PURGE VALVE OPERABILITY REPORT

SNUPPS PROJECT

DECEMBER 1983

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INTRODUCTION

This report has been prepared to allow the NRC to close out the confirmatory items (Number 28, TMI Action Plan II.E.4.2) for Callaway and Wolf Creek.

The information contained herein addresses the items contained in the NRC paper "Clarification of Sept. 27 [1979] Letter to Licensees Regarding Demonstration of Operability of Purge and Vent Valves." In addition, the information contained herein addresses Attachments 4 and 5 of NRC letter B. Youngblood (NRC) to D. Schnell (UE) dated September 7, 1982.

The following data are provided on the valve/actuator design used on the SNUPPS plants' mini-purge containment isolation valves:

Valve Assembly Construction

18-inch Type 9220 Fisher Controls Company butterfly valve

Class 150 flanged × BWE body

2-inch shaft, Class 4, 17-4PH

Plate disc, carbon steel

Adjustable EPDM T-ring seal

Graphite-bronze bushing #2

GH Bettis Actuator T416B-SR3-12

ASCO Type NP8320A185V, 3-way solenoid

NAMCO EA180-31302/32302 limit switches

Fisher type 95H pressure regulator

Versa VSP-3601-155H 3-way switching valve

Texsteam Model 35R pressure relief valve

Hoffman junction box, Model A1008-CHNF with G.E. No. EB-25 terminal strips and assorted seals

Fisher P595 filter with brass element

Service Conditions

Fluid: Normal (air), accident (steam)

Design P/T: 70 psig/320 F

Shutoff ΔP: 60 psid

Service temperature: 120 F

90 degree rotation

Closure time: 3 seconds maximum

Maximum LOCA containment pressure: 47.2 psig (61.9 psia)

Containment pressure at valve closure: 22.9 psig (37.6 psia)

Seismic qualification level: 4.5 g RIM (Seismic Category I)

Applicable ASME Section III Code

ASME Code Class 2

Summer 1975 Addenda

The responses to items in the clarification paper follow.

OPERABILITY OF PURGE AND VENT VALVES

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

RESPONSE:

The LOCA containment conditions utilized in predicting the ΔP across the valves were extracted from the containment pressure-temperature analysis for the large LOCA which results in the peak contained pressure. The pressure response is shown in Figure 1, which also indicates the times that the valve receives a signal to close and the time that the valve is fully closed. When the valve is fully closed the containment pressure is 22.9 psig. As shown in Figure 1, the containment pressure rises from approximately 14 psig to 22.9 psig during the closure stroke; however, the pressure drop at incremental valve positions was calculated at the maximum pressure (22.9 psig: valve closed). This was done for conservatism.

The calculated pressure drops considered the line and component pressure drops for the steam-air mixture discharging through the line. The calculations also conservatively assumed that the redundant valve in the same line had failed in the open position. These assumptions resulted in conservatively high pressure drops during the closure cycle of the operable valve.

Refer to the responses to Items 3 and 5 for further discussions on the installed configuration and the calculated pressure drops.

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?

RESPONSE:

a. Dynamic torque factors used in butterfly valve sizing were developed from test data obtained from models with similar disc configurations and flow characteristics. The dimensionless aspect ratio (defined as the ratio of the disc diameter to the thickness) was judged to be a significant parameter for evaluation of dynamic torques at various opening angles. Therefore, a series of water flow tests was conducted with a group of 4-inch and 6-inch butterfly valve models constructed with various aspect ratios, ranging from 3:1 to 14:1 (such as 3:1, 8:1, 11:1, and 14:1), in various disc configurations (conventional, offset, cammed), and in both flow directions.

The tests were conducted using the Fluid Controls Institute (FCI) specifications for test arrangement and conduct, per FCI paper 58-2.

The basis followed by the vendor (Fisher) in using incompressible (water) flow model tests to establish dynamic torque coefficients applicable to large diameter valves in compressible flow service is presented in the ISA Transactions article provided as Attachment 1 ("Effects of Fluid Compressibility on Torque in Butterfly Valves" by Floyd P. Harthun).

1. On page 282 of the referenced paper, a relationship is developed relating torque to the nominal valve diameter and pressure drop for incompressible flow conditions, namely:

   $T_D = K_1 D^3 \Delta P$ (equation 10-A)

   in which K_1 is a dimensionless torque coefficient for incompressible flow. It may be noted that this relationship applies to various valve sizes as well as to various pressure drops. The K_1 value is determined by test (using water) for various rotation angles as shown in Figure 1, page 283, of the ISA paper.

