090 | .7510 | .0009 | .9998 | .7248 | .9844 | .1889 |
| 50.0 | .3639 | .6682 | .0010 | .9999 | .7117 | .9841 | .1861 |
| 45.0 | .3164 | .5808 | .0010 | .9999 | .6957 | .9846 | .1796 |
| 40.0 | .2678 | .4917 | .0010 | .9999 | .6772 | .9859 | .1707 |
| 35.0 | .2198 | .4036 | .0011 | .9999 | .6566 | .9880 | .1608 |
| 30.0 | .1739 | .3193 | .0011 | 1.0000 | .6347 | .9909 | .1509 |
| 25.0 | .1315 | .2414 | .0011 | 1.0000 | .6124 | .9946 | .1421 |
| 20.0 | .0942 | .1729 | .0011 | 1.0000 | .5909 | .9991 | .1349 |
| 15.0 | .0634 | .1164 | .0011 | 1.0000 | .5719 | 1.0000 | .1301 |
| 10.0 | .0407 | .0747 | .0011 | 1.0000 | .5570 | 1.0000 | .1281 |
| 5.0 | .0275 | .0505 | .0011 | 1.0000 | .5482 | 1.0000 | .1289 |

| ALPHA (DEG) | YCV (....) | W (LBM/HR) | TQ (IN-LBF) |
|---|---|---|---|
| 80.0 | 2792.22 | 5379.22 | 1.75 |
| 75.0 | 2721.19 | 5279.12 | 2.73 |
| 70.0 | 2581.20 | 5080.24 | 3.48 |
| 65.0 | 2391.36 | 4797.44 | 4.05 |
| 60.0 | 2167.90 | 4445.59 | 4.47 |
| 55.0 | 1929.20 | 4039.57 | 4.72 |
| 50.0 | 1681.20 | 3594.25 | 4.85 |
| 45.0 | 1434.82 | 3124.51 | 4.86 |
| 40.0 | 1195.69 | 2645.22 | 4.76 |
| 35.0 | 969.71 | 2171.24 | 4.60 |
| 30.0 | 759.76 | 1717.46 | 4.40 |
| 25.0 | 570.61 | 1298.73 | 4.20 |
| 20.0 | 406.80 | 929.94 | 4.02 |
| 15.0 | 273.16 | 625.94 | 3.90 |
| 10.0 | 176.05 | 401.61 | 3.84 |
| 5.0 | 118.38 | 271.79 | 3.88 |

Table 27

CASE: V-20-17 & 18 MAX. OPERATING PRESSURE
DATE: 3-3-82
PATM: 14.70(PSIA)
PSU = 49.70(PSIA)
MEDIUM: GAS - A
FLOW - CF
DU = 12.000(IN)

TSU = 609.67(R)
GAMMA = 1.40
OPTION = 2

UNITS SYSTEM: ES
SHAFT: US

MW = 29.0

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

CHOKING PRESSURE RATIOS: PSC/P0U = .752 DPS/PSU = .186
SOLUTION: W80 = 70.58(LBM/S)

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

PSD/P0U = .7523

| ALPHA | CF | WR | DPS/PSU | PSU/P0U | PSC/P0U | P0D/P0U | TQR1 |
|---|---|---|---|---|---|---|---|
| 80.0 | .5447 | 1.0000 | .1861 | .9243 | .7523 | .8701 | .0676 |
| 75.0 | .5345 | .9814 | .1888 | .9274 | .7509 | .8592 | .1115 |
| 70.0 | .5144 | .9444 | .1939 | .9332 | .7477 | .8473 | .1415 |
| 65.0 | .4858 | .8918 | .2005 | .9409 | .7427 | .8347 | .1600 |
| 60.0 | .4501 | .8264 | .2080 | .9498 | .7351 | .8219 | .1693 |
| 55.0 | .4090 | .7510 | .2155 | .9590 | .7248 | .8094 | .1713 |
| 50.0 | .3639 | .6682 | .2227 | .9678 | .7117 | .7976 | .1679 |
| 45.0 | .3164 | .5808 | .2291 | .9759 | .6957 | .7870 | .1608 |
| 40.0 | .2678 | .4917 | .2346 | .9829 | .6772 | .7776 | .1515 |
| 35.0 | .2198 | .4036 | .2390 | .9885 | .6566 | .7698 | .1412 |
| 30.0 | .1739 | .3193 | .2423 | .9929 | .6347 | .7634 | [illegible] |
| 25.0 | .1315 | .2414 | .2446 | .9959 | .6124 | .7587 | [illegible] |
| 20.0 | .0942 | .1729 | .2461 | .9979 | .5909 | .7554 | .1148 |
| 15.0 | .0634 | .1164 | .2470 | .9991 | .5719 | .7536 | .1099 |
| 10.0 | .0407 | .0747 | .2474 | .9996 | .5570 | .7529 | .1078 |
| 5.0 | .0275 | .0505 | .2476 | .9998 | .5482 | .7532 | .1087 |

