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MIDDLE SOUTH UTILITIES SYSTEM

D. L. ASWELL
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W3P81-1835
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Mr. R. L. Tedesco
Assistant Director of Licensing
U. S. Nuclear Regulatory Commission
Washington, D. C. 20555

SUBJECT: Waterford 3 SES
TMI Item II.E.4.2 - Containment
Purge Valve Operability Study

Dear Mr. Tedesco:

In accordance with TMI Item II.E.4.2, a containment purge and vent valve operability study was performed for Waterford 3. In accordance with the results of the study (attached), modifications are being made to limit the valve opening to 40 degrees. We believe that the attached study demonstrates the compliance of Waterford to Item II.E.4.2, Part 5.

Yours very truly,

DL Aswell

D. L. Aswell

DLA/RMF/ddc

Attachment

cc: E. L. Blake, W. M. Stevenson, S. Black

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1.0 Introduction

Item II.E.4.2 of NUREG-0737 delineated the NRC staff position on ensuring the operability of Containment Purge Isolation valves. Pursuant to paragraph 2 and of Attachment 1 to Item II.E.4.2, the purge valve vendor (Fisher Controls) was requested to perform a sensitivity analysis to determine operability limits of the valves. Reference 1 and its subsequent clarifications was used to identify related concerns which could have adverse impact on valve operability. These concerns are discussed below.

2.0 Response to Reference 1 (Questions are those of the clarification)

1. <u>Question</u>

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

<u>Response</u>

The analyzed containment condition was a constant [illegible]4 psig and 300°F in determining the fluid conditions across the valve at all angles of rotation. (A constant 300°F was used in order to account for the long thermal time constant of the valve assembly rather than the brief peak temperature of 414°F which occurs after the valve is closed).

ΔP across the valve was considered equal to peak containment pressure (psig). Material properties were selected (during stress calculations) at peak containment temperatures. The effect of compressible flow in sizing Fisher butterfly valves is best explained by the following:

AIR VS WATER SERVICE

Whenever a Fisher butterfly valve is in a gas flow application the effects due to compressible flow are taken into consideration while determining the dynamic torque effects for each individual valve selection. This consideration is built into our valve selection procedures and requires a conscious liquid or gas decision in calculating the effective pressure drop of which the dynamic torque is a function.

Fisher's philosophy concerning the effects of compressible flow on butterfly valves is presented in ISA Transactions, Vol 8, No 4, entitled "Effect of Fluid Compressibility on Torque in Butterfly Valves", written by Floyd P Harthun (Manager, Product Evaluation, Fisher Controls Co). A copy of this transaction is included as Attachment 2 to this letter.

2. Question

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 (especia.ly larger valves) than the sizes known to have undergone flow tests?

Response

In determining allowable pressure drops across a particular butterfly valve at various angles of the disc, Fisher Controls uses classical "mechanics of materials" type equations to calculate stress levels at various worst-case locations in the valve assembly (specifically, various locations along the valve shaft). The approach to the analysis, the equations used, and the combination of the calculated stresses all make up a portion of Fisher's design philosophy for butterfly valves. This analysis approach addresses all of the different states of shear and tensile stress which are applicable to the loading conditions defined.

Establishing the loads that actually exist makes up the remaining portion of our design philosophy for butterfly valves. These loads range from easily calculated loads, such as bending due to pressure differential across the disc, to loads such as packing and dynamic torques which require a certain amount of testing combined with scaling in order to analyze all valve sizes. It is the factor of dynamic torque that produces different stresses at different disc rotations and disc geometries. Through testing and scaling Fisher has produced dynamic torque factors for incremental disc rotations for plate discs (which is the configuration for the subject valves).

The model tests used to establish the dynamic torque values used in sizing were conducted using 4" and 6" test valves with various aspect ratios ranging from 2:1 to 14:1 (such as 3:1, 4:1, 5:1, 8:1, 11:1 and 14:1). The dimensionless aspect ratio (defined as the ratio of the disc diameter to the hub diameter) was judged to be a significant parameter for evaluation of dynamic torques at various open angles.

Capacity and Torque curves obtained from a typical test are enclosed (Attachment 3) to illustrate the method and general shape of the curves for type 9200 butterfly valves.

3. Question

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?

Response

All Fisher sizing data is based on dynamic torque determination tests which were performed with uniform flow profiles and on valve discs with representative geometries. Upstream of the Waterford purge valves is only a straight run of duct with no elbows, T-connections, etc. Flow through the valve is expected to be uniform.

4. Question

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. constant or other) taken into account? For air operated valves equipped with spring return operators, has the lag time from the time the 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 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 55 psig, thus the lag time.

Response

When calculations were performed to determine the maximum allowable open angle (provided in Attachment 1) the assumption was made that the valve had to close against peak containment pressure. Since this (conservative) approach was taken, a time history study was not made, and therefore, valve closure rates and response lags did not have to be considered.

5. Question

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

Valve Position Min Degress-90° (full open)	Predicted ΔP (across valve)	Maximum ΔP (capability)

Response

In determining allowable ΔP vs angle of opening for the subject valves, there are several considerations to take into account.

1) Allowable ΔP based on the strength of the valve.
2) Allowable ΔP based on available torque from actuator.
3) Allowable ΔP based on the strength of the actuator.

The allowable ΔP based on valve strength is determined by a Fisher computer program. The computer program can be described as follows.

For a given valve at some angle of opening, the program begins by calculating the loading. This includes a hydrostatic load on the disc, seating torque, bushing and packing torque and dynamic torque.

After the loading is determined, the program calculates stresses in the shaft, key, pin and bushing for a specific ΔP and compares these stresses to a material strength. This strength is based on 1.5 x "S". "S" is the allowable stress figure found in Section VIII of the ASME Boiler and Pressure Vessel Code. S is equal to 1/4 of the minimum tensile strength or 2/3 of the minimum yield strength, whichever is less. For shear stresses 0.75 S is used.

The program calculates stress and changes ΔP iteratively until the allowable strength matches the stress. This determines the maximum allowable pressure drop for that angle of opening based on the stress at a single point. Therefore, this process is done for cases 1, 2, 3, 4 and 5 (as defined below) for each angle of opening.

Case 1 - stress in the shaft at the disc hub due to bending and torsion

Case 2 - stress in the shaft at the disc hub due to torsion and transverse shear

Case 3 - stress at the pinned disc-shaft connection

Case 4 - stress at the keyed actuator-shaft connection

Case 5 - stress at the shaft bushing

The program output shows a ΔP which is calculated at each point for each angle of opening, including two ΔP for case 1 (one based on maximum shear stress, one based on maximum tensile stress) for a total of 6 ΔP's. The smallest ΔP of these 6 is then repeated as allowable ΔP at the bottom of the column. The actuator torque for the lowest ΔP (allowable ΔP) is also listed. Above 40° open, the allowable ΔP's (based on valve strength) drop below the accident criterion of 44 psig, as shown in Attachment 1.

The required actuator torque vs angle of opening for a 44 psig drop (P_1=44 psig, P_2=ambient) is shown in Attachment 1. The torque at 0° is the torque required to close the valve. This shutoff torque is 41,795 in-lb_f. The minimum end-of-stroke torque output for the Bettis actuator is 48,500 in-lb_f. Therefore the Bettis provides adequate torque for shutoff against a 44 psig pressure drop.

The specified required torque at open angles less than 60° is torque required to open the valve further. Dynamic torques at open positions up through 60° tend to close the valve. Therefore no consideration of available actuator torque at open angle is needed. However the actuator strength must be considered for the case in which the valve is held in the open position. If the valve is held at 40° or less the maximum torque the actuator would be called on to supply is 46,716 in-lb (See Attachment 1). This torque is within the output rating of the actuator and would not cause damage to the actuator.

As stated in the response to request #4, no time-history study has been performed and therefore no "Predicted P" values are provided; the remaining information requested is presented in Attachment 1 to this document.

6. Question

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?

Response

The subject 48" butterfly valves were designed according to the ASME Boiler and Pressure Vessel Code, Sections III and VIII. Allowable stresses were also taken from the ASME B & PV Code. Loads considered in the design of these valves includes all typical pressure and flow induced loads. Worst case load combinations are used. Pressure and temperature ratings for these valves can be found in Fisher bulletin 51.4:9200 (Attachment #5).

NOTE: No requests numbered 7 and 8 were included in Reference 1.

9. Question

For these 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.

Response

The subject 48" valves are equipped with Bettis spring-return actuators. This actuator design includes a vent to ambient on the spring side of the piston; therefore, if the pressure side of the piston is vented (through the solenoid) to the same ambient as the spring side, no pressure differential will exist across the piston as a result of the accident containment pressure.

10. Question

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

Response

This request is not applicable to Waterford Unit #3, since none of the subject valves are equipped with accumulators as a fail-safe feature.

11. Question

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

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

Response

This request is not applicable to Waterford Unit #3, since none of the subject valves are equipped with inflatable seal rings.

12. Question

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.

Response

The valve actuators will be provided with internal piston travel stops to limit the actuator stroke to a maximum of 40° valve open position. This modification will not adversely affect the valve closure or seating performance.

13. Question

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

Response

The response to Request #12 and #5 above, in conjunction with the information provided in Attachment 1 adequately address this subject.

14. Question

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

Response

This request is not applicable to Waterford Unit #3, since none of the subject valves are equipped with torque limiting devices.

15. Question

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 voltage available. Does this reduced voltage operation result in any significant change in stroke timing? Describe the emergency mode power source used.

Response

This request is not applicable to Waterford Unit #3, since none of the subject valves are equipped with electric motor operators.

16. Question

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?

Response

This request is not applicable; see Request #15, above.

17. Question

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

Response

No analyses or tests were done to environmentally qualify the subject valves.

1) Pressure-Temperature - The temperature pressure environmental conditions of 44 psig and 300°F (with a 414°F spike) fall within the design rating of the valve.

   In accordance with Fishers recommendation, all elastomeric parts will be replaced every 4 years.

2) Aging-Radiation - The subject valves have EPPM seats. Seat material degradation cannot be accurately predicted in terms of seat leakage. However, between the in-containment and the in-annulus valve is a 1" line leading into the annulus. This line will draw any leakage through the valve into the annulus to be eventually exhausted by the ESF grade Shield Building Ventilation System. Additionally, in accordance with Fisher's recommendation, all elastomeric parts will be replaced every 4 years.

3) Seismic - The seismic testing done to qualify the subject valves is described in Attachment , "Dynamic Test Program on Bettis T-420-SR1-M3".

4) Wind loading, missiles generated by tornadoes and explosion conditions qualification is not applicable for these valves.

5) Qualification of solenoids and limit switches will be addressed by the NUREG-0588 response presently scheduled for submittal in October 1981.

19. Question

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 chemical) should be pointed out.

Response

See Response 17 and Attachment #6 for details of the dynamic testing performed on the subject valves.

20. Question

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

Response

This request is not applicable to Waterford Unit #3, since an analysis was not used to qualify the subject valves.

21. Question

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

Response

The maintenance instruction have been reviewed and will be followed.