2. The referenced ISA paper then considers dynamic torques developed during compressible fluid flow. In this case, the actual torque is no longer linear with respect to ΔP (incompressible flow torque has a linear relationship with respect to pressure drop

RESPONSE TO ITEM 2 CONTINUED

across the valve up to the choked flow condition); for convenience, however, a similar relationship is developed using a ΔP_e factor as follows:

$T_D = K_1D^3\Delta P_e$ (equation 25)

This relationship (equation 25) permits determination of dynamic torques for various valve sizes under conditions of compressible flow, providing K_1 and ΔP_e are determined.

3. Figure 5 on page 284 summarizes the rationale presented in the ISA paper, this figure shows a plot of calculated and experimental torque values for various pressure drops and flow conditions. The straight line labeled "Incompressible Flow" is a representation of the relationship: $T_D = K_1D^3\Delta P$, while the curved line represents compressible flow torque values, following the relationship: $T_D = K_1D^3\Delta P_e$. Test data for compressible flow are also plotted, showing close agreement with the calculated values.

4. Figures 6, 7, and 8 on page 285 of the ISA paper show a comparison of test data and calculated data for various angles of rotation. Close agreement with ΔP_e torque values for compressible flow is indicated. (Note: There are a few typographical errors in the ISA document figures: the $\Delta P/P_1$ value given in the caption to Figure 7 should be 0.155, and in the caption to Figure 8 this value should be 0.466.)

b. Application of the conclusions from the ISA paper is presented in Attachment 2, Selection Figures 1 and 2: "Torque-Pressure Drop Relationships Used in Dynamic Torque Coefficient Selection," which are representations of the Figure 5 curves from the ISA paper. These show how the torque values for compressible flow are conservatively determined and related to incompressible flow torques. It should be noted that the compressible flow curve reaches a critical flow condition at larger ΔP values, resulting in a maximum torque value (T_c) that cannot be exceeded, regardless of how large ΔP becomes.

1. A ΔP_{eff} value is obtained from the Fisher sizing tables based on the absolute inlet pressure value, type of disc, and flowing fluid.

2. If the actual ΔP is greater than ΔP_{eff}, then the ΔP_{eff} value is used, corresponding to T_c (the maximum possible torque). This is conservative, predicting a higher torque than is actually present (see Figure 1 in Attachment 2).

3. If the actual ΔP is less than ΔP_{eff}, then ΔP_{act} is used, which predicts a more conservative (greater) torque, but less than T_c (see Figure 2 in Attachment 2).

RESPONSE TO ITEM 2 CONTINUED

4. An actuator is selected, based on the predicted maximum torque requirements. The actuator must be able to provide the necessary torque to open, close, or maintain the valve in position.

c. Summarizing our previous statement, dynamic torque sizing equations (equations 10-A and 25) were developed relating torque to the nominal valve disc diameter, pressure drop, and a constant K_1. Through testing of various valve models, data were obtained for K_1 at incremental angles of opening. Consequently, the scaling method was used to obtain dynamic torque for larger size valves.

Note: The ΔP_{eff} value (not to be confused with ΔP_e used in the ISA paper) is based on critical flow conditions that will produce the maximum possible dynamic torque.

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?

RESPONSE:

The installation of the containment purge valves is shown in Figure 2. The 36-inch large volume, shutdown purge valves are closed during modes 1, 2, 3, and 4, as required by the plant technical specifications. The 18-inch mini-purge valves on the supply (GT-HZ-04 and GT-HZ-05) and the exhaust (GT-HZ-11 and GT-HZ-12) are designed, tested, and qualified in order to allow them to be open at any time during the plant life.

The mini-purge valves and piping are connected to and are part of the containment penetration for the 36-inch shutdown purge system. Plant arrangements and available space have dictated the arrangements shown in Figure 2. It was not possible at the time of the initial routing to provide separate penetrations or straight piping on either side of the mini-purge valves or between them.

Asymmetric flow effects are known to be a concern to the NRC and industry. Significant margins in available torque have been provided to ensure that the valves will close as required in the installed configuration.

As shown in Figure 2, the valves inside the containment have elbows either upstream or downstream of the valves. Valve GT-HZ-05 has a 45-degree elbow upstream and valve GT-HZ-11 has a 90-degree elbow downstream. Any asymmetric flow conditions developed due to these elbows will tend to assist valve closure. Refer to Figure 2 for the direction of flow and valve closure. The installation of all mini-purge valves is similar in that the valve shafts are vertical and the Bettis actuators are installed below the piping with the common actuator shaft (connecting the air side and spring side actuators) parallel to the piping. The actuator shaft drives a scotch yoke which is attached to the valve shaft. The valve shaft (2-inch diameter) penetrates the center of the valve disc (~4-1/2 inches thick) and is completely enclosed by the valve disc. The shaft/disc arrangement is symmetrical.