| ALPHA (DEG) | YCU (....) | W (LBM/HR) | TQ (IN-LBF) | TQA (IN-LBF) |
|---|---|---|---|---|
| 80.0 | 2847.28 | 254077.06 | 1080.48 | 4088.73 |
| 75.0 | 2765.31 | 249349.19 | 1813.43 | 6765.09 |
| 70.0 | 2609.77 | 239955.00 | 2377.96 | 8639.44 |
| 65.0 | 2403.30 | 226597.44 | 2804.47 | 9851.18 |
| 60.0 | 2166.39 | 209978.81 | 3105.82 | 10518.71 |
| 55.0 | 1915.11 | 190801.25 | 3288.93 | 10747.20 |
| 50.0 | 1660.91 | 169767.44 | 3362.78 | 10633.93 |
| 45.0 | 1411.67 | 147580.25 | 3341.70 | 10270.78 |
| 40.0 | 1172.79 | 124941.66 | 3245.94 | 9744.52 |
| 35.0 | 948.32 | 102554.34 | 3100.05 | 9135.86 |
| 30.0 | 741.73 | 81120.72 | 2930.41 | 8517.74 |
| 25.0 | 556.48 | 61343.09 | 2762.76 | 7953.68 |
| 20.0 | 396.45 | 43923.86 | 2619.92 | 7496.31 |
| 15.0 | 266.08 | 29565.16 | 2520.37 | 7186.37 |
| 10.0 | 170.48 | 18969.11 | 2477.57 | 7052.30 |
| 5.0 | 115.31 | 12837.54 | 2499.60 | 7110.56 |

Table 28

```
CASE: V-27-1,2,3,4  NORMAL OPERATING PRESSURE
DATE: 3-3-82                                            UNITS SYSTEM:  ES
PATM:     14.70(PSIA)                                   SHAFT:  US
PSU =     15.70(PSIA)     TSU =  559.67(R)
MEDIUM:   GAS - A         GAMMA =  1.40                 MW =  29.0
FLOW = UF                 OPTION = 1
WB0 =     6200.00(SCFM )
DU =  18.000(IN)
```

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

SOLUTION: DP80 = .06(PSIG)

PSD/P0U = .9947

| ALPHA | CF | WR | DPS/PSU | PSU/P0U | PSC/P0U | P0D/P0U | TQR1 |
|---|---|---|---|---|---|---|---|
| 80.0 | .5747 | 1.0000 | .0035 | .9983 | .7558 | .9957 | .0530 |
| 75.0 | .5640 | .9814 | .0036 | .9983 | .7547 | .9914 | .1033 |
| 70.0 | .5427 | .9444 | .0037 | .9985 | .7520 | .9878 | .1359 |
| 65.0 | .5125 | .8918 | .0039 | .9986 | .7474 | .9850 | .1545 |
| 60.0 | .4749 | .8264 | .0041 | .9988 | .7405 | .9829 | .1625 |
| 55.0 | .4316 | .7510 | .0043 | .9990 | .7306 | .9816 | .1628 |
| 50.0 | .3840 | .6682 | .0045 | .9992 | .7177 | .9811 | .1578 |
| 45.0 | .3338 | .5808 | .0047 | .9994 | .7018 | .9814 | .1498 |
| 40.0 | .2826 | .4917 | .0048 | .9996 | .6831 | .9825 | .1403 |
| 35.0 | .2320 | .4036 | .0050 | .9997 | .6620 | .9844 | .1308 |
| 30.0 | .1835 | .3193 | .0051 | .9998 | .6394 | .9871 | .1222 |
| 25.0 | .1387 | .2414 | .0052 | .9999 | .6163 | .9906 | .1150 |
| 20.0 | .0993 | .1729 | .0052 | .9999 | .5940 | .9950 | .1093 |
| 15.0 | .0669 | .1164 | .0052 | 1.0000 | .5741 | 1.0000 | .1049 |
| 10.0 | .0429 | .0747 | .0052 | 1.0000 | .5586 | 1.0000 | .1012 |
| 5.0 | .0290 | .0505 | .0053 | 1.0000 | .5491 | 1.0000 | .0970 |

| ALPHA (DEG) | YCU (....) | W (LBM/HR) | TQ (IN-LBF) |
|---|---|---|---|
| 80.0 | 6781.33 | 27792.64 | 17.16 |
| 75.0 | 6596.57 | 27275.48 | 34.00 |
| 70.0 | 6241.04 | 26247.89 | 46.28 |
| 65.0 | 5765.04 | 24786.74 | 54.99 |
| 60.0 | 5212.51 | 22968.87 | 60.74 |
| 55.0 | 4620.13 | 20871.12 | 63.93 |
| 50.0 | 4016.59 | 18570.30 | 64.91 |
| 45.0 | 3420.25 | 16143.31 | 64.18 |
| 40.0 | 2845.42 | 13666.95 | 62.26 |
| 35.0 | 2303.24 | 11218.08 | 59.69 |
| 30.0 | 1803.00 | 8873.53 | 56.92 |
| 25.0 | 1353.21 | 6710.12 | 54.37 |
| 20.0 | 964.32 | 4804.69 | 52.18 |
| 15.0 | 647.36 | 3234.03 | 50.35 |
| 10.0 | 414.78 | 2074.97 | 48.68 |
| 5.0 | 280.57 | 1404.26 | 46.72 |

Table 29

```
CASE: V-27-1,2,3,4  MAX OPERATING PRESSURE
DATE: 3-3-82                                           UNITS SYSTEM:  ES
PATM:     14.70(PSIA)                                  SHAFT:  US
PSU -     76.70(PSIA)     TSU -  799.67(R)
MEDIUM:   GAS - A         GAMMA -  1.40                MU -  29.0
FLOW - CF                 OPTION - 2
DU -  18.000(IN)

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

CHOKING PRESSURE RATIOS:  PSC/P0U -    .756      DPS/PSU -    .174
SOLUTION: W80 -     227.79(LBM/S)

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

PSD/P0U -     .7558
```