References

1. Letter from D G Eisenhut to All Light Water Reactors dated September 27, 1979 , and Clarifications

Attachment 1 - Allowable ΔP vs. Angle of Opening
Required Torque vs. Angle of Opening

Attachment 1

OPENING ANGLE (Degrees)	ALLOWABLE ΔP (PSIG)	REQUIRED ** CLOSING TORQUE (in-lb)	MIN. END-OF-STROKE ACTUATOR OUTPUT (in-lb)	MAXIMUM ** REQUIRED TORQUE (in-lb)	MAX. ACTUATOR RATING (in-lb)
0	65	41,795	48,500	41,795	55,500
10	65	10,203	48,500	32,389	55,500
20	62	*	48,500	46,716	55,500
30	62	*	48,500	46,716	55,500
40	62	*	48,500	46,716	55,500
50	13	*	48,500	63,779	55,500
60	6	*	48,500	93,164	55,500
70	2	*	48,500	143,390	55,500
80	1.6	*	48,500	149,072	55,500
90	1.6	*	48,500	149,072	55,500

*Dynamic torque which tends to close the valve exceeds the frictional torque in these cases.

**All required torques are based on a 44 psig drop with P_1 = 44 psig and P_2 - ambient.

ALLOWABLE PRESSURE DROP CALCULATION

WATERFORD STEAM GENERATING STATION UNIT 3, LOUISIANA POWER AND LIGHT CO.
48 INCH 9220 BUTTERFLY VALVES ADJUSTABLE T-RINGS
CON 40230-27 THRU -32 SN 8F226076-081 TAG 2HV-B150 THRU 2HV-B155
PLATE DISC:SA516 GR70 STUB SHAFT:SA564 GR630 BUSHING:GRAPHITE-BRONZE (#2)
CLASS 2 SHAFT (2.5 INCH DIAM.) T=300 F DP=44 PSIG FLUID:AIR
LEE WAITE JUNE 8, 1981

INPUT DATA

ANG	0.0	20.000	30.000	40.000	50.000	60.000	70.000	80.000	90.000	0.0
DDISC	45.394	45.394	45.394	45.394	45.394	45.394	45.394	45.394	45.394	0.0
DSHFT	2.497	2.497	2.497	2.497	2.497	2.497	2.497	2.497	2.497	0.0
CLO	0.746	0.746	0.746	0.746	0.746	0.746	0.746	0.746	0.746	0.0
TS	20449.000	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
TI	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
[illegible]	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0	0.0
[illegible]	0.0	1240.000	1240.000	1240.000	2890.000	6780.000	18900.000	24200.000	24200.000	0.0
DPIN	44.000	44.000	44.000	44.000	44.000	44.000	44.000	44.000	44.000	0.0
DELTPF	1.000	0.350	0.350	0.350	0.250	0.180	0.110	0.090	0.090	0.0
STSH	100.000	100.000	100.000	100.000	100.000	100.000	100.000	100.000	100.000	0.0
S-BSH	85.000	85.000	85.000	85.000	85.000	85.000	85.000	85.000	85.000	0.0
C1	0.500	0.500	0.500	0.500	0.500	0.500	0.500	0.500	0.500	0.0
C2	0.750	0.750	0.750	0.750	0.750	0.750	0.750	0.750	0.750	0.0
ISHTYP	2	2	2	2	2	2	2	2	2	0

GENERATED VARIABLES

ST	52500.00	52500.00	52500.00	52500.00	52500.00	52500.00	52500.00	52500.00	52500.00	0.0
SS	26250.00	26250.00	26250.00	26250.00	26250.00	26250.00	26250.00	26250.00	26250.00	0.0
S[illegible]	8500.00	8500.00	8500.00	8500.00	8500.00	8500.00	8500.00	8500.00	8500.00	0.0

OUTPUT

DELTP(1)	122.0409	119.8160	119.8160	119.8160	110.8480	90.6952	43.9011	37.6369	37.6369	0.0
DELTP(2)	111.6922	107.1012	107.1012	107.1012	87.4571	30.4996	4.1827	3.2748	3.2748	0.0
DELTP(3)	67.8788	62.1952	62.1952	62.1952	42.8551	10.4415	4.0422	3.1878	3.1878	0.0
DELTP(4)	95.9576	71.2831	71.2831	71.2831	12.8905	5.7083	2.0864	1.6332	1.6332	0.0
DELTP(5)	96.8846	84.5788	84.5788	84.5788	42.7046	8.3071	3.0916	2.4256	2.4256	0.0
DELTP(6)	65.4436	65.4436	65.4436	65.4436	65.4436	65.4436	65.4436	65.4436	65.4436	0.0
ALLW.DP	65.4436	62.1952	62.1952	62.1952	12.8905	5.7083	2.0864	1.6332	1.6332	0.0
ACT.TORQ	47360.5508	51023.6328	51023.6328	51023.6328	42861.2383	41405.2305	40670.9375	40579.0937	40579.0937	0.0

This actuator torque is the maximum torque associated with the maximum allowable pressure drop.

Attachment 2 - ISA Transactions, Vol. 8,
No. 4; "Effects of Fluid
Compressibility on Torque
in Butterfly Valves".

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Effect of Fluid Compressibility on Torque in Butterfly Valves

FLOYD P. HARTHUN

ISA TRANSACTIONS 8 (281-286) 1969

Effect of Fluid Compressibility on Torque in Butterfly Valves*

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▶ A technique is presented by which the shaft torque resulting from fluid flow through butterfly valves can be determined with reasonable accuracy for both compressible and incompressible flow. First, the general torque relationship for incompressible flow is established. Then, an effective pressure differential is defined to extend this relationship to include the effect of fluid compressibility. The application of this technique showed very good agreement with experimental test results.

INTRODUCTION

THE APPLICATION of butterfly valves in various automatic control systems requires proper actuator sizing for efficient control. Thus, a thorough knowledge of the fluid reaction forces acting on the valve disc is required. Extensive experimental work[1] has been performed in the past to establish a relationship to determine these forces and thus determine the resultant shaft torque. The general form of this relationship has been established and confirmed. However, by using the classical fluid momentum approach, a similar relationship can be obtained in which the torque is shown to be directly proportional to the measured valve pressure differential for a given disc position. This relationship along with most of the previously published torque information is adequate for incompressible flow. Although the effect of fluid compressibility on torque has been recognized, no useful relationship has been developed. The primary objective of this investigation is to extend the established torque relationship to include the effect of fluid compressibility.

*Presented at the 1968 ISA Annual Conference; revised August, 1969.
†Research Engineer.

DEVELOPMENT OF GENERAL TORQUE RELATIONSHIP

The total shaft torque required to operate butterfly valves can be separated into two major components:

1. Dynamic torque—that portion of the total operating torque attributable to the fluid reaction force of the flowing medium acting on the valve disc.
2. Friction torque—that portion of the total operating torque attributable to friction in the packing and bushings.

Since each of these components is independent of the other, a separate evaluation of each component affords the best approach to this problem. This investigation is limited to an evaluation of the dynamic torque component. If the friction on the valve shaft is assumed to be independent of direction of rotation, it can be readily isolated. The torque required to rotate the valve disc is measured in a clockwise and a counterclockwise direction through full travel. Since friction always opposes motion the difference between these values will be twice the actual shaft friction.

The dynamic torque for butterfly valves is a function of the fluid reaction forces acting on the valve disc. It would be difficult to determine these forces by purely analytical techniques. Experimental determination of the pressures and velocity profiles in the immediate area of the disc would also be quite difficult. However, if a control volume is selected so the boundaries are points of known pressure and velocity, an analysis of these forces can be made from the change in fluid momentum through this control volume.

INCOMPRESSIBLE FLOW

An expression for dynamic torque is developed assuming incompressible flow. This torque is a function of the fluid reaction force, F, and a moment arm, D, which is a characteristic dimension of the valve disc.

$$T_D = f(F, D) \tag{1}$$

Using the fluid momentum approach, the force, F, is given by:

$$F = M\Delta V \tag{2}$$

where

F = sum of external forces acting on fluid
M = mass flow rate
ΔV = fluid velocity change through the control volume

The mass flow rate, M, is given by

$$M = \rho A V \tag{3}$$

By using a proportionality constant, B_1, the mass flow rate can also be defined as

$$M = B_1 A(\rho \Delta P)^{1/2} \tag{4}$$

Equations (3) and (4) are combined to obtain the following expression for fluid velocity:

$$V = B_1(\Delta P/\rho)^{1/2} \tag{5}$$

The velocity change through the control volume, ΔV, in Equation (2) can be expressed in terms of the velocity at the valve disc by use of a proportionality constant, B_2

$$F = B_2 M V \tag{6}$$

By substituting the expressions for mass flow rate Equation (4) and fluid velocity Equation (5) into Equation (6) the force on the valve disc is

$$F = B_1{}^2 B_2 A \Delta P \tag{7}$$

For a given valve size, the flow area, A, for any angle of disc rotation, θ, can be written as

$$A = B_\theta \frac{\pi D^2}{4} \tag{8}$$

The force, F, acts upon a moment arm which is a function of the disc diameter, D. Now, the dynamic torque can be written as

$$T_D = B_3 F D \tag{9}$$

Combining Equations (7), (8), and (9)

$$T_D = \frac{B_1{}^2 B_2 B_3 B_\theta \pi D^3 \Delta P}{4} \tag{10}$$

or

$$T_D = K_1 D^3 \Delta P \tag{10-A}$$

where

$$K_1 = \frac{B_1{}^2 B_2 B_3 B_\theta \pi}{4} = \frac{T_D}{D^3 \Delta P} \tag{10-B}$$

Equation (10-B) is defined as the dimensionless torque coefficient which can be determined experimentally from tests conducted with incompressible flow.

COMPRESSIBLE FLOW

The dynamic torque for butterfly valves is proportional to the mass flow rate and velocity change through a selected control volume for both compressible and incompressible flow (i.e., $T_D \propto M\Delta V$). Therefore, the approach used to obtain an expression for this torque assuming incompressible flow can be extended to compressible flow by re-defining these two variables.

First, assume that the velocity at the valve disc, V_d, is proportional to the velocity change through the control volume. Then, the dynamic torque can be expressed as

$$T_D \propto M V_d \tag{11}$$

The velocity at the valve disc is given by

$$V_d = \frac{M}{\rho_d A} \tag{12}$$

By combining Equations (11) and (12) the dynamic torque is shown to vary directly as the square of the mass flow rate and inversely with the fluid density at the valve disc.

$$T_D \propto \frac{M^2}{\rho_d} \tag{13}$$

Determining the flow rate of a compressible fluid through a control valve by analytical techniques is quite difficult because of valve geometry. The major problem is to establish the pressure differential between the valve inlet and the vena contracta. However, by defining the physical system in which the valve is installed to conform with specifications given by the Fluid Controls Institute (FCI),[2] empirical relationships developed specifically for determining flow rate for control valves can be considered. Several such empirical relationships have been developed; however, only one, the Universal Gas Sizing Equation,[3] has been shown to accurately define the flow rate for any valve configuration. This equation is given by

$$Q = \sqrt{\frac{520}{GT}} P_1 C_1 C_2 C_v \sin\left[\frac{59.64}{C_1 C_2}\sqrt{\frac{\Delta P}{P_1}}\right]_{rad} \tag{14}$$

Equation (14) can be rewritten to obtain an equivalent expression for mass flow rate.

$$M = 1.06\sqrt{\rho_1 P_1}\, C_1 C_2 C_v \sin\left[\frac{59.64}{C_1 C_2}\sqrt{\frac{\Delta P}{P_1}}\right]_{rad} \qquad (15)$$

The sine function in Equations (14) and (15) is used to define the transition between incompressible flow occurring at low pressure ratios ($\Delta P/P_1$) and critical flow.