The valves located outside of the containment (GT-HZ-4 and GT-HZ-12) are located at the end of 36-inch × 18-inch tees. The flow into the valve will be highly turbulent and well mixed; however, if the flow is postulated to flow predominantly to the outside of the piping, the asymmetric flow would tend to resist valve closure. The asymmetric flow is not expected to be as significant for flow exiting the 36-inch × 18-inch tee as it would be for an 18-inch 90-degree elbow.

The effects of asymmetric flow on the dynamic torque coefficients have not been quantitatively analyzed. However, should the dynamic torque

RESPONSE TO ITEM 3 CONTINUED

required double, as suggested by the NRC as a worst case*, significant margins would still remain between the available torque from the actuator and the required torque to close the valves outside of the containment. As shown in Figure 5 and Table 1, the excess torque available ratios for the design case range from 7.0 to 14.5 in the open positions. Should the dynamic torques required double, a margin ratio of 3.5 to 7.2 would still remain between the available torque and the required torque to close the isolation valves outside of the containment.

It should be noted that there is no credible failure mode for the valves once the valve receives a signal to close and the solenoid valve vents the small quantity of air holding the Versa valve in position. The Versa valve then opens to vent the air cylinder through a 3/4-inch opening. However, the stated required torques are based on only one valve closing. Once a valve starts to close, the line losses and the pressure drop through the redundant valve become minimal. These reduced flow conditions would help ensure that the redundant valve would close even more rapidly than the required design value.

Also, the required torques have additional margins in that they are calculated based on a 22.9 psig drop across the valve at all positions. Since the containment pressure will only be 14 psig maximum when the valve receives its closure signal, the pressure drop and the required torque will be significantly less than those calculated. Similarly, the pressurization transient shown in Figure 1 is for a worst case large LOCA and has many conservatisms built into the analyses.

Another noteworthy conservatism exists in the reported dynamic torque requirements shown in Table 1. The maximum torque required to open or close the valve in the specified position is listed. For the 9200 series valves, flow generally tends to close the valves for straight line installations. The maximum torques listed are those required to open the valve. The torque required to close the valve is less.

The actuator torque output ratio to torque required for case 2, the peak calculated containment pressure, varies from 4.0 to 7.9 for all open angle positions (refer to Table 1). Therefore, even if it is postulated that the inside containment valves fail to receive a signal to close and the closure of the outside valve is delayed until the peak pressure is reached inside the containment, excess torque available ratios of 2.0 to 4.0 would still exist for the closure of the outside containment isolation valves should the dynamic torque required double due to assymmetric flow considerations.

In summary, the dynamic torque requirements do not include asymmetric flow considerations; however, significant conservatisms are provided in all aspects of the design and analyses to ensure that the valves will close as required in the installed configuration.

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*See item 3 of NRC paper entitled "Operability Qualification of Purge and Vent Valves."

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. 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 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.

RESPONSE:

As noted in the response to Item 1 and as shown in Figure 1, the containment pressure during the closure cycle was assumed to be 22.9 psig, which corresponds to the time at which the valve is fully closed (T = 6 seconds). The lag time between the receipt of the signal to close at 3 seconds and the initiation of valve motion has been taken into account. Tests have ensured that closure is accomplished within the required 3 seconds from receipt of the closure signal.

Since the air cylinder is vented through the large Versa valve opening (3/4-inch diameter), the air cylinder will depressurize rapidly. As shown in Figure 3, approximately 80 psig in the air cylinder is required to balance the spring force and maintain the actuator/valve in the open position without flow. A pressure regulator is provided on the valve to allow a reduced pressure in the air cylinder. This regulator will be set to ensure that the valve remains fully opened. Depressurization of the air cylinder with respect to time is shown in Figure 4. The times required to vent the cylinder from any initial pressure less than 100 psig can be taken from Figure 4. For example, depressurization from 100 to 80 psig is accomplished in approximately 0.1 second when venting to atmospheric pressure. As noted in the response to Item 9, venting of the accumulators to a positive containment pressure will not affect the closure time.

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

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

RESPONSE:

Table 1 - Case 1 provides the calculated pressure drops across the valve for each 10 degrees of valve position. These pressure drops were calculated based on the installed piping system resistances, assuming that the redundant purge valve in the same line had failed in the open (least resistance) position. The predicted pressure drops are all calculated at 22.9 psig, which is the maximum pressure that would exist prior to valve closure. Data also presented for Case 1 in Table 1 include the actuator torque available and the ratio of torque available to torque required at the corresponding positions. The ratio of excess torque varies from 7.0 to 14.5 for all opening positions.