| ALPHA | CF | WR | DPS/PSU | PSU/P0U | PSC/P0U | P0D/P0U | 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) | VCU (....) | W (LBM/HR) | TQ (IN-LBF) | TQA (IN-LBF) |
|---|---|---|---|---|
| 80.0 | 7067.39 | 820030.00 | 3031.46 | 14100.26 |
| 75.0 | 6848.30 | 804771.00 | 7068.37 | 32296.06 |
| 70.0 | 6436.71 | 774451.75 | 10011.55 | 44268.45 |
| 65.0 | 5897.95 | 731340.00 | 12100.50 | 51340.03 |
| 60.0 | 5289.47 | 677703.25 | 13452.94 | 54615.77 |
| 55.0 | 4653.79 | 615808.50 | 14151.80 | 55050.81 |
| 50.0 | 4019.45 | 547922.25 | 14291.97 | 53487.03 |
| 45.0 | 3404.62 | 476313.12 | 13995.05 | 50665.02 |
| 40.0 | 2820.83 | 403247.37 | 13403.25 | 47219.42 |
| 35.0 | 2276.20 | 330892.50 | 12660.98 | 43665.02 |
| 30.0 | 1777.64 | 261815.75 | 11895.48 | 40381.58 |
| 25.0 | 1332.31 | 197983.94 | 11197.63 | 37600.14 |
| 20.0 | 948.55 | 141763.69 | 10614.6[illegible] | 35394.72 |
| 15.0 | 636.39 | 95421.12 | 10139.94 | 33677.72 |
| 10.0 | 407.67 | 61222.50 | 9713.52 | 32199.75 |
| 5.0 | 275.73 | 41432.98 | 9222.7[illegible] | 30550.74 |

Table 30

CASE: V-26-16 & 18 NORMAL OPERATING PRESSURE
DATE: 3-3-82
PATM: 14.70(PSIA)
PSU = 15.70(PSIA)    TSU = 559.67(R)
MEDIUM: GAS - A    GAMMA = 1.40
FLOW - UF    OPTION = 1
W80 = 30200.00(SCFM )
DU = 20.000(IN)

UNITS SYSTEM: ES
SHAFT: US

MW = 29.0

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

SOLUTION: DP80 = .91(PSIG)

PSD/P0U = .9168

| ALPHA | CF | WR | DPS/PSU | PSU/P0U | PSC/P0U | P0D/P0U | TQR1 |
|---|---|---|---|---|---|---|---|
| 80.0 | .5827 | 1.0000 | .0583 | .9735 | .7566 | .9577 | .0387 |
| 75.0 | .5719 | .9814 | .0592 | .9745 | .7555 | .9514 | .0909 |
| 70.0 | .5504 | .9444 | .0611 | .9764 | .7530 | .9451 | .1240 |
| 65.0 | .5197 | .8910 | .0636 | .9791 | .7486 | .9390 | .1424 |
| 60.0 | .4816 | .8264 | .0665 | .9821 | .7418 | .9332 | .1495 |
| 55.0 | .4376 | .7510 | .0695 | .9853 | .7321 | .9280 | .1488 |
| 50.0 | .3894 | .6682 | .0724 | .9884 | .7193 | .9236 | .1429 |
| 45.0 | .3385 | .5808 | .0751 | .9912 | .7034 | .9199 | .1341 |
| 40.0 | .2866 | .4917 | .0774 | .9937 | .6846 | .9173 | .1243 |
| 35.0 | .2352 | .4036 | .0793 | .9958 | .6635 | .9157 | .1148 |
| 30.0 | .1861 | .3193 | .0808 | .9974 | .6407 | .9152 | .1065 |
| 25.0 | .1407 | .2414 | .0818 | .9985 | .6174 | .9158 | .0997 |
| 20.0 | .1007 | .1729 | .0825 | .9992 | .5948 | .9176 | .0945 |
| 15.0 | .0678 | .1164 | .0829 | .9996 | .5747 | .9204 | .0902 |
| 10.0 | .0435 | .0747 | .0831 | .9999 | .5590 | .9241 | .0859 |
| 5.0 | .0294 | .0505 | .0831 | .9999 | .5493 | .9288 | .0800 |

| ALPHA (DEG) | YCU (....) | W (LBM/HR) | TQ (IN-LBF) |
|---|---|---|---|
| 80.0 | 8164.78 | 135377.06 | 282.84 |
| 75.0 | 7938.62 | 132857.94 | 676.57 |
| 70.0 | 7507.24 | 127852.59 | 954.49 |
| 65.0 | 6929.97 | 120735.41 | 1143.31 |
| 60.0 | 6261.94 | 111880.69 | 1259.00 |
| 55.0 | 5547.52 | 101662.56 | 1313.58 |
| 50.0 | 4819.87 | 90455.28 | 1318.78 |
| 45.0 | 4102.46 | 78633.56 | 1287.53 |
| 40.0 | 3411.97 | 66571.28 | 1233.51 |
| 35.0 | 2761.08 | 54642.86 | 1169.72 |
| 30.0 | 2160.79 | 43222.66 | 1106.69 |
| 25.0 | 1621.74 | 32684.77 | 1051.17 |
| 20.0 | 1155.62 | 23403.48 | 1004.93 |
| 15.0 | 775.71 | 15752.87 | 964.36 |
| 10.0 | 497.03 | 10107.10 | 920.35 |
| 5.0 | 336.20 | 6840.09 | 858.28 |