Let

$$\theta = \left[\frac{59.64}{C_1 C_2}\sqrt{\frac{\Delta P}{P_1}}\right]_{rad} \qquad (16)$$

Rewriting Equation (15) in the following manner:

$$M = 1.06\sqrt{\rho_1 P_1}\, C_1 C_2 C_v F \qquad (17)$$

The factor, F, is bounded by the following:

$$\begin{aligned} F &= \sin\theta \qquad \text{for } \theta < \pi/2 \\ F &= 1.0 \qquad \text{for } \theta \geq \pi/2 \end{aligned} \qquad (18)$$

By substituting Equation (17) for the mass flow rate in Equation (13), the dynamic torque for a given valve is given by

$$T_D \propto \frac{\rho_1 P_1 (C_1 C_2 \sin\theta)^2}{\rho_d} \qquad (19)$$

The only parameter in Equation (10) that cannot be readily obtained is the density at the valve disc, ρ_d. Assuming that the change in the ratio of fluid density at the valve inlet to fluid density at the valve disc with increasing pressure ratio is small relative to the total change in mass flow rate, the torque expression can be simplified in the following manner:

$$T_D \propto P_1 (C_1 C_2 \sin\theta)^2 \qquad (20)$$

Therefore, for compressible flow:

$$T_D = K_2 P_1 (C_1 C_2 \sin\theta)^2 \qquad (21)$$

For small values of pressure ratio ($\Delta P/P_1$) Equation (21) reduces to the incompressible torque relationship given by Equation (10-A).

As

$$\Delta P/P_1 \rightarrow 0$$

$$\sin\theta = \theta \text{ (radians)}$$

$$T_D = K_2 (59.64)^2 \Delta P \qquad (22)$$

The expression in Equation (22) is equivalent to the expression in Equation (10-A):

$$K_2 (59.64)^2 \Delta P = K_1 D^3 \Delta P$$

$$K_2 = \frac{K_1 D^3}{(59.64)^2} \qquad (23)$$

By substituting the expression in Equation (23) for the coefficient K_2 in Equation (21), a general expression for dynamic torque for compressible flow is obtained using the dimensionless torque coefficient established for incompressible flow.

$$T_D = K_1 D^3 P_1 \left[\frac{C_1 C_2}{59.64}\right]^2 \sin^2\theta \qquad (24)$$

For convenience the form of Equation (24) is simplified.

$$T_D = K_1 D^3 \Delta P_e \qquad (25)$$

where

$$\Delta P_e = P_1 \left[\frac{C_1 C_2}{59.64}\right]^2 \sin^2\theta \qquad (26)$$

Equation (26) is defined as the pressure differential contributing to the dynamic torque on butterfly valves with conditions of compressible flow.

EXPERIMENTAL RESULTS

The first step in the experimental evaluation was to establish the dimensionless torque coefficient, K_1, as a function of valve disc rotation as defined by Equation (10-B). A test was conducted on a 4-in. valve under the following controlled conditions:

1. The valve was installed in a 4-in. test line with a minimum of 12 pipe diameters of straight pipe upstream.
2. The pressure taps were located according to FCI specifications and attached to the test line according to specifications in the *ASME Power and Test Code*.[4]
3. Water at ambient temperature was used as the flowing medium.
4. The inlet pressure and outlet pressure were held constant.
5. The test was conducted at a low pressure ratio ($\Delta P/P_1 = 0.088$) to ensure incompressible flow.

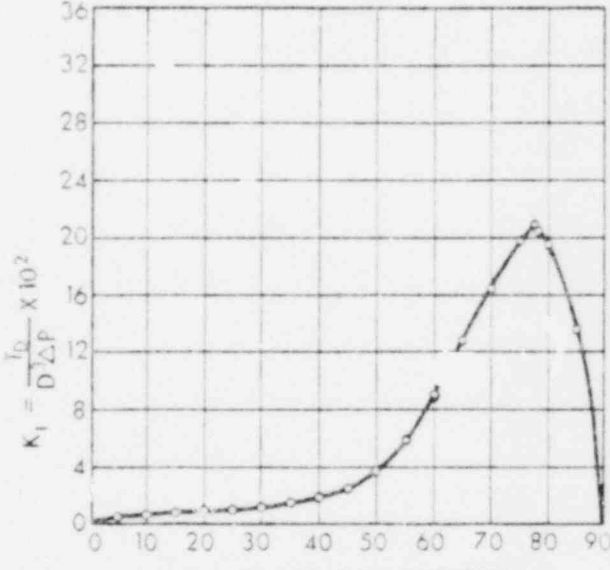

Figure 1. Dimensionless torque coefficient, 4-in. butterfly valve incompressible flow: $P_1 = 100$ psig, $P = 10$ psi.

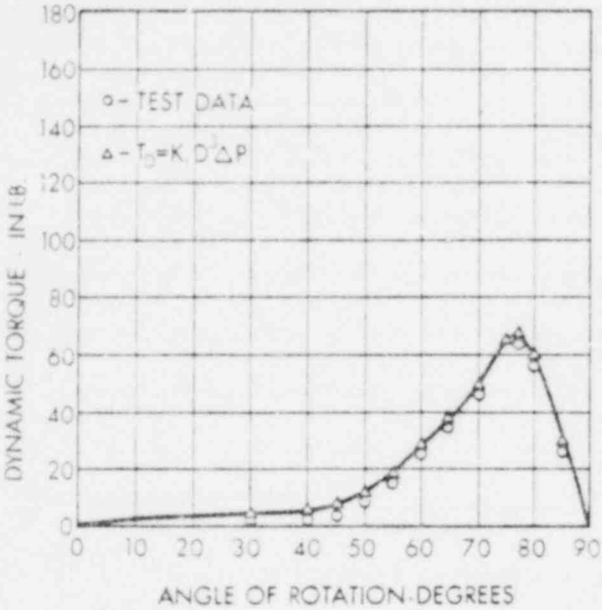

Figure 2. Dynamic torque vs. angle of disc rotation, 4-in. butterfly valve, comparison of experimental results with calculated torque, incompressible flow: P_1 = 100 psig, ΔP = 5 psi.

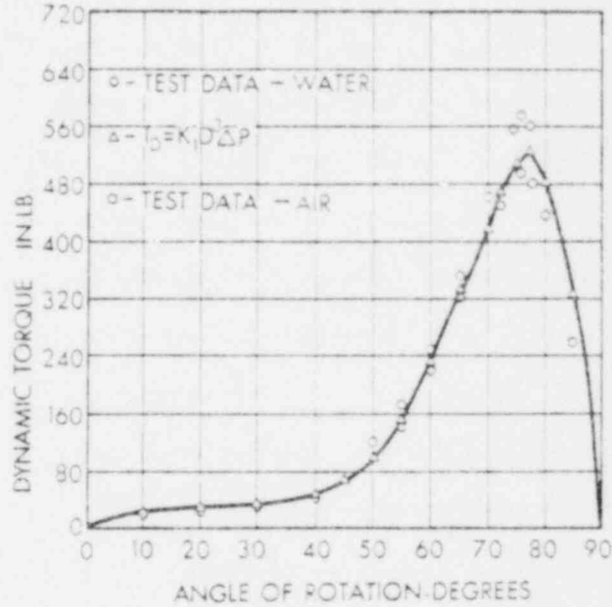

Figure 4. Dynamic torque vs. angle of disc rotation, 8-in. butterfly valve, comparison of experimental results with calculated torque, incompressible flow: P_1 = 100 psig, ΔP = 5 psi.

Torque measurements were made at selected increments of disc rotation (0–90°). A transducer, consisting of a steel bar with strain gages attached, was fixed to the valve shaft and used in conjunction with an oscillograph to measure and record the shaft torque. The data from this test were used to determine the dimensionless torque coefficient plotted as a function of disc rotation on Figure 1. The curves plotted on Figure 2 show excellent agreement between measured torque and the torque calculated using this coefficient.

The next step was to verify that the torque coefficient is indeed applicable to other valve sizes provided geometric similarity is reasonably well maintained. The results on Figures 3 and 4 again show very good agreement between measured torque and calculated torque for two 8-in. valves.

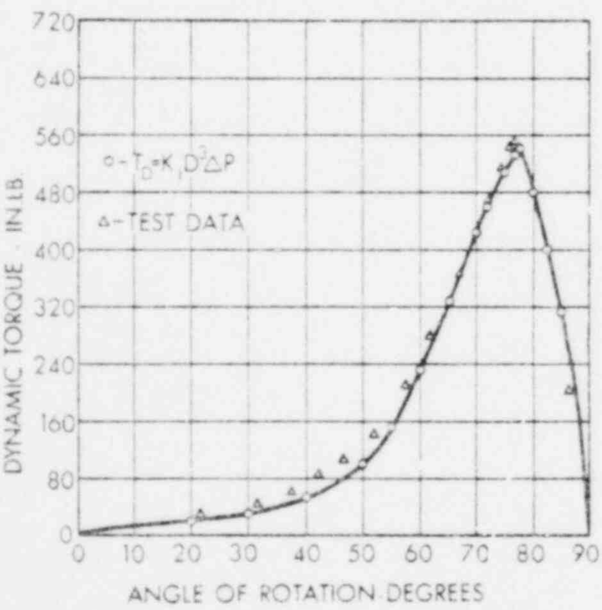

Figure 3. Dynamic torque vs. angle of disc rotation, 8-in. butterfly valve, comparison of experimental results with calculated torque, incompressible flow: P_1 = 100 psig, ΔP = 5 psi.

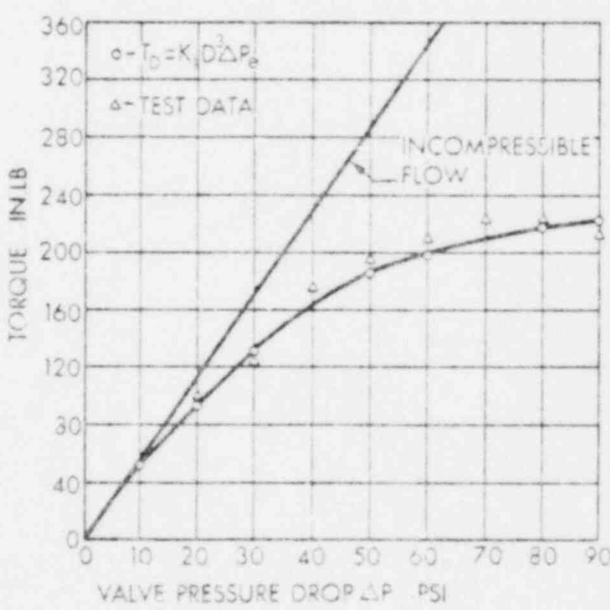

Figure 5. Dynamic torque vs. valve pressure drop, 4-in. butterfly valve, 60° disc rotation, comparison of experimental results with calculated torque, compressible flow: P_1 = 214.4 psia, flowing medium = air.