Table 1 - Case 2 provides data for the worst possible case wherein the valve closure is assumed to be delayed until the peak calculated containment pressure of 47.2 psig is attained. This is not a design case, but is provided to demonstrate the large margins available to ensure valve closure. As can be seen from the data provided, excess torque margins of 4.0 to 7.9 exist even for this worst case. As with the design case, all pressure drops are calculated with 47.2 psig at the inlet to the purge line and the redundant valve in the line failed in the open position.

The available actuator torques and the required torque at the valve for each 10 degrees of opening are shown in Figure 5.

Stresses on valve components have been calculated for both cases, and none of the components exceed the allowable stress values reported in the response to Item 6.

The specific maximum ΔP capability has not been calculated for containment pressures greater than 47.2 psig since these pressures will not exist. However, it can be concluded from the excess torque available ratios and the stress analyses that the valves would close at much greater differential pressures than those to which the valves could be exposed.

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?

RESPONSE:

Type 9220 butterfly valves are designed according to the ASME Boiler and Pressure Vessel Code, Sections III and VIII. Allowable stresses are also taken from the ASME Boiler and Pressure Vessel Code. Loads considered in the design of these valves include all typical pressure- and flow-induced loads. Worst case load combinations were used. The design is compatible with ANSI Class 150 pressure/temperature ratings. These valve bodies are BWE × flanged style to mate with ANSI Class 150 (B16.5) raised-face flanges.

A summary of critical loading is presented as follows:

Stress Consideration	Allowable* Stress** ksi	Calculated Stresses, ksi Closed†	Case 1†† at 70 Degrees	Case 2†† at 70 Degrees
1. Shaft at disc hub (1.5S) (bending and torsion)	52.5	4.7	2.3	4.3
2. Shaft at disc hub (0.75S) (torsion and transverse shear)	26.25	6.1	3.1	5.7
3. Shaft at pin connection (0.75S)	26.25	5.8	3.7	6.3
4. Shaft at key connection (0.75S)	26.25	5.3	3.1	5.4
5. Bushing	8.5§	1.9	0.7	1.5

NOTES

*These allowables were derated to 98 percent of that shown to account for the 320 F design temperature.

**Based on ASME Code Section III values (S) of 35 ksi for 17-4PH, Condition H1100 (Table I-7.1, Appendix I). The allowable stress of 35 ksi is a conservative figure since an (S) value of 36.2 ksi is allowed by the code for H1075 shaft material.

†Closed position stresses based on 60 psid across valve.

††Refer to Table 1 for the definitions for cases 1 and 2. Stresses are reported for 70 degrees open since they are the maximum for opening angles 10 through 90 degrees.

§Graphite-filled bronze.

The shaft is considered to be the most critical valve component under most conditions, since the pins and keys are selected to be stronger than the shaft. Therefore, separate calculations for pins and keys are not necessary. Stress concentration factors are considered when evaluating

RESPONSE TO ITEM 6 CONTINUED

the shafts at the pins and keyways. The maximum disc load occurs when the disc is in the closed condition, and acceptable disc strength values have been established based on testing and experience.

As noted in the FSAR, the reactor coolant pressure boundary is Seismic Category I and is not postulated to fail during a seismic event. The LOCA which results in the pressure transient shown in Figure 1 is not based on a mechanistic failure but is postulated to occur as required by 10 CFR 50 Appendix A. The stresses reported above are based on dynamic loadings due to the LOCA pressurization transient. Shaft loadings which result from a seismic event are not specifically calculated or combined with LOCA loadings because the events are independent and not postulated to occur simultaneously. The purge valves are Seismic Category I and have been tested for operability during and after a seismic event.

As can be seen from the LOCA-induced stresses reported above, significant margins exist (factor of 7 minimum) between the stress allowables and the calculated stresses for the design case (Case 1). Seismic loadings on the shaft are expected to be less than the LOCA loadings because the actuator loadings are transmitted to the valve body.

In summary, the valve is designed for independant LOCA and seismic events. However, due to the margins which exist and the expected response of the valve, it is anticipated that the valve would function properly for the nondesign basis case wherein the events would be postulated to occur simultaneously.

7 and 8. These item numbers were omitted by the NRC.

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.

RESPONSE:

The subject 18-inch valves are equipped with Bettis spring-return actuators. The unpressurized side of the piston actuator is vented to the local ambient conditions. During valve closure the pressure side is also vented (through the Versa valve and solenoid) to the same local ambient conditions; therefore, no pressure differential will exist across the piston as a result of a surrounding local pressure rise upon full depressurization. The spring will still drive the actuator to the safety-mode (closed) position and maintain that position.

Figure 4 shows the depressurization transient for the actuator. As shown therein, the actuator depressurizes very rapidly. Since the unpressurized side is continuously vented to the containment, it will be pressurized along with the containment. There will be no adverse effect on the valve closure rate. In fact, the closure rate may increase due to prepressurization of the vented side.