Table 31

```
CASE: V-26-16 & 18  MAX. OPERATING PRESSURE
DATE: 3-3-82                                    UNITS SYSTEM:  ES
PATM:     14.70(PSIA)                           SHAFT:  US
PSU =     49.70(PSIA)    TSU =  609.67(R)
MEDIUM:  GAS - A         GAMMA =  1.40          MW =  29.0
FLOW = CF                OPTION = 2
DV =  20.000(IN)

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

CHOKING PRESSURE RATIOS:  PSC/P0U =   .757     DPS/PSU =   .171
SOLUTION: W80 =    212.15(LBM/S)

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

PSD/P0U =    .7566
```

| ALPHA | CF | WR | DPS/PSU | PSU/P0U | PSC/P0U | P0D/P0U | TQR1 |
|---|---|---|---|---|---|---|---|
| 80.0 | .5827 | 1.0000 | .1705 | .9121 | .7566 | .8807 | .0294 |
| 75.0 | .5719 | .9814 | .1738 | .9157 | .7555 | .8690 | .0814 |
| 70.0 | .5504 | .9444 | .1799 | .9226 | .7530 | .8575 | .1142 |
| 65.0 | .5197 | .8918 | .1879 | .9317 | .7486 | .8442 | .1321 |
| 60.0 | .4816 | .8264 | .1969 | .9420 | .7418 | .8305 | .1388 |
| 55.0 | .4376 | .7510 | .2059 | .9527 | .7321 | .8172 | .1375 |
| 50.0 | .3894 | .6682 | .2143 | .9630 | .7193 | .8046 | .1311 |
| 45.0 | .3385 | .5800 | .2219 | .9723 | .7034 | .7932 | .1220 |
| 40.0 | .2866 | .4917 | .2282 | .9803 | .6846 | .7832 | .1118 |
| 35.0 | .2352 | .4036 | .2333 | .9868 | .6635 | .7749 | .1021 |
| 30.0 | .1861 | .3193 | .2372 | .9918 | .6407 | .7682 | .0936 |
| 25.0 | .1407 | .2414 | .2399 | .9953 | .6174 | .7633 | .0867 |
| 20.0 | .1007 | .1729 | .2416 | .9976 | .5948 | .7599 | .0814 |
| 15.0 | .0678 | .1164 | .2426 | .9989 | .5747 | .7580 | .0770 |
| 10.0 | .0435 | .0747 | .2431 | .9996 | .5590 | .7574 | .0727 |
| 5.0 | .0294 | .05 5 | .2433 | .9998 | .5493 | .7578 | .0668 |

| ALPHA (DEG) | YCU (....) | W (LBM/HR) | TQ (IN-LBF) | TQA (IN-LBF) |
|---|---|---|---|---|
| 80.0 | 8958.55 | 763742.50 | 1992.65 | 8230.10 |
| 75.0 | 8674.78 | 749531.25 | 5646.90 | 22885.49 |
| 70.0 | 8143.31 | 721293.00 | 8264.58 | 32351.23 |
| 65.0 | 7450.65 | 681140.50 | 10080.86 | 37777.78 |
| 60.0 | 6671.87 | 631185.50 | 11215.08 | 40123.73 |
| 55.0 | 5861.98 | 573539.50 | 11754.27 | 40213.31 |
| 50.0 | 5056.98 | 510312.62 | 11798.11 | 38766.59 |
| 45.0 | 4279.29 | 443618.87 | 11469.05 | 36407.55 |
| 40.0 | 3542.79 | 375568.50 | 10907.78 | 33658.19 |
| 35.0 | 2857.12 | 308273.25 | 10245.77 | 30925.77 |
| 30.0 | 2230.38 | 243844.69 | 9594.06 | 28489.65 |
| 25.0 | 1671.15 | 184394.19 | 9022.12 | 26489.77 |
| 20.0 | 1189.58 | 132032.94 | 8549.02 | 24920.27 |
| 15.0 | 798.02 | 88871.41 | 8138.42 | 23626.42 |
| 10.0 | 511.18 | 57020.22 | 7698.53 | 22305.20 |
| 5.0 | 345.73 | 38589.02 | 7083.06 | 20506.57 |

branch. For Figure 16,valve V-26-16 is closer than 2 diameters (the limit of elbow test data) but flow going through the other branch should make it equal or less severe than the 2 diameter elbow data. For Figure 17,the 8 diameter elbow data may be used for valve V-23-14 and the 4 diameter elbow data may be used for V-23-13. The downstream elbows will probably make actual conditions less severe. For Figure 18,valve V-26-18, the 2 diameter elbow data is applicable since the upstream elbow is radiused (not mitered) and some back pressure would be present due to the downstream tee. For Figure 19 straight line data will be used since the upstream tee branches from an 18 inch line to a smaller 8 inch line. The 8 inch branch has only 20% of the flow area of the main line and would not substantially alter the flow pattern. For Figure 20&21, valves V-27-1 and V-28-17, will be represented by the 2 diameter elbow data. For V-27-1, the worst case orientation will be used even though the valve shaft axis is installed $30^{o}$ from this position. For valves V-23-15, V-27-2, V-28-18, and V-27-4, straight line torque modification based on 2 diameter elbow tests will be used as a worst case assumption (actual tests of 2 valves in series have been performed, but results have not been analyzed at the time this paper was written). The resultant torques are summarized in Table 32 thru 40 . The tables show model test valve angle and actual valve angle for the supplied units. There is a

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

Table 32



Valve No. V-23-16 (8")

Model Data For Torque Modification: Straight line flow
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 | 4 | 1548 | 1.0 | 4 | 1548 |
| 70 | 80 | 7 | 2798 | " | 7 | 2798 |
| 60 | 70 | 8 | 3381 | " | 8 | 3381 |
| 50 | 60 | 9 | 3472 | " | 9 | 3472 |
| 40 | 50 | 9 | 3238 | " | 9 | 3238 |
| 30 | 40 | 8 | 2858 | " | 8 | 2858 |
| 20 | 30 | 8 | 2519 | " | 8 | 2519 |
| 10 | 20 | 8 | 2405 | " | 8 | 2405 |