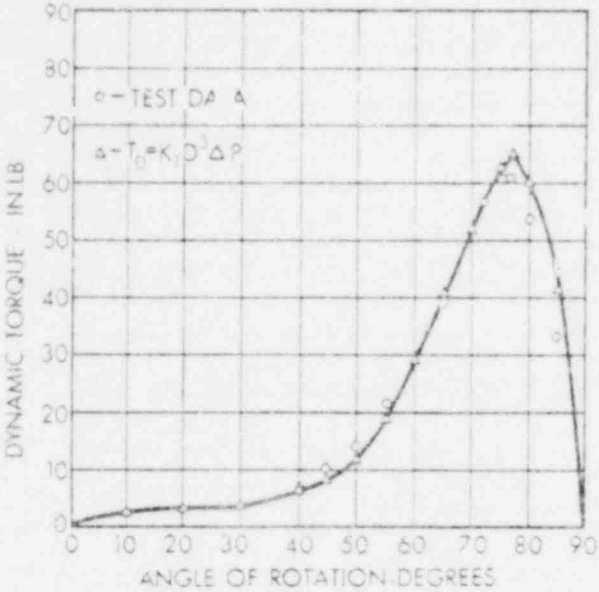

Figure 6. Dynamic torque vs. angle of disc rotation, 4-in. butterfly valve, comparison of experimental results with calculated torque, compressible flow: $P_1 = 114.4$ psia, $\Delta P = 5$ psi ($\Delta P/P_1 = 0.0446$), flowing medium = air.

It should be noted that discs in the two 8-in. valves were of substantially different geometric shape. Using the ratio of disc diameter to hub diameter as an indicator, these ratios were 4.56 : 1 and 3.55 : 1 for the valves used to obtain the data for Figures 3 and 4, respectively. The difference in torque magnitude for these valves with a 5 psi pressure differential shown in Figures 3 and 4 is the result of this difference in geometry. The disc in the 8-in. valve used for the test in Figure 3 was geometrically

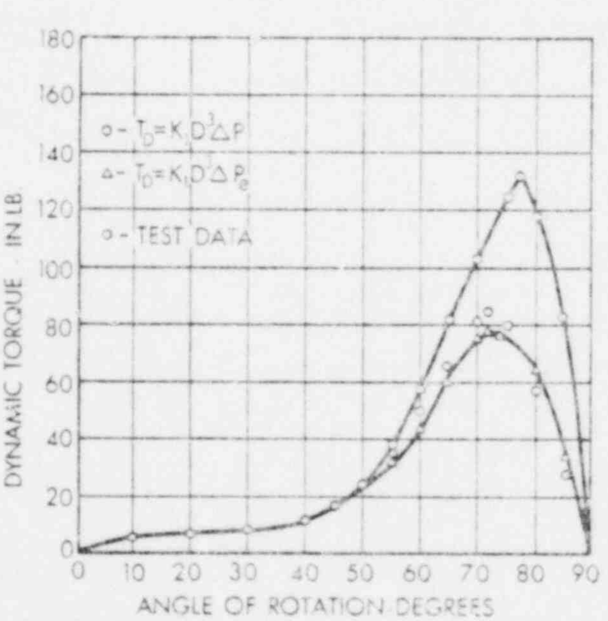

Figure 7. Dynamic torque vs. angle of disc rotation, 4-in. butterfly valve, comparison of experimental results with calculated torque, compressible flow: $P_1 = 64.4$ psia, $\Delta P = 10$ psi ($\Delta P/P_1 =$ ~~0.115~~), flowing medium = air.
0.155

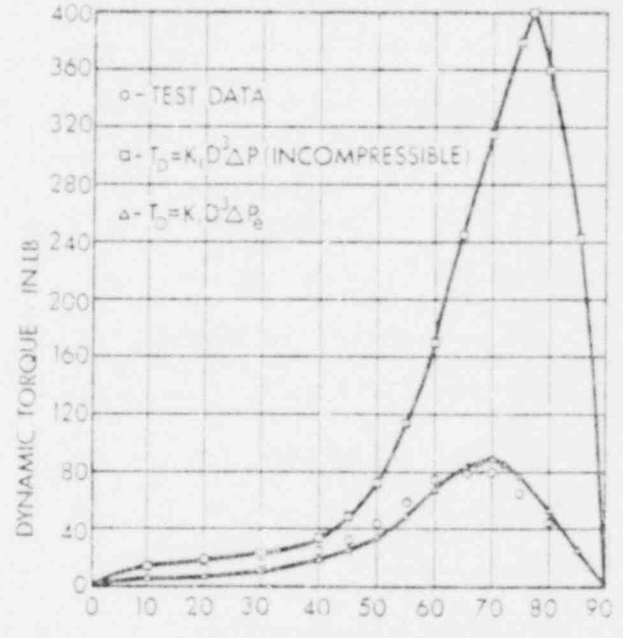

Figure 8. Dynamic torque vs. angle of disc rotation, 4-in. butterfly valve, comparison of test results with calculated torque, compressible flow: $P_1 = 64.4$ psia, $\Delta P = 30$ psi ($\Delta P/P_1 =$ ~~0.466~~), (critical flow) flowing medium = air.
0.466

similar to the disc in the 4-in. test valve used to establish the torque coefficient, K_1.

The extension of the dynamic torque relationship to include the effect of fluid compressibility is accomplished by defining an effective pressure differential as shown in Equation (25). The curves on Figure 5 show the transition from incompressible flow to critical flow with increasing pressure ratio for a 4-in. valve set at 60° disc rotation. Here again there is very good agreement between the torque calculated using Equation (24) and the experimental results. The incompressible torque curve is also shown on Figure 5 to emphasize the effect of fluid compressibility.

The curves on Figures 6 through 8 are presented to compare experimental results with torque calculated using Equation (24) for full 90° disc rotation. At low pressure ratios, the torque using air as the flowing medium is essentially equal to the torque for incompressible flow (Figure 6). As the pressure ratio is increased, the effect of fluid compressibility becomes more pronounced as shown in Figure 7. Once critical flow has been attained, no further increase in torque is realized by increasing the valve pressure differential as shown on Figure 8.

CONCLUSIONS

A technique is presented which can be used to determine the dynamic torque for butterfly valves with reasonable accuracy. The basic torque relationship developed for incompressible flow is extended to include the effect of fluid compressibility. The method presented is developed using the Universal Gas Sizing Equation to define an effective pressure differential for the transition from incompressible flow to critical flow. Application of this method shows excellent agreement with experimental test results.

NOTATION

A = Flow area, in.^{2}
$B_1, B_2,$
B_3, B_4 = Constants of proportionality
C_1 = C_g/C_v
C_2 = Correction factor for variation in specific heat ratio
C_g = Gas sizing coefficient
C_v = Flow coefficient
C = Nominal valve diameter, in.
F = Force, lb
G = Specific gravity
K_t = Dimensionless torque coefficient
M = Mass flow rate, lb/s
P_1 = Inlet pressure, psia
ΔP = Valve pressure differential, psi
ΔP_e = Pressure differential affecting dynamic torque
Q_i = Flow rate incompressible fluid, scfh
C_c = Flow rate compressible fluid, scfh
T = Absolute temperature, °R
T_D = Dynamic torque, in.-lb
V = Fluid velocity, in./s
ρ_1 = Fluid density at upstream pressure tap, lb/in.^{3}
ρ_d = Fluid density at valve disc, lb/in.^{3}

REFERENCES

1. Keller, J. C., and Salzmann, F. F. January 1936. "Aerodynamic Model Tests on Butterfly Valves." *Escher-Wyss News* 9.
2. *Recommended Voluntary Standards for Measurement Procedure for Determining Control Valve Flow Capacity.* 1958. Fluid Controls Institute, Inc., paper FCI 58-2.
3. Buresh, J. F., and Schuder, C. B. October 1964. "The Development of a Universal Gas Sizing Equation for Control Valves." *ISA Trans.* 3:322–328.
4. *Flow Measurement: Instruments and Apparatus. Supplement to the ASME Power Test Codes.* ASME report PTC 19.5; 4-1959.

Attachment 3 - Capacity and Dynamic Torque
Curves Determined in Fis'-r
Lab. Tests for 6" Type 92..
Butterfly Valve.