The solenoid used (ASCO Type NP8320A185V) is a three-way direct-acting solenoid which operates a 3/4-inch three-way pneumatic switching valve (Versa VSP-3601-155H) to minimize restriction and provide a fast closure response. The solenoid is connected in the normally-closed mode so that de-energization will vent the actuator casing to ambient atmosphere through the Versa switching valve.

As shown in Figure 5, adequate spring-driven torque output is available from the actuator to close the valve from any open position (regardless of external ambient pressure). Since both the unpressurized side of the piston casing and the solenoid and Versa valves are vented to the same external ambient pressure, there is no effect on valve closure.

Note: There are no four-way solenoids or switching valves used in the subject valve assembly.

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

RESPONSE:

This item is not applicable to SNUPPS. The closure of the valve is accomplished by spring force, not by air pressure.

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

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

RESPONSE:

This item is not applicable to SNUPPS. Inflatable seals are not used.

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.

RESPONSE:

The first part of this item is not applicable to SNUPPS because no modifications have been made to limit the valve open position. Valve seating and ambient temperature effects are addressed as follows.

The purge supply air temperature is tempered to a minimum of 50 F. The exhaust air will be that of the bulk containment (120 F maximum).

Seating torque is considered to be independent of temperature, within the calculated flow temperature range of 50 F initial (minimum) temperature to 231 F maximum fluid temperature during closure. The 320 F containment design temperature would not be experienced by the T-ring seal during the closure transient. The initial effect of a high temperature would be to soften the material, but it would also expand (compensating effects). In any event, however, the actuator spring-driven torque capability at closure (approximately 29,900 in.-lb) is so much greater than the required seating torque at 60 psid differential (approximately 6220 in.-lb) that complete closure is assured. If it is postulated that seal expansion prevents complete rotation, the disc would still be driven into the seal far enough to provide the required shutoff, regardless of final travel position.

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

RESPONSE:

The required torque at the valve and the actuator-supplied torque are shown in Figure 5. No concerns exist that could affect valve operability.

14. Has the maximum torque value determined in #13 been found to be compatible with torque limiting settings where applicable?

RESPONSE:

This item is not applicable to SNUPPS.

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.

RESPONSE:

This item is not applicable to SNUPPS.

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.

RESPONSE:

This item is not applicable to SNUPPS.

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

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

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

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

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

RESPONSE TO ITEMS 17 THROUGH 21:

The subject valves have been qualified in accordance with NUREG 0588 and the requirements of IEEE 323-1974. The qualification programs have been fully described in the SNUPPS submittals to the NRC (Letter SLNRC 83-0015 dated March 10, 1983), which are contained in the two-volume set entitled "Report of Independent Review of Environmental Qualification Programs to NUREG 0588." The specific section of that report which addresses the purge valves is found in Volume 2 under Specification M-237.

22. Assess the structural capability of any ducting or piping in the purge system which is upstream or downstream of the valves and is exposed to the flow condition associated with the LOCA and seismic event. In particular, consider the effects of loose debris from the pipe or duct system on the closure capability of these valves.

RESPONSE:

The valves are located in piping which is ASME Section III, Class 2. The boundary of this piping is extended to seismic anchors as recommended by Position C.3 of Regulatory Guide 1.29.

As shown on FSAR Figure 9.4-6, Sheet 4, debris screens are provided on the containment side of the mini-purge supply and exhaust isolation valves to prevent the entry of lightweight debris, which could preclude tight valve closure. The piping which contains the screens is ANSI B31.1 (150-pound design pressure) piping, which is seismically analyzed in accordance with Position C.3 of Regulatory Guide 1.29. The screens are located approximately two pipe diameters away from the isolation valves and are inherently designed to withstand post-LOCA differential pressures due to their rugged design and the negligible pressure drop through the screen material (No. 2 mesh, .063-inch wire with a 76.4-percent free area). The screen material is welded over the 17-inch-diameter opening in a 1/4-inch-thick flange which is bolted into place.

The purge isolation valves and debris screens are located adjacent to the containment wall, outside of the secondary shield walls, and are protected from missiles which could be postulated following a LOCA. Also, motor-operated dampers are located one pipe diameter away from the screens on the containment side. These dampers and the connecting piping provide additional protection for the wire mesh screens.