Table 33

Valve No. V-23-15 (8")

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 | Torque for Straight Flow Maximum | Torque Modification Factor | Torque for Installed Condition Normal | Torque for Installed Condition Maximum |
|---|---|---|---|---|---|---|
| 80 | 90 | 4 | 1548 | 2.21 | 9 | 3421 |
| 70 | 80 | 7 | 2798 | 1.56 | 11 | 4365 |
| 60 | 70 | 8 | 3381 | 1.12 | 9 | 3787 |
| 50 | 60 | 9 | 3472 | 1.05 | 9 | 3645 |
| 40 | 50 | 9 | 3238 | 1.0 | 9 | 3238 |
| 30 | 40 | 8 | 2858 | 1.0 | 8 | 2858 |
| 20 | 30 | 8 | 2519 | 1.0 | 8 | 2519 |
| 10 | 20 | 8 | 2405 | 1.0 | 8 | 2405 |

Table 34
Valve No. V-26-16 & 18 (20")

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 | Torque for Straight Flow Maximum | Torque Modification Factor | Torque for Installed Condition Normal | Torque for Installed Condition Maximum |
|---|---|---|---|---|---|---|
| 80 | 90 | 283 | 8,230 | -2.04 | -941 | -16,789 |
| 70 | 80 | 955 | 32,350 | +1.63 | +1557 | 52,730 |
| 60 | 70 | 1259 | 40,125 | 1.12 | 1410 | 44,940 |
| 50 | 60 | 1319 | 38,770 | 1.16 | 1530 | 44,973 |
| 40 | 50 | 1234 | 33,650 | 1.19 | 1468 | 40,043 |
| 30 | 40 | 1107 | 28,500 | 1.05 | 1162 | 29,925 |
| 20 | 30 | 1005 | 24,970 | 1.0 | 1005 | 24,920 |
| 10 | 20 | 920 | 22,305 | 1.0 | 920 | 22,305 |

Table 35
Valve No. V-23-14 (8")

Model Data For Torque Modification: Mitered elbow 8 diameter upstream Geometry 2
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 | 1 | 2742 | 1.0 | 1 | 2742 |
| 70 | 80 | 3 | 4955 | 1.33 | 4 | 6590 |
| 60 | 70 | 3 | 5990 | 1.06 | 3 | 6350 |
| 50 | 60 | 4 | 6150 | 1.0 | 4 | 6150 |
| 40 | 50 | 4 | 5736 | .97 | 4 | 5564 |
| 30 | 40 | 3 | 5062 | .97 | 3 | 4910 |
| 20 | 30 | 3 | 4462 | .98 | 3 | 4373 |
| 10 | 20 | 3 | 4260 | 1.0 | 3 | 4260 |

Table 36
Valve No. V-23-13 (8")

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 | Torque for Straight Flow Maximum | Torque Modification Factor | Torque for Installed Condition Normal | Torque for Installed Condition Maximum |
|---|---|---|---|---|---|---|
| 80 | 90 | 1 | 2742 | 2.21 | 2 | 6060 |
| 70 | 80 | 3 | 4955 | 1.56 | 5 | 7730 |
| 60 | 70 | 3 | 5990 | 1.12 | 6 | 6709 |
| 50 | 60 | 4 | 6150 | 1.05 | 4 | 6458 |
| 40 | 50 | 4 | 5736 | 1.0 | 4 | 5736 |
| 30 | 40 | 3 | 5062 | 1.0 | 3 | 5062 |
| 20 | 30 | 3 | 4462 | 1.0 | 3 | 4462 |
| 10 | 20 | 3 | 4260 | 1.0 | 3 | 4262 |

Table 37
Valve No. V-27-3 (18")

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 | Torque for Straight Flow Maximum | Torque Modification Factor | Torque Installed Condition Normal | Torque Installed Condition Maximum |
|---|---|---|---|---|---|---|
| 80 | 90 | 17 | 14,100 | 1.0 | 17 | 14,100 |
| 70 | 80 | 46 | 44,268 | " | 46 | 44,268 |
| 60 | 7C | 61 | 54,615 | " | 61 | 54.615 |
| 50 | 60 | 65 | 53,487 | " | 65 | 53,487 |
| 40 | 50 | 62 | 47,220 | " | 62 | 47,220 |
| 30 | 40 | 57 | 40,381 | " | 57 | 40,381 |
| 20 | 30 | 52 | 35,395 | " | 52 | 35,395 |
| 10 | 20 | 49 | 32,200 | " | 49 | 32,200 |

Table 38

Valve No. V-27-1, 2, & 4 (18")

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

| Model Test Valve Angle | Actual Valve Angle | Torque for Straight Flow | | Torque Modification Factor | Torque for Installed Condition | |
|---|---|---|---|---|---|---|
| | | Normal | Maximum | | Normal | Maximum |
| 80 | 90 | 17 | 14,100 | .42 | 7 | 5,972 |
| 70 | 80 | 46 | 44,268 | 1.63 | 75 | 72,156 |
| 60 | 70 | 61 | 54,515 | 1.15 | 70 | 62,807 |
| 50 | 60 | 65 | 53,487 | 1.09 | 70 | 58,300 |
| 40 | 50 | 62 | 47,220 | 1.08 | 67 | 50,998 |
| 30 | 40 | 57 | 40,381 | 1.02 | 58 | 41,188 |
| 20 | 30 | 52 | 35,395 | 1.0 | 52 | 35,395 |
| 10 | 20 | 49 | 32,200 | 1.0 | 49 | 32,200 |