PROBLEM 983

DATE 7-2-76 FISHER CONTROLS COMPANY REPORT 15

wcs FIGURE 1

FLOW VS TRAVEL CHARACTERISTIC

BODY SIZE 6" DESIGN/TYPE 9200 B/F BODY DWG. F41629-B

SEAL CONSTRUCTION ______ SEAL DWG. ______

MEASURED PROTECTOR RING DIA. ______ PROTECTOR RING DWG. ______

BALL/DISC TYPE 7 TO 1 ASPECT RATIO BALL/DISC DWG. 75-115DXX2012-A

VALVE FLOW DIRECTION: ☐ NORMAL ☒ REVERSE

⊙ WATER TEST

BODY INLET PRESSURE >100 PSIG BODY PRESSURE DROP 2 PSI

AVERAGE C_v = 1690

+ AIR TEST

BODY INLET PRESSURE ______ PSIA BODY PRESSURE DROP ______

AVERAGE C_g = ______

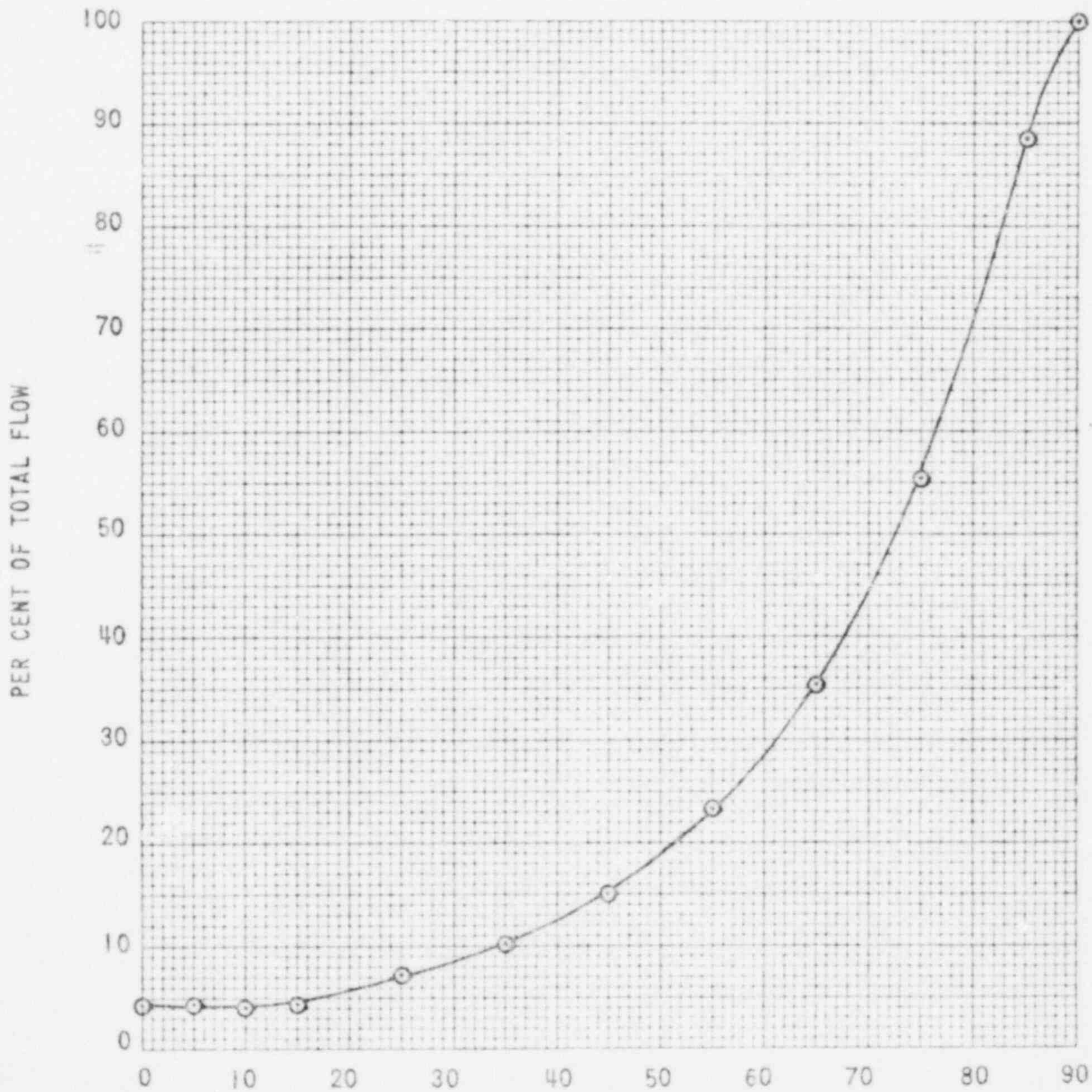

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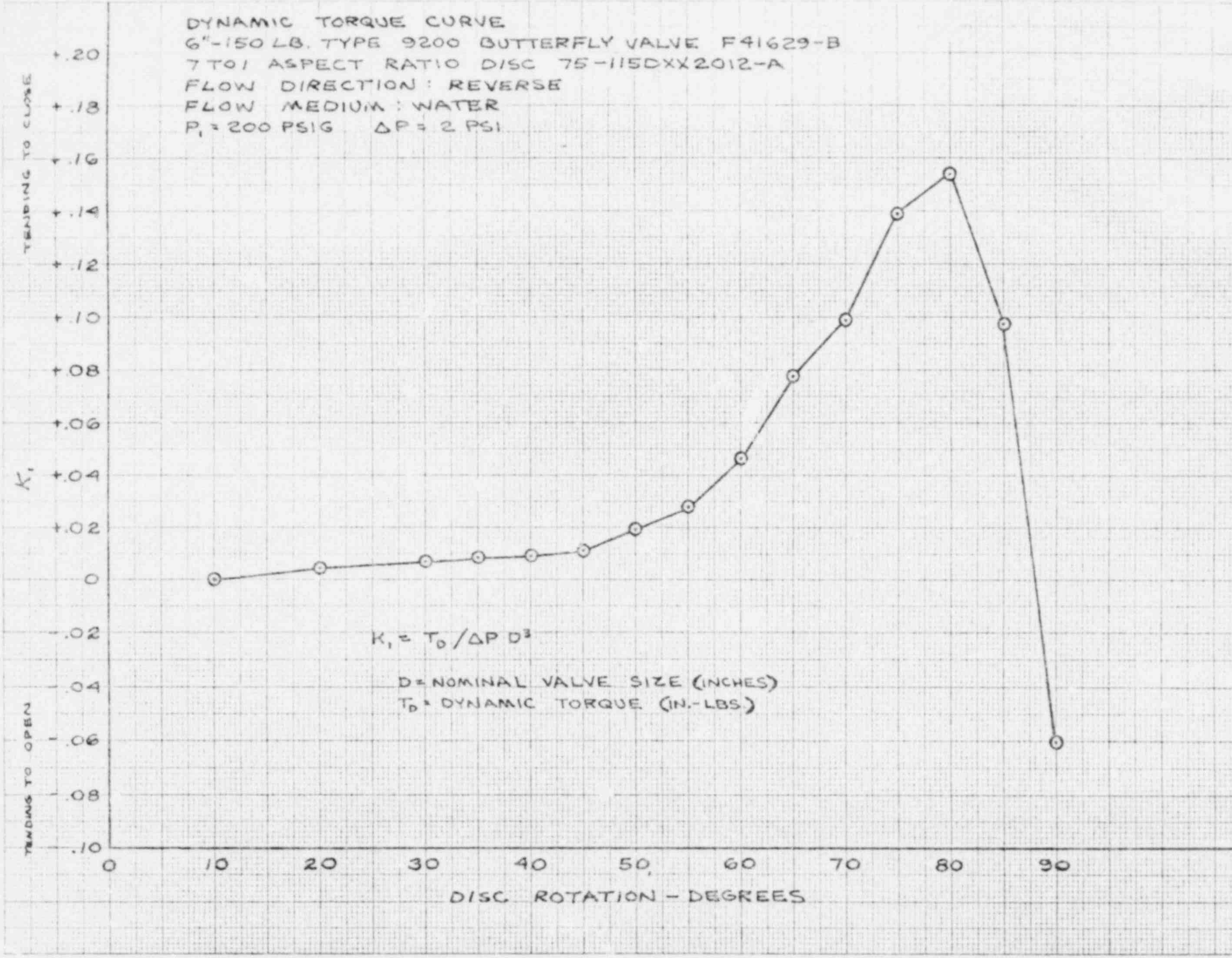
FIG. 2



Attachment 4 - Deleted

Attachment 5 - Fisher Bulletin 51.4:9200,
July 1976; "9200 Series
Butterfly Control Valve
Bodies for Nuclear Service".

9200 Series Butterfly Control Valve Bodies for Nuclear Service

Fisher 9200 Series valves are offset-disc butterfly valves with an adjustable elastomer T-ring seat suitable for extremely stringent shutoff requirements. These valves are often used as nuclear-service valves for on/off applications such as containment isolation and for throttling or on/off flow control of component cooling water or auxiliary service fluids.

Design criteria for pressure-retaining components of 9200 Series valves meet the requirements of the ASME (American Society of Mechanical Engineers) Boiler and Pressure Vessel Code, Sections III and VIII, and the valve body assemblies can be furnished with the ASME "N"-stamp symbol.

The standard plate steel or cast steel 9200 Series wafer-style valve is installed between pipeline flanges. An optional steel single-flange style body is available with a pipeline flange on one end and a buttwelding connection on the other end or with a buttwelding connection on both ends. This optional construction can be welded directly to a containment vessel wall.

These valves are available in 4-inch through 96-inch sizes for process temperatures to 400°F and, depending upon size, pressure drops to 150 psi.

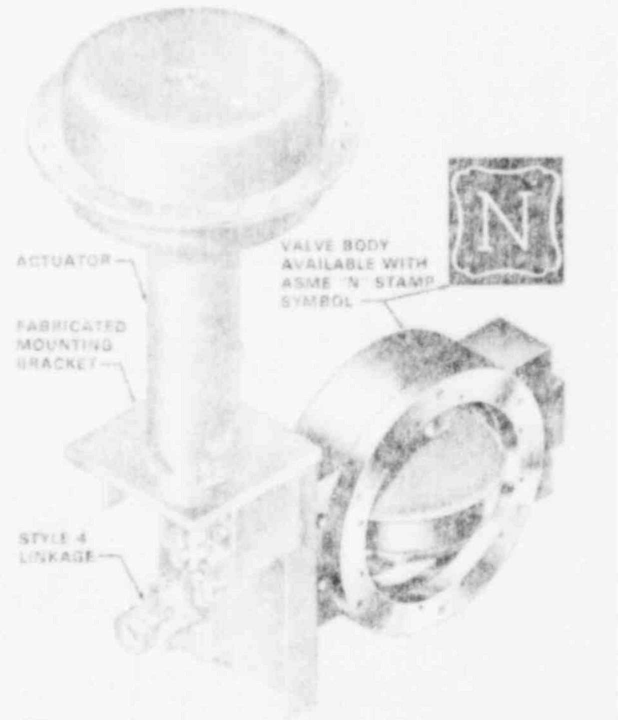


Figure 1. 9200 Series Specification B-1 Valve Body with Type 656 Actuator

Features

• **Compliance with Nuclear Code and Other Requirements**—Fisher Controls Company holds the ASME Certificate of Authorization to use the "N"-stamp symbol on these valve body assemblies. All ASME requirements for Class 1, 2, and 3 nuclear-service valves, as well as special customer assembly, cleaning, painting, and packaging requirements, can be met. In addition, compliance of valve and actuator assemblies with specified seismic and environmental criteria can be documented with seismic analysis calculations and/or actual test results.

• **Economical**—Standard plate steel construction requires less extensive nondestructive examinations than cast construction, reducing cost and delivery time. Standardized valve/actuator size combinations ensure sufficient actuator power while reducing actuator selection time and documentation cost and delay.

• **Excellent Shutoff without Excessive Seating Torque**—Offset disc design allows disc/T-ring contact around 360 degrees of disc circumference. Elastomer T-ring is field adjustable so that shutoff can be maintained without excessive disc/T-ring interference and associated high seating torque. Reduced interference also minimizes T-ring wear to prolong T-ring life.