TABLE 1

COMPARISON OF TORQUE AVAILABLE TO TORQUE REQUIRED

		CASE 1 Closure During Transient Design Case			CASE 2 Closure at Peak Calculated Pressure		
Angle of Opening	Torque Available in.-lb	Maximum* Predicted ΔP	Torque Required at ΔP = 22.9 in.-lb	Ratio Avail./Req. Torque	Maximum Predicted ΔP	Torque Required at ΔP = 47.2 in.-lb	Ratio Avail./Req. Torque
Closed	29,900	22.9	6,220**	4.8†	47.2	6,220**	4.8†
10	25,600	22.9	1,785	14.5	47.2	3,239	7.9
20	23,000	22.8	2,223	10.4	47.2	3,961	5.8
30	21,700	22.7	2,223	9.8	46.8	3,961	5.5
40	21,400	22.2	2,223	9.6	45.8	3,961	5.4
50	22,100	21.0	2,728	8.0	44.0	4,791	4.6
60	23,900	18.2	3,412	7.0	39.2	5,918	4.0
70	27,200	12.8	3,662	7.4	30.8	6,330	4.3
80	33,000	7.3	3,581	9.2	16.1	6,197	5.3
90	43,000	4.3	3,581	12.0	9.9	6,197	6.9

*During the 3-second closure period, the containment pressure rises from 14 psig to 22.9 psig. All predicted ΔPs are based on flow conditions at 22.9 psig at the inlet to the purge piping.

**The 6,220 in.-lb torque required based on 60 psig, which is the containment/valve design pressure. The maximum calculated LOCA pressure is 47.2 psig.

†See Figure 5.

Figure 1

CONTAINMENT PRESSURE TRANSIENT WITH PURGE VALVE CLOSURE TIMES

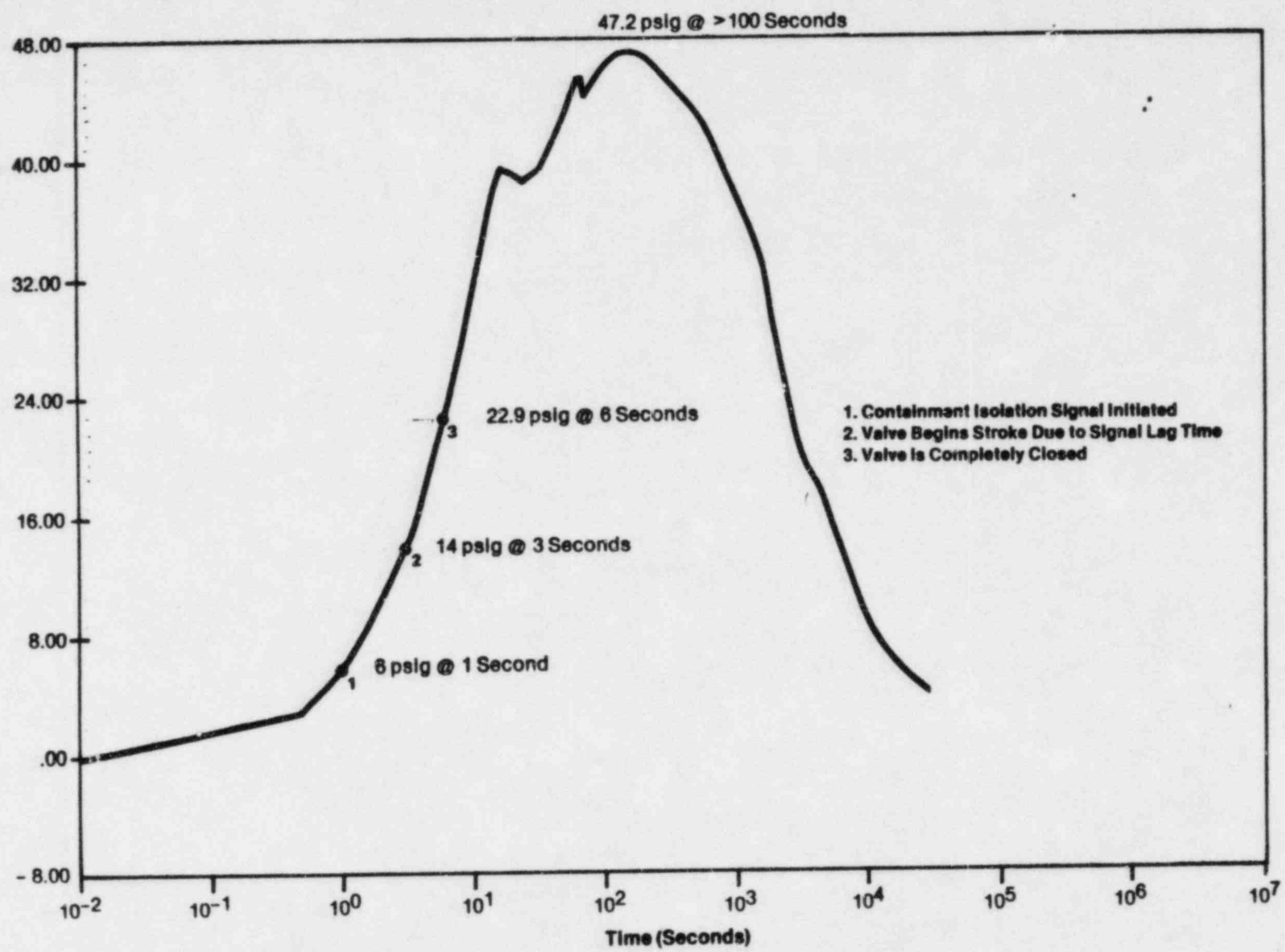


Figure 2
(Sheet 1 of 2)