Table 39

Valve No. V-28-17 (12")

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

| Model Test Valve Angle | Actual Valve Angle | Torque Straight Flow | | Torque Modification Factor | Torque for Installed Condition | |
|---|---|---|---|---|---|---|
| | | Normal | Maximum | | Normal | Maximum |
| 80 | 90 | 2 | 4,088 | 2.21 | 4 | 9,034 |
| 70 | 80 | 3 | 8,640 | 1.56 | 5 | 13,480 |
| 60 | 70 | 4 | 11,377 | 1.12 | 5 | 11,377 |
| 50 | 60 | 5 | 10,633 | 1.05 | 5 | 11,164 |
| 40 | 50 | 5 | 9,744 | 1.0 | 5 | 9,744 |
| 30 | 40 | 4 | 8,517 | 1.0 | 4 | 8,517 |
| 20 | 30 | 4 | 7,500 | 1.0 | 4 | 7,500 |
| 10 | 20 | 4 | 7,052 | 1.0 | 4 | 7,052 |

Table 40
Valve No. V-28-18 (12")

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

| 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 | 2 | 4,088 | .08 | 0 | 327 |
| 70 | 80 | 3 | 8,640 | 1.07 | 3 | 9,245 |
| 60 | 70 | 4 | 10,518 | 1.07 | 4 | 11,254 |
| 50 | 60 | 5 | 10,633 | 1.0 | 5 | 10,633 |
| 40 | 50 | 5 | 9,744 | 1.0 | 5 | 9,744 |
| 30 | 40 | 4 | 8,517 | 1.0 | 4 | 8,517 |
| 20 | 30 | 4 | 7,500 | 1.0 | 4 | 7,500 |
| 10 | 20 | 4 | 7,052 | 1.0 | 4 | 7,032 |

### 5.3.3 CONCLUSIONS CONCERNING VALVE OPERABILITY

For a LOCA flow condition, it can be seen in Table 32 thru 40 that torques for all valves except V-26-16 & 18 are positive (closing) torques for all disc positions. For these valves, any flow condition from none to the maximum, in combination with the timed bench tests, show the valves will close within 5 seconds or less. For valves V-26-16 & 18 the aerodynamic torque will be negative (tending to hold the disc open) for only the first 5 degrees. The magnitude of this torque will depend on pressure in containment. If the valve is activated to close on initiation of a pressure increase in containment, the valve disc will pass thru this position before pressure and aerodynamic torque rises to a significant level. If the valve were to be activated after a full pressure increase in containment (failure of automatic signal), the aerodynamic torque tending to hold the valve open would be a maximum (Note: Conservative assumptions applied throughout development of torques) of 16,789 in lbs. The actuator output (spring to fail open, air to close) would be greater than 60,000 in lb with a minimum 80 psig air supply. The safety factor even after full containment pressure had developed would be more that 3.5!

For the presented data and supplemental test reports, it has been shown that the valves will operate as designed under the prescribed conditions. This has been shown using the

conservative assumption of no credit taken for pressure ramp in containment and no credit taken for back pressure due to downstream piping. Further, no credit has been taken for activation of the first valve under back pressure conditions produced by closure of the second valve or decrease in upstream pressure 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. During this test rotameters were used to measure any leakage, the smallest flow rate that was detectable was .0001 SCFM, so any value below this was considered to be zero.

Table 41

VALVE SEALING CHARACTERISTICS

| VALVE MARK NO. | VALVE SIZE (IN.) | TEST PRESSURE PSIG | PRESSURIZED SIDE SHAFT SIDE | CLAMP RING SIDE | LEAKAGE (SCFM) |
|---|---|---|---|---|---|
| V-23-13 | 8 | 35 | X | | 0 |
| V-23-13 | 8 | 35 | | X | 0 |
| V-23-14 | 8 | 35 | X | | 0 |
| V-23-14 | 8 | 35 | | X | 0 |
| V-23-15 | 8 | 35 | X | | 0 |
| V-23-15 | 8 | 35 | | X | 0 |
| V-23-16 | 8 | 35 | X | | 0 |
| V-23-16 | 8 | 35 | | X | 0 |
| V-28-17 | 12 | 35 | X | | 0 |
| V-28-17 | 12 | 35 | | X | 0 |
| V-28-18 | 12 | 35 | X | | 0 |
| V-28-18 | 12 | 35 | | X | 0 |
| V-27-1 | 18 | 35 | X | | 0 |
| V-27-1 | 18 | 35 | | X | 0 |
| V-27-2 | 18 | 35 | X | | 0 |
| V-27-2 | 18 | 35 | | X | 0 |
| V-27-3 | 18 | 35 | X | | 0 |
| V-27-3 | 18 | 35 | | X | 0 |
| V-27-4 | 18 | 35 | X | | 0 |
| V-27-4 | 18 | 35 | | X | 0 |
| V-26-16 | 20 | 35 | X | | 0 |
| V-26-16 | 20 | 35 | | X | 0 |
| V-26-18 | 20 | 35 | X | | 0 |
| V-26-18 | 20 | 35 | | X | .025 |

## 6.2 LONG TERM SEALING

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

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

## 6.3 DEBRIS EFFFCTS ON SFALING

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: Wyle Laboratories
Scientific Services and Systems Group
Huntsville, Alabama

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

1. Report WR 81-53 for Clow 8" Wafer Stop Valve (Jan. 82)
   Mark Nos. V-23-13, -14, -15, -16
   Clow Job No. 80-8170-01, -02

2. Report WR 81-54 for Clow 12" Wafer Stop Valve (Dec. 81)
   Mark Nos. V-28-17, -18
   Clow Job No. 80-8170-03

3. Report WR 81-55 for Clow 18" Wafer Stop Valve (Jan. 82)
   Mark Nos. V-27-1, 2, 3, 4
   Clow Job No. 80-8170-04

4. Report WR 81-56 for Clow 20" Wafer Stop Valve (Jan. 82)
   Mark Nos. V-26-16, -18
   Clow Job No. 80-8170-05

B. Seismic Qualification Test Reports

prepared by: Wyle Laboratories
Scientific Services and System Group
Huntsville, Alabama

1. 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.