• **Reduced Leak-Off Piping Requirements**—Sizes through 24 inches use valve shaft packing on one side of valve only; only one leak-off connection to pipe

Specifications

AVAILABLE CONFIGURATIONS AND BODY SIZES

9200 Series Specification B-1: Offset-disc butterfly control valve body with adjustable elastomer T-ring seat contained between the retaining ring and valve body as shown in figure 2. Valve shaft sealed with packing on actuator side and with a blank-off plate on the other side. Available in ■ 4, ■ 6, ■ 8, ■ 10, ■ 12, ■ 14, ■ 16, ■ 18, ■ 20, and ■ 24-inch sizes

9200 Series Specification C-1: Offset-disc butterfly control valve body with adjustable elastomer T-ring seat contained between the retaining ring and the valve disc face as shown in figure 3. Valve shaft sealed with packing on both sides. Available in ■ 30 through ■ 96-inch sizes in 6-inch increments

BODY STYLE

Flangeless (wafer-type) body with four flange bolt holes (see figures 1 and 4) for installation between two pipeline flanges

END CONNECTION STYLES

4 Through 24-Inch Sizes: Mate with ANSI Class 150 (B16.5) raised-face flanges

30 Through 96-Inch Sizes: Mate with ■ ANSI Class 125 (B16.1) flat-face flanges (through 72-inch size only), ■ AWWA C207 flanges, or ■ MSS SP-44 flanges

MAXIMUM INLET PRESSURE[1]

4 Through 24-Inch Sizes: Compatible with ANSI Class 150 pressure/temperature ratings for temperatures from +20 to +400°F

30 Through 96-Inch Sizes: ■ 75 psig for temperatures from +20 to +400°F (in accordance with ASME Code Case 1678, approved December 16, 1974) or ■ higher pressures upon request

MAXIMUM PRESSURE DROP[1]

Shutoff (0 Degrees of Disc Rotation)

4 Through 24-Inch Sizes: 150 psi

30 Through 96-Inch Sizes: 75 psi

Flowing: See table 1

CONSTRUCTION MATERIALS

Body,[2] Disc,[2] and Retaining Ring[3]: ■ Steel plate (ASME SA515 GR 70) or ■ other materials available upon request

Shaft: 17-4PH stainless steel (ASME SA564 GR630 H1075)

Taper Pins: Same material as shaft

Blank-Off Plate[2] (Through 24-Inch Size Only): 316 stainless steel (ASME SA240 GR316)

Blank-Off Plate Bolting (Through 24-Inch Size Only)

Studs:[2] Steel (ASME SA193 GR B8M)

Nuts:[2] Steel (ASME SA194 GR 8M)

T-Ring: ■ EPDM (ethylene-propylene) or ■ nitrile; ■ Viton[4] is also available for non-nuclear applications

Retaining Ring O-Ring (Through 24-Inch Size Only): EPDM (ethylene-propylene)

Blank-Off Plate Gasket (Through 24-Inch Size Only): Spiral-wound gasket of 304 stainless steel and asbestos

Packing: Alternated rings of Crane[5] 187-I and laminated graphite (Grafoil[6]) packing

Bushings: ■ Graphite-impregnated bronze (bushing 2) or ■ alloy 6 (bushing 3)

Bushing Retainers and Retainer Tube: 316 stainless steel

Packing Follower: Steel

Packing Lantern Rings and Washers: 316 stainless steel

Packing Box Studs and Nuts: Steel

Thrust Collars

Shaft Diameters to 1-1/2 Inches (Through 18-Inch Valve Size): Cadmium-plated steel clamp-type collars with brass washers between collars and bearing surfaces

Shaft Diameters over 1-1/2 Inches: Bronze collars pinned to valve shaft

Actuator Mounting Bracket: Fabricated Steel

OPERATIVE TEMPERATURE[1, 7]

With EPDM T-Ring: +20 to +300°F

With Nitrile T-Ring: +20 to +200°F

1. None of the pressure or temperature limitations in this bulletin, nor any applicable code limitation, should be exceeded.
2. Pressure-retaining part.
3. Pressure-retaining part on 4-inch through 24-inch sizes only.
4. Trademark of Du Pont Co.
5. Trademark of Crane Packing Co.
6. Trademark of Union Carbide Corp.
7. This term is defined in SAMA Standard PMC 20.1-1973.

Specifications (Continued)

With Viton T-Ring: +20 to +400°F (Do not use with water over 180°F or steam)

ACTUATOR TORQUE REQUIRED See table 1

FLOW DIRECTION Flow is permissible in either direction, but valve is normally installed with T-ring retaining ring facing downstream

FLOW COEFFICIENTS See Fisher Catalog 10

SHUTOFF CLASSIFICATION Fisher Class VI (less than one bubble per minute using air at a pressure drop of 150 psi for 4 through 24-inch sizes and 75 psi for 30 through 96-inch sizes)

DISC ROTATION ■ Clockwise to open or ■ counterclockwise to open (when viewed from actuator side of valve) through 90 degrees of disc rotation

ACTUATOR MOUNTING Fabricated actuator-mounting bracket is used to mount Fisher Type 480-15, 481-15, 656 and 864 actuators. Style 4 adjustable linkage, shown in figure 1, is used with Fisher actuators for travels of 4 inches and less and valve shaft diameters of 1-1/2 inches and less. Fixed linkage is used for longer travels and larger valve shafts.

Actuator can be ■ perpendicular to (standard) or ■ parallel with pipeline (adaptor required for parallel mounting of actuators requiring a mounting bracket) with actuator to ■ right (standard) or ■ left of valve (when viewed from valve inlet)

With perpendicular mounting in horizontal pipeline, actuator can extend ■ above (standard) or ■ below pipeline. With parallel mounting, actuator can extend ■ upstream or ■ downstream.

ACTUATOR/VALVE ACTION ■ Push-down-to-open (extending actuator stem opens valve) or ■ Push-down-to-close (extending actuator stem closes valve)

MATING FLANGE CAPABILTIES Compatible with welding-neck and slip-on flanges

CODE CLASSIFICATIONS Valve body, disc, and shaft components designed in accordance with allowable stress levels as specified in ASME Boiler and Pressure Vessel Code, Sections III and VIII

Valve body assemblies available as nuclear code Class 1, 2, or 3 valve with ASME "N"-stamp symbol

TESTING REQUIRED All nondestructive examinations (NDE) required for Class 1, 2, and 3 nuclear-service valves can be furnished; for current list of NDE requirements, see Fisher Catalog 11

PACKING BOX TYPE Leak-off type packing box with 1/2-inch NPT female leak-off connection

VALVE SHAFT DIAMETERS See figure 7

APPROXIMATE WEIGHTS See figure 7

OPTION Single-flange steel valve body with ■ full set of flange bolt holes on one end and buttwelding-end connection on the other end as shown in figure 2 or with ■ a buttwelding end connection on both ends. Flanged end connection available as noted in "End Connection Styles" above; buttwelding-end connection available per ■ ANSI B16.25 or ■ as specified

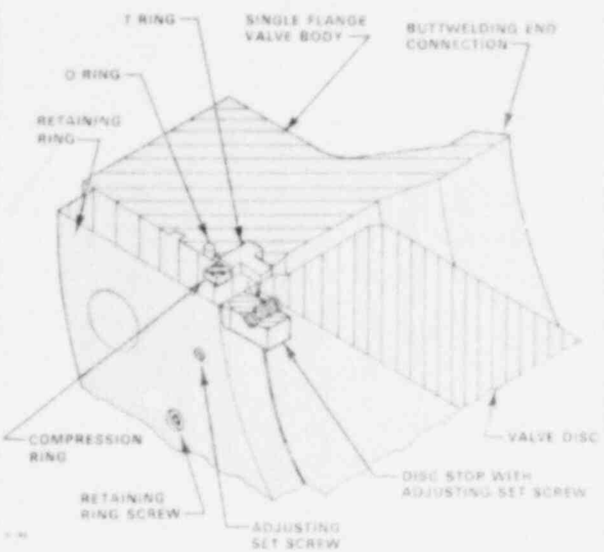


Figure 2. 9200 Series Specification B-1 Valve T-Ring Details (Optional Single-Flange/ Buttwelding End Construction)

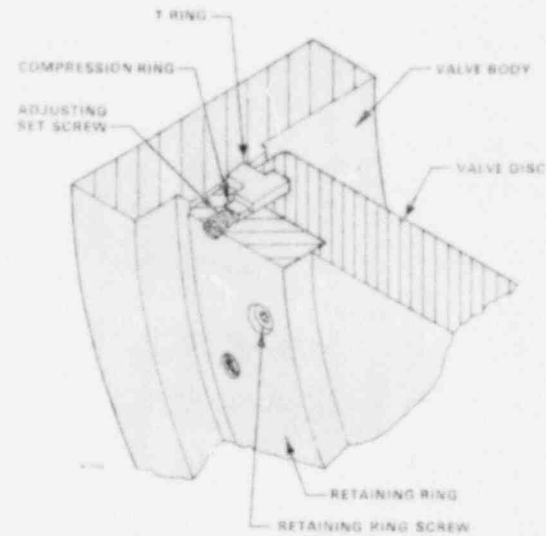


Figure 3. 9200 Series Specification C-1 Valve T-Ring Details

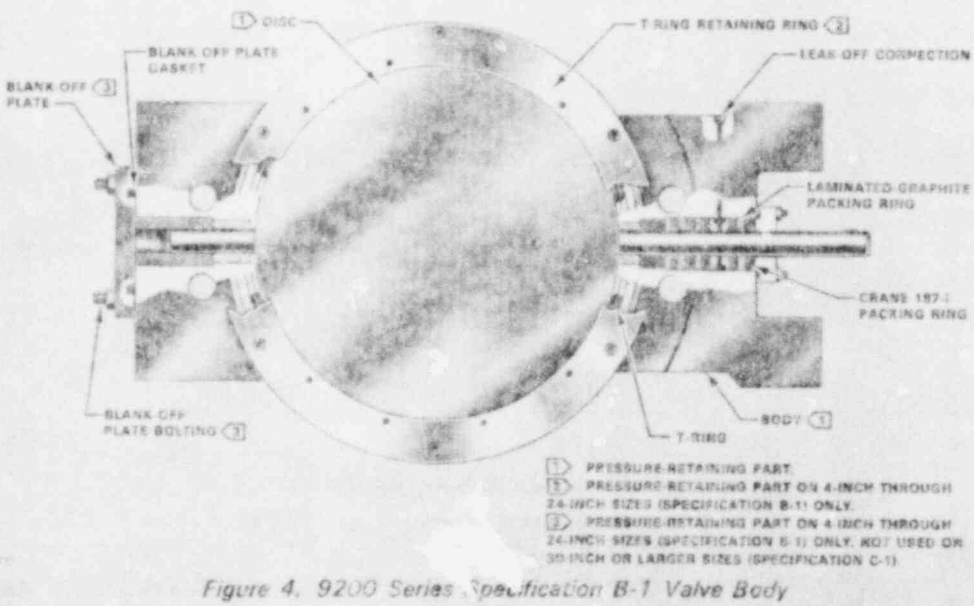


Figure 4. 9200 Series Specification B-1 Valve Body

Valve and Actuator Selection

Note

Valve and actuator selection can be made from table 1 for 4-inch through 72-inch valves, pressure drops to 150 psi (depending upon valve size), and process temperatures from +20 to +400°F (depending upon elastomer T-ring material selected and application).

The torques in the "Actuator Torque Required" column of the table are the maximum torques encountered when the disc is being closed (or opened) against the shutoff (0-degrees of disc rotation) pressure drop shown in the table. Pressure drops shown for open disc angles (60 or 90 degrees) are the maximum flowing drops that the torques in the "Actuator Torque Required" column will permit. (Where necessary, maximum pressure drops shown for open angles have been limited by strength capabilities of construction materials.)