LOCATION OF VALVES IN CONTAINMENT PURGE SYSTEM

(Supply Line)

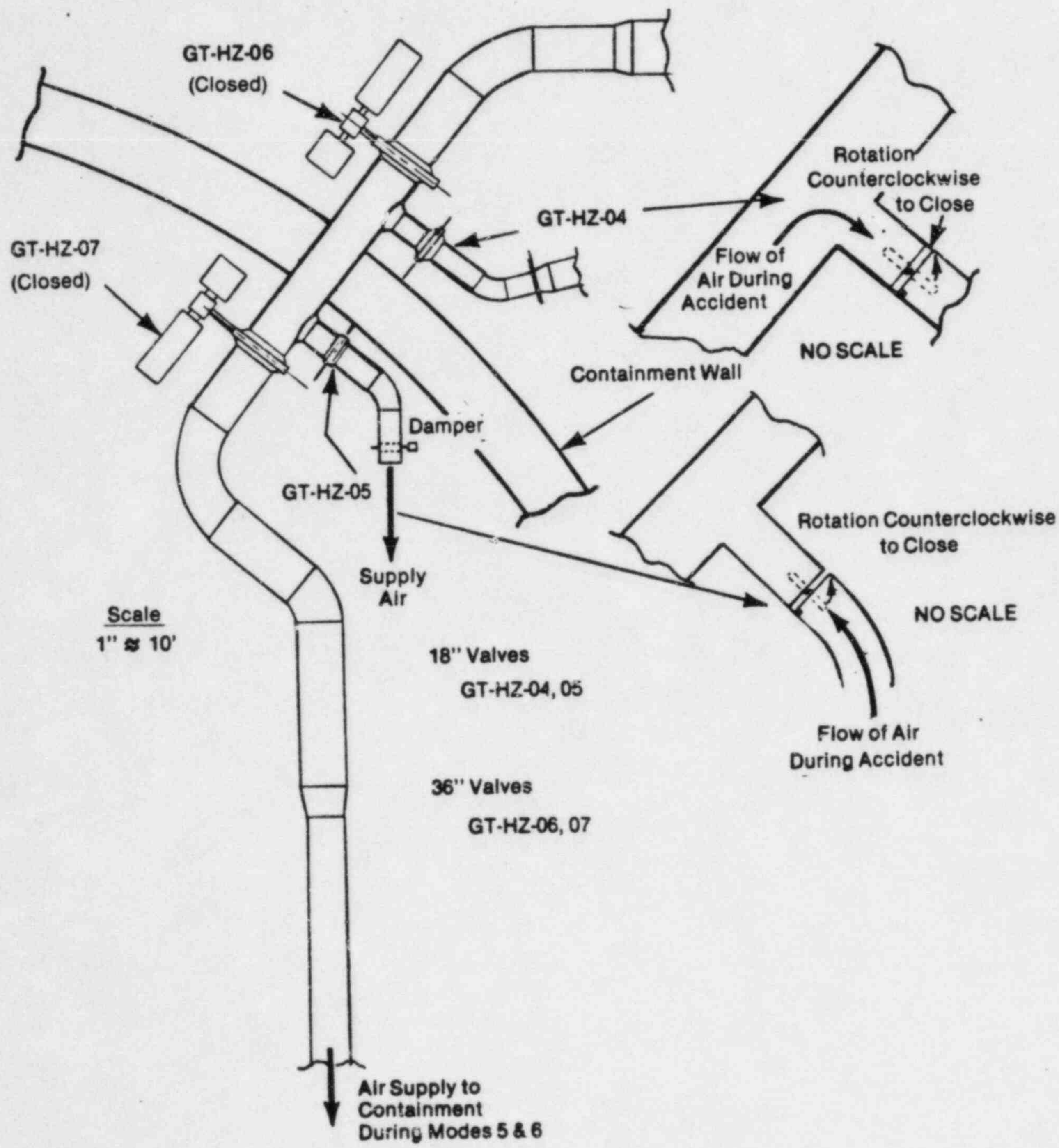


Figure 2
(Sheet 2 of 2)

LOCATION OF VALVES IN CONTAINMENT PURGE SYSTEM

(Exhaust Line)