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

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

7.0 REFERENCES (con't)

3. 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.

2. Report on"Aerodynamic Torque and Mass Flowrate For Compressible Flow Through Three Geometrically Similar Scale-Model Clow Valves Located Downstream of a 90$^{o}$ Mitered Elbow" (Jan. 82).

3. Vought 12" Valve Flow Test (See B-3 above)

D. Other Report and Information

1. Operating Instructions for Clow Tricentric Wafer Stop Valve covers installation, maintenance, and operating instructions for 80-8170 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. Jersey Central Power & Light Company Procurement Specification No. 492-7, Revision 3, dated 7/29/81.

2. ASME Code Section III, Division 1, subsection NC and Appendices I and XIV.

7.0 REFERENCES (con't)

3. "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.

# 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.e., quality, redundancy, reliability and other appropriate criteria.

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

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

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

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

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

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

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

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

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

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

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

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

GUIDELINES FOR DEMONSTRATION OF OPERABILITY OF PURGE AND VENT VALVES

OPERABILITY

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

1. Valve closure rate versus time - i.e., constant rate or other.
2. Flow direction through valve; $\Delta P$ 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 upstream and downstream and valve orientation are simulated.
3. Whether the following load and environmental factors were considered

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

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

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

In-Situ Testing

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

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

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

End CSB 6-4

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

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

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

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

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

5. Provide the necessary information for the table shown below for valve positions from the initial open position to the seated position ($10^{0}$ increments if practical).

| Valve Position (in degrees - $90^{0}$ = full open) | Predicted $\Delta P$ (across valve) | Maximum $\Delta P$ (capability) |
|---|---|---|

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

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

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

10. Where air operated valve assemblies use accumulators as the fail-safe feature, describe the accumulator air system configuration and its operation. Provide necessary information to show the adequacy 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 cperation 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)

DESCRIPTION OF OPERATIONAL TESTS

OF A 12 INCH CLOW TRICENTRIC VALVE

FOR

NUCLEAR PURGE SYSTEM SERVICE

BY

J.E. KRUEGER
NUCLEAR VALVE DESIGN ENGINEER

NOVEMBER 30, 1981

INTRODUCTION -

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

OBJECTIVE -

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

TEST SET-UP -

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

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

INSTRUMENTATION -

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

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

VALVE AND ACTUATOR DESIGN PARAMETERS -

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

CONDUCT OF TEST -

The test consisted of applying the static loads to the actuator and establishing a 65 psig upstream pressure with the flow 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

ASME SECTION III

DIVISION 1 EXCERPTS

(FROM ASME APPENDIX 1)

FOR

BOLT FATIGUE

ANALYSIS

# TABLE I-9.1
## TABULATED VALUES OF $S_a$, ksi, FROM FIGS. I-9.0[1,2]

| Figure | Curve | 1E1 | 2E1 | 5E1 | 1E2 | 2E2 | 5E2 | 8.5E2 [Note (4)] | 1E3 | 2E3 | 5E3 | 1E4 | 1.2E4 [Note (4)] | 2E4 | 5E4 | 1E5 | 2E5 | 5E5 | 1E6 |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | Number of Cycles [Note (3)] | | | | | | | | | | | | | | | | | |
| I-9.1 | UTS 115–130 ksi | 420 | 320 | 230 | 175 | 135 | 100 | ... | 78 | 62 | 49 | 44 | 43 | 36 | 29 | 26 | 24 | 22 | 20 |
| I-9.1 | UTS ≤ 80 ksi | 580 | 410 | 275 | 205 | 155 | 105 | ... | 83 | 64 | 48 | 38 | ... | 31 | 23 | 20 | 16.5 | 13.5 | 12.5 |
| I-9.2 | ... | 650 | 470 | 317 | 240 | 185 | 136 | ... | 109 | 89 | 70 | 59 | ... | 51 | 42.5 | 37.5 | 33 | 28.5 | 26 |
| I-9.3 | $S_y$ = 18.0 ksi | 260 | 190 | 125 | 95 | 73 | 52 | ... | 44 | 36 | 28.5 | 24.5 | ... | 21 | 17 | 15 | 13.5 | 12.5 | 12.0 |
| I-9.3 | $S_y$ = 30.0 ksi | 260 | 190 | 125 | 95 | 73 | 52 | ... | 44 | 36 | 28.5 | 24.5 | ... | 19.5 | 15 | 13 | 11.5 | 9.5 | 9.0 |
| I-9.3 | $S_y$ = 45.0 ksi | 260 | 190 | 125 | 95 | 73 | 52 | 46 | 39 | 24.5 | 15.5 | 12 | ... | 9.6 | 7.7 | 6.7 | 6.0 | 5.2 | 5.0 |
| I-9.4 | MNS ≤ 2.7$S_m$ [Note (5)] | 1150 | 760 | 450 | 320 | 225 | 143 | ... | 100 | 71 | 45 | 34 | ... | 27 | 22 | 19 | 17 | 15 | 13.5 |
| I-9.4 | MNS = 3$S_m$ [Note (5)] | 1150 | 760 | 450 | 300 | 205 | 122 | ... | 81 | 55 | 33 | 22.5 | ... | 15 | 10.5 | 8.4 | 7.1 | 6 | 5.4 |

NOTES:

(1) All notes on the referenced figures apply to these data.