Table 1. Valve and Actuator Selection

VALVE SIZE[1] (INCHES)	VALVE SHAFT DIAMETER (INCHES)	OPERATIVE TEMPERATURE	BUSHING TYPE[2]	MAXIMUM PRESSURE DROP (PSI) Angle of Disc Opening (Degrees) 0	60	90	ACTUATOR TORQUE REQ'D (INCH-POUNDS)	RECOMMENDED ACTUATOR TYPE AND SIZE
4	5/8	EPDM T-Ring: -20 to +300°F; Nitrile T-Ring: -20 to +200°F; Viton T-Ring: -20 to +400°F	2	150	49.7	16.9	410[3]	656 Size 40 w/35 psig supply[4]
			3	150	59.1	21.5	525[3]	656 Size 40 w/35 psig supply[4]
6	3/4		2	150	36.4	12.1	925[3]	656 Size 60 w/35 psig supply[4]
			3	150	46.1	15.0	1250[3]	656 Size 60 w/35 psig supply[4]
8	1		2	150	30.5	10.1	1820[3]	656 Size 60 w/35 psig supply[4]
			3	150	40.4	14.2	2575[3]	480-15 Size 40 w/80 psig supply[5]
10	1		2	150	24.7	8.1	2780[3]	480-15 Size 40 w/80 psig supply[5]
			3	144	16.0	8.1	3960	480-15 Size 60 w/80 psig supply[5]
12	1 1/4		2	150	22.7	7.6	4425[3]	480-15 Size 60 w/80 psig supply[5]
			3	150	27.0	9.0	6550[3]	480 Size 80 w/80 psig supply[5]
14	1 1/4		2	150	19.7	6.5	5360[3]	480-15 Size 60 w/80 psig supply[5]
			3	140	19.5	6.4	7950	864 Size[6] 6 x 20 w/80 psig supply[5]
16	1 1/2		2	150	18.9	6.3	7785[3]	480 Size 80 w/80 psig supply[5]
			3	150	22.5	7.5	11,900[3]	864 Size[6] 6 x 20 w/80 psig supply[5]
18	1 1/2		2	150	15.7	5.0	10,550[3]	864 Size[6] 6 x 20 w/80 psig supply[5]
			3	129	15.0	4.9	16,950	864 Size[6] 8 x 20 w/80 psig supply[5]
20	1 3/4		2	150	16.5	5.4	13,100[3]	864 Size[6] 6 x 20 w/80 psig supply[5]
			3	148	18.0	5.9	20,550	864 Size[6] 8 x 20 w/80 psig supply[5]
24	2		2	150	15.0	4.9	20,700	864 Size[6] 8 x 20 w/80 psig supply[5]
			3	137	15.0	4.9	33,170	864 Size[6] 10 x 20 w/80 psig supply[5]
30	2		2	75	6.5	2.1	17,732[3]	864 Size[6] 8 x 16 w/80 psig supply[5]
			3	75	6.2	2.0	27,932	864 Size[6] 8 x 20 w/80 psig supply[5]
36	2 1/2		2	75	6.3	2.0	28,830	864 Size[6] 8 x 20 w/80 psig supply[5]
			3	75	5.9	2.0	46,830	864 Size[6] 10 x 24 w/80 psig supply[5]
42	2-1/2		2	75	5.5	1.7	40,020	864 Size[6] 10 x 20 w/80 psig supply[5]
			3	75	5.2	1.7	64,770	864 Size[6] 12 x 20 w/80 psig supply[5]
48	3		2	75	5.3	1.7	58,675	864 Size[6] 12 x 20 w/80 psig supply[5]
			3	75	5.1	1.7	97,750	Contact Fisher Representative
54	3 1/2		2	75	5.3	1.7	83,555	Contact Fisher Representative
			3	75	5.0	1.6	141,680	Contact Fisher Representative
60	3 1/2		2	75	4.8	1.5	103,585	Contact Fisher Representative
			3	75	4.6	1.5	175,660	Contact Fisher Representative
66	4		2	75	4.7	1.5	137,570	Contact Fisher Representative
			3	75	4.5	1.5	237,320	Contact Fisher Representative
72	4 1/2		2	75	4.7	1.5	180,130	Contact Fisher Representative
			3	75	4.5	1.5	313,630	Contact Fisher Representative

1. For larger sizes, contact the Fisher sales representative.
2. Bushing 2 - graphite impregnated bronze; bushing 3 - alloy 6.
3. Within torque capabilities of Fisher manual handwheel actuators.
4. With or without valve positioner. Selection valid only if full actuator travel, standard 6 to 30 psig (nominal) spring, and positioner supply for maximum diaphragm input signal of 35 psig are used.
5. With or without valve positioner. If 480 Series actuator without positioner is desired, substitute Type 481 or 481-15 for Type 480 or 480-15. Selection valid only if full actuator travel and minimum positioner supply pressure for cylinder operating pressure of 80 psig are used.
6. Cylinder bore diameter (inches) x maximum actuator travel (inches).

All pressure drops shown are within the strength capabilities of the materials shown in the "Specifications" table.

After determining the proper valve size using Fisher Catalog 10 and the sizing nomographs or slide rule, refer to table 1. Check the maximum allowable pressure drop at the appropriate open angle (either 60 or 90 degrees) to be certain it equals or exceeds that which will be encountered in service.

Recommended Fisher actuator types, sizes, and operating pressures for each selection are shown at the right of the table. In addition, other actuator types, such as electric and spring-return pneumatic rotary actuators are also available in recommended combinations with 9200 Series valve bodies. All combinations in table 1 are predetermined to have sufficient torque output at the stated operating conditions.

Selection from among the recommended combinations reduces documentation cost and possibility of delay. Contact the Fisher sales representative if other combinations are required.

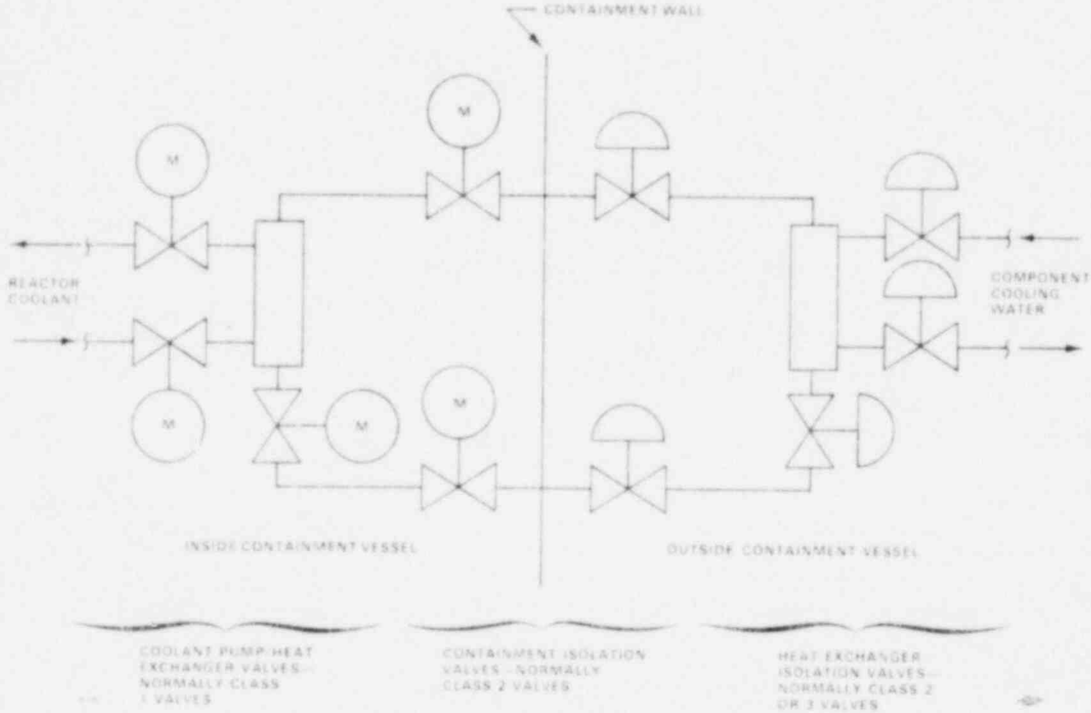


Figure 5. Typical Applications for Class 1, 2, and 3 Nuclear-Service Valves Shown in a Component Cooling System

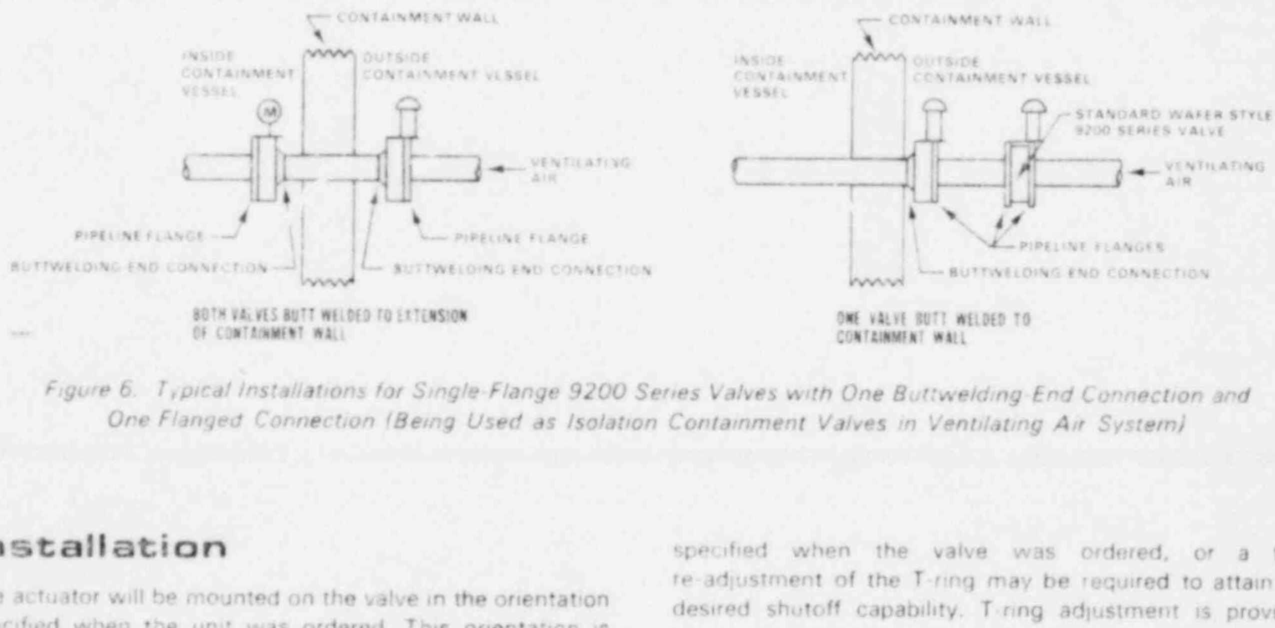


Figure 6. Typical Installations for Single-Flange 9200 Series Valves with One Buttwelding-End Connection and One Flanged Connection (Being Used as Isolation Containment Valves in Ventilating Air System)

Installation

The actuator will be mounted on the valve in the orientation specified when the unit was ordered. This orientation is normally selected based upon the desired mounting position in the pipeline, available space at the point of installation, etc.

For 30-inch and larger valve sizes, factory seat leak testing must be performed with the valve in the same position as is intended for the actual installation. For these larger sizes, install the valve in the same position as was specified when the valve was ordered, or a field re-adjustment of the T-ring may be required to attain the desired shutoff capability. T-ring adjustment is provided by a compression ring and adjusting set screws as shown in figures 2 and 3.

Flow through the valve can be in either direction, but the valve is normally installed with the T-ring retaining ring facing downstream. For 30-inch and larger sizes, it may be desired to install the valve such that the T-ring retaining ring faces the nearest manhole or other pipeline access point. This will facilitate T-ring inspection and maintenance.