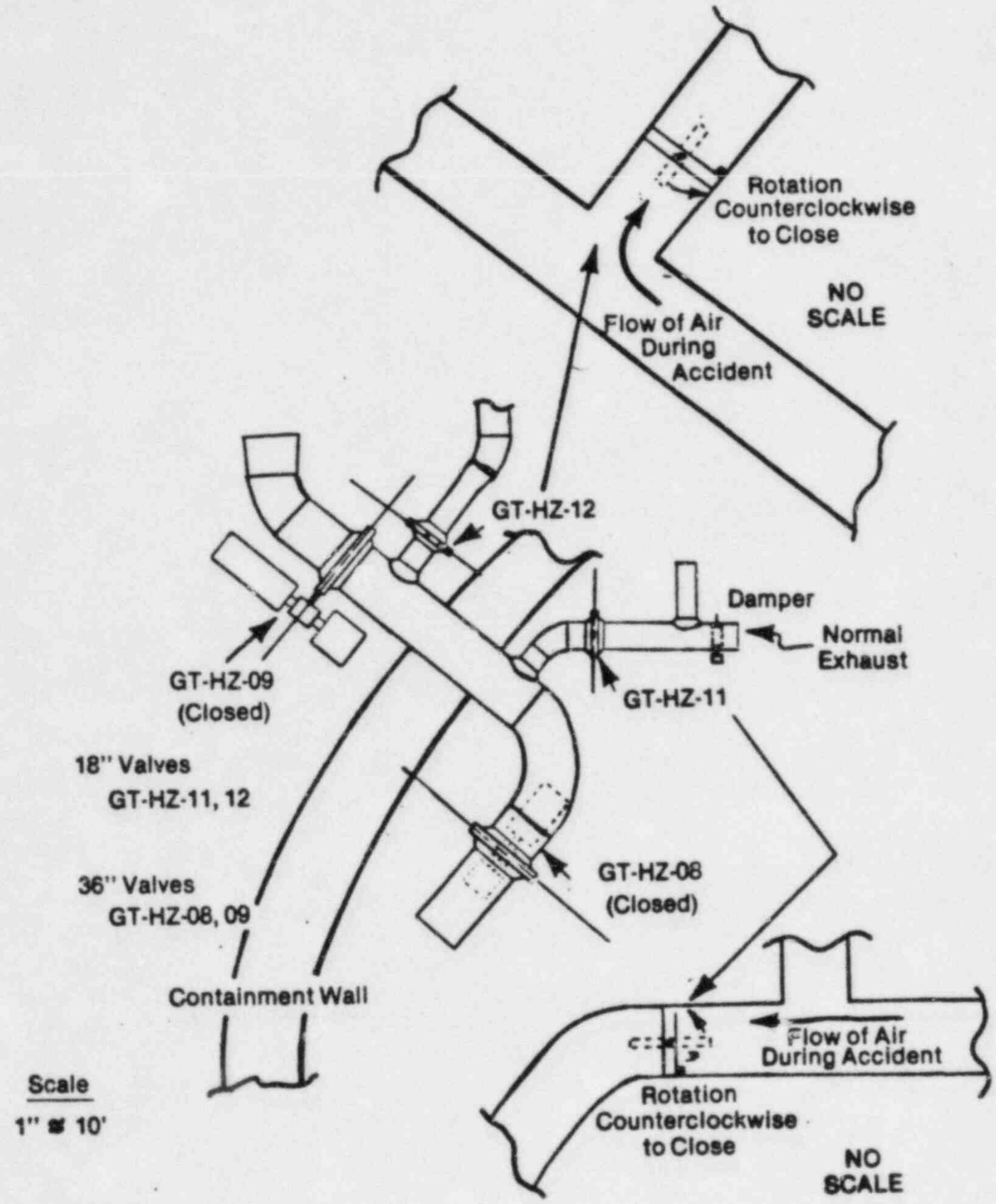
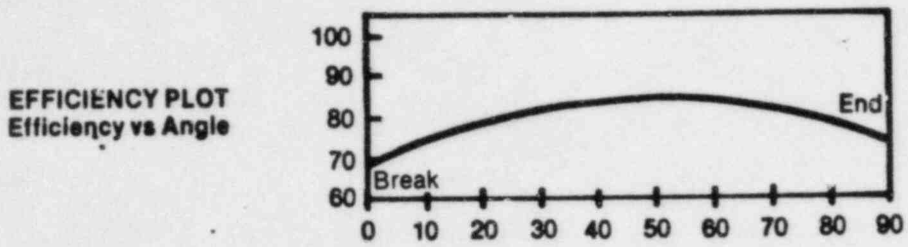


Figure 3

ACTUATOR DATA — SCOTCH YOKE