(2) Interpolation between tabular values is permissible based upon data representation by straight lines on a log–log plot. Accordingly, for $S_i > S > S_j$

$$N/N_i = (N_j/N_i)^{[\log(S_i/S)/\log(S_i/S_j)]}$$

where $S$, $S_i$, and $S_j$ are values of $S_a$; $N$, $N_i$, and $N_j$ are corresponding numbers of cycles from design fatigue data.

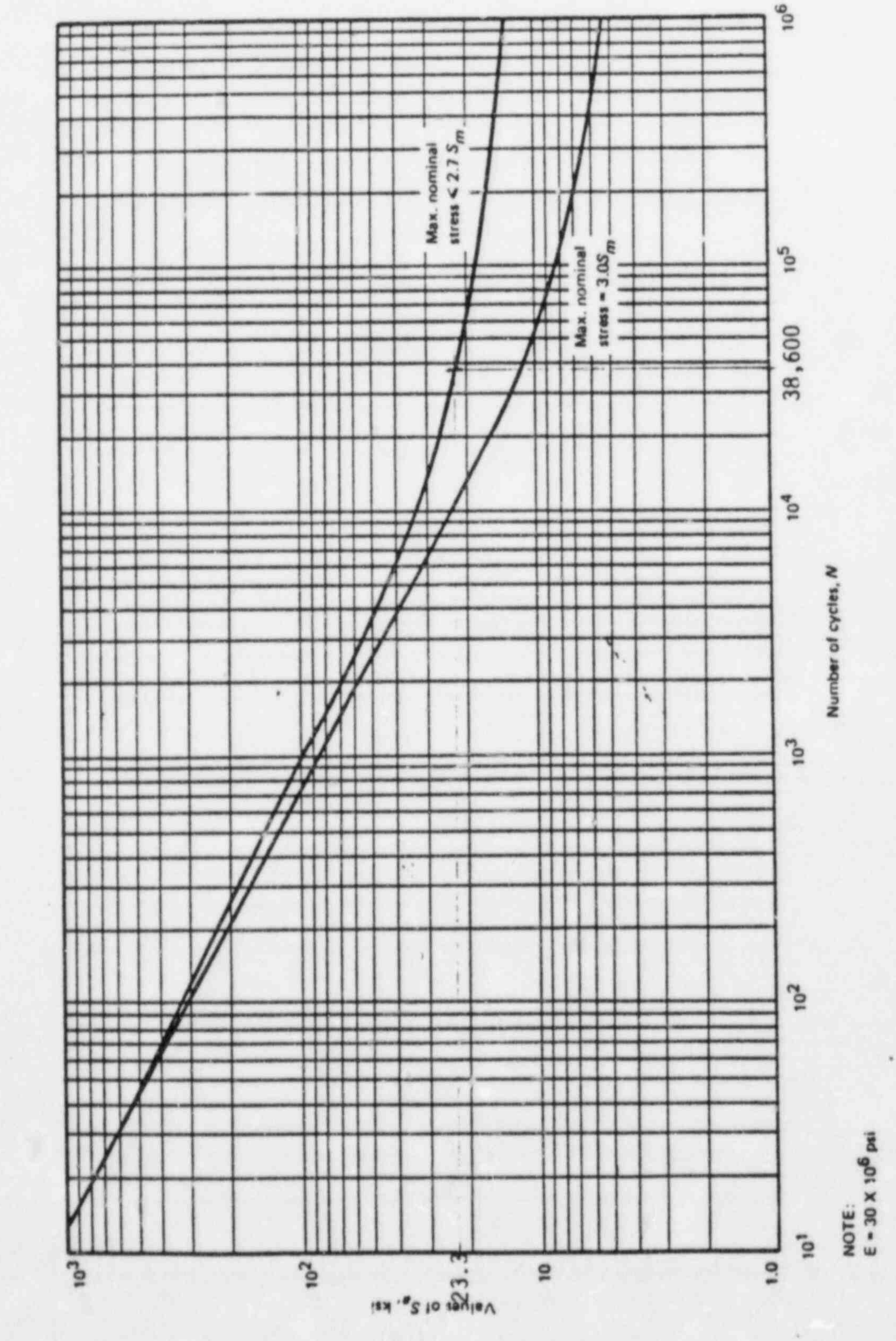
Example: From the data given in the Table above, use the interpolation formula above to find the number of cycles $N$ for $S_a$ = 53.5 ksi when UTS ≤ 80 ksi in Fig. I-9.1:

$$N/2000 = (5000/2000)^{[\log(64/53.5)/\log(64/48)]} \therefore N = 3450 \text{ cycles}$$

(3) The number of cycles indicated shall be read as follows: IEJ = 1 × $10^J$, e.g., 5E2 = 5 × $10^2$ or 500.

(4) These data points are included to provide accurate representation of curves at branches or cusps.

(5) MNS is the Maximum Nominal Stress.




NOTE:
$E = 30 \times 10^6$ psi

FIG. I-9.4 DESIGN FATIGUE CURVE FOR HIGH STRENGTH STEEL BOLTING FOR TEMPERATURES NOT EXCEEDING 700°F
Table I-9.1 Contains Tabulated Values and a Formula for Accurate Interpolation of These Curves