VALVE SIZE	LETTERED DIMENSION						MINIMUM ALLOWABLE DIAMETER OF MATING PIPE OR FLANGE	APPROXIMATE WEIGHT OF VALVE BODY ASSEMBLY (POUNDS)
	A	B	C	E*	F	S		
4	4.00	6.50	8.50	6.25	3.25	5/8	3.64	70
6	6.00	7.50	10.50	6.25	3.75	3/4	5.56	100
8	8.00	9.00	10.50	6.25	3.75	1	7.81	130
10	10.00	10.00	12.00	6.25	3.75	1	9.81	175
12	12.00	12.00	14.00	7.62	4.75	1-1/4	11.50	320
14	13.25	13.00	15.50	7.62	4.75	1-1/4	13.00	375
16	15.25	14.50	17.50	7.62	4.75	1-1/2	15.12	475
18	17.00	15.00	18.50	7.62	5.00	1-1/2	16.75	520
20	19.00	17.00	20.50	8.75	5.50	1-3/4	18.88	685
24	23.00	20.00	21.50	8.75	6.00	2	22.75	1040

*Standard E dimensions shown are valid for actuator selections shown in table 1. Special E dimensions may be required for other actuator types.

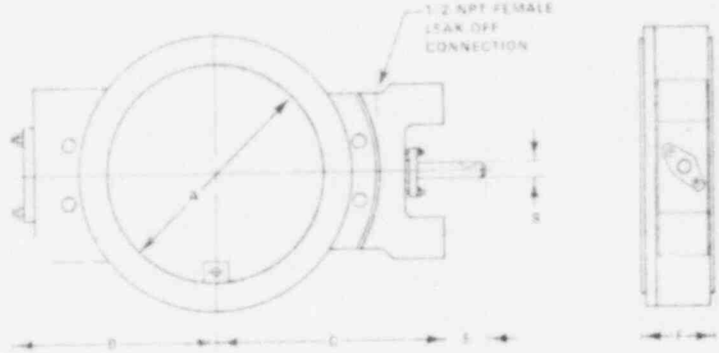


Figure 7. Dimensions (Inches)

The 9200 Series valves are supplied with a disc travel stop. If the valve body and actuator have been ordered separately or if the actuator has been removed for maintenance, be certain that proper rotation direction will be obtained from the actuator before installing.

If spiral-wound line flange gaskets are to be used with the 4-inch through 24 inch sizes, be certain the gaskets are of a type and size that will not overlap the cap screw or adjusting screw holes in the T-ring retaining ring. Under line flange bolting compression, spiral-wound gaskets can be damaged by the cap screw or adjusting screw holes.

Ordering Information

Application

When ordering, specify:

1. Type of Application
   a) Throttling or on/off
   b) Reducing, relief, or back pressure
2. Controlled fluid (include chemical analysis of fluid if possible)
3. Specific gravity of controlled fluid
4. Fluid temperature (normal and minimum and maximum anticipated)
5. Range of flowing inlet pressures
6. Pressure Drops
   a) Range of flowing pressure drops
   b) Maximum at shutoff
7. Flow Rates
   a) Minimum controlled flow
   b) Normal flow
   c) Maximum flow
8. Maximum allowable leakage rate
9. Specify the position in which the valve will be installed (e.g., valve in horizontal pipeline with valve shaft horizontal). Seat leak testing will be performed with the valve in the same position as is intended for the actual installation.
10. Nuclear-code class and all nuclear and special requirements
11. Line size and schedule

Valve Body Information

Refer to the "Specifications" on page 2. Review the description at the right of each specification and in the referenced table. Indicate the choice wherever there is a selection to be made.

Actuator and Accessory Information

Specify the desired actuator type and size from the appropriate actuator bulletin. Also refer to the specific actuator and accessory bulletins for additional ordering information.

Attachment 6 - "Dynamic Test Program on
Bettis T-420-SR1-M3 for
EBASCO Services, Inc.,
Agents for Louisiana Power
and Light, Waterford Steam
Electric Station, Unit No. 3".

CUSTOMER ORDER NO. NY-403483
SELLER ORDER NO. 4D230-27 thru 32 and
7C627-DD, FF & GG
AGENT ORDER NO. 026M-58148 and
001-70850

FISHER CONTROLS COMPANY
CONTINENTAL DIVISION

DYNAMIC TEST PROGRAM

ON

BETTIS T420-SR1-M3 ACTUATOR

FOR

EBASCO SERVICES, INC.

AGENTS FOR

LOUISIANA POWER AND LIGHT
WATERFORD STEAM ELECTRIC STATION
UNIT NO. 3

REGISTERED PROFESSIONAL CARL D. WILSON ENGINEER

DATE: June 29, 1978; Revision 1: September 18, 1978

PREPARED BY: Eric P. Ringle
Eric P. Ringle, Project Engineer

REVIEWED BY: Carl D. Wilson
Carl D. Wilson, Manager, Testing and Analysis

APPROVED BY: Richard Hoober
Richard E. Hoober, Manager of Engineering

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1.0 - INTRODUCTION

Fisher Controls Company has completed a dynamic test program on a Bettis T420-SR1-M3 Actuator to demonstrate its ability to perform during a seismic event. This test report is being submitted to qualify some identical and similar actuators on Ebasco Purchase Order NY403483. This report is written to meet the seismic requirements, under dynamic testing, in Ebasco Specification LOU-1564.109A.

In addition, Fisher Controls Company will qualify the Bettis T416-SR3-M3 Actuator with this test. The basis of this cross qualification lies in the similarities of the two actuators. It can be demonstrated that the test actuator (T420-SR1-M3), by being larger in both size and weight, is a worse case condition when compared with the T416-SR3-M3 actuator, thus qualifying them both.

Both actuators are models in the Bettis T-4 series of spring-return actuators. The difference is in the overall size and torque capabilities. The two digit number following the T-4 in the actuator model number represents the size of the piston used in the air stroke. The T416 has a 16" diameter piston and the T420 has a 20" diameter piston. The SR number refers to the size of the spring used in the return stroke. The SR1 in the T420 actuator is larger in size and weight than the SR3 in the T416 actuator. The M3 on the model number refers to the attached handwheel, which is identical on both.

In summary, due to the differences noted above, the T420-SR1-M3 actuator is larger in size (79-3/16" as compared to 74-1/16") and weight (805# as compared to 615#) when compared to the T416-SR3-M3 actuator. It is clearly a worse case condition when seismic loadings are considered and brackets of equal design are used.

2.0 TEST ASSUMPTIONS

The test was conducted to qualify many similar actuator/valve combinations. Therefore, the test was not conducted for any one customer or valve type. Some assumptions were made to broaden the scope of the qualification. Those which affect Ebasco will be explained in this section.

The test in question was performed without the total valve assembly. The bracket and actuator were mounted on a test fixture in place of an actual valve body. Justification for this is based on Fisher Controls' experience in past dynamic tests. In all tests completed, we have never found a case where binding in the shaft, due to deflections in the valve body or disc, has affected the actuator's ability to stroke the valve. When stroke time is measured, it remains constant under seismic loading. It is our opinion that the valve itself will not affect the actuator's ability to function, therefore, we conclude it reasonable to only verify that the torque output of the actuator will not vary under seismic loadings.

Qualification of accessories will depend on the particular customer's requirements. In Ebasco's case, we are dealing with Namco limit switches and Asco solenoid valves. The limit switches will be qualified by analysis. It is our intent to mount all limit switches on the idle end of the valve body, opposite the actuator. An analysis of the mounting bracket will be submitted showing that the resonant frequency of the switch bracket will exceed 33 Hertz, and the resulting stresses will be within Ebasco's specified limitations. As a supplement, Namco's qualification report will be submitted for qualification.

To qualify the Asco, an accelerometer was located in the approximate mounting location of the solenoid valve to record the actual loadings it would see. Read out from this accelerometer showed that the increased response due to the flexible construction did not exceed the g-levels to which Asco has previously qualified their equipment. Therefore, Asco's qualification is still valid, and we will submit that report for qualification.

FISHER CONTROLS COMPANY
CONTINENTAL DIVISION

3.0 - SCOPE OF QUALIFICATION

This report applies to the following item and tag numbers:

CONTINENTAL ITEM NO.	EBASCO ITEM NO.	EBASCO TAG NO.	VALVE SIZE	VALVE TYPE	ACTUATOR
4D230-27 4D230-28 4D230-29 4D230-30 4D230-31 4D230-32	6	2HV-B150B 2HV-B151A 2HV-B152A 2HV-B153B 2HV-B154B 2HV-B155A	48"	9220	Bettis T420-SR1-M3
7C627-DD	21	3HV-B217B 3HV-B218A	42"	9220	Bettis T420-SR1-M3
7C627-FF	23	3HV-B223B 3HV-B224A	30"	9220	Bettis T416-SR3-M3
7C627-GG	24	3HV-B226A 3HV-B227B	36	9220	Bettis T420-SR1-M3

4.0 - TEST PROCEDURE

1.0 TEST SPECIMEN:

1.1 Specimen Description:

A Bettis Type T420-SR1-M3 Pnuematic, Spring-return Actuator, identical to the actuator supplied on Ebasco Item Number 6, was used in testing. Specimen is shown in Photograph #1.

1.2 Specimen Mounting:

The specimen, mounted on a fabricated bracket (Photograph #3), was attached to a Fisher designed and fabricated mounting fixture (Photograph #4) and the fixture, in turn, was fastened to a seismic simulator table. The fixture included a short piece of shafting, instrumented with strain gauges, to verify the torque output of the actuator. The specimen was oriented such that its longitudinal axis was colinear with the longitudinal axis of the test table. The specimen was rotated 90 degrees in the horizontal plane, for the second axis of tests.

2.0 EXCITATION:

2.1 Simultaneous Biaxial Excitation:

Each horizontal axis was excited separately, each one simultaneously with the vertical axis. The horizontal and vertical inputs were tested in the in-phase condition.

2.2 Resonant Search:

A low level biaxial sine sweep, with a minimum input of .2 g's in each test axis was performed to establish resonances. The tests were performed over a frequency range of 1 Hz. to 40 Hz. at a rate of one octave per minute. A resonance is defined as a response order of magnitude greater than or equal to two.

2.3 Sine Beat Tests:

The specimen was subjected to biaxial sine beat tests at the frequencies determined in the resonant search. The input acceleration was a minimum of 1.0 g's horizontal and .67 g's vertical, as specified in Ebasco's specification. The specimen was operated throughout the sine beat tests to ensure operability. A regulated air source with a minimum of 85 PSI, was supplied to the actuator to operate actuator in one direction. The internal spring was used for the return stroke. The sine beat test consisted of ten oscillations per beat, five beats per test frequency with a two-second pause between beats.

3.0 INSTRUMENTATION

3.1 Excitation Control:

A control accelerometer was mounted on the table near the base of the test fixture (Photograph #2).

3.2 Specimen Acceleration Response:

Four (4) biaxial piezo-electric accelerometers were located on the specimen during testing. Placement of accelerometers is shown in Photograph #1.

4.0 TEST RESULTS

4.1 Sine Sweep Test:

Resonant frequencies were located at 25 Hz., 31 Hz., 32 Hz., 34 Hz., 35 Hz., and 40 Hz. during the sine sweep test.

4.2 Sine Beat Test:

The actuator operated successfully when subjected to sine beats at the frequencies determined above. The specimen suffered no physical damage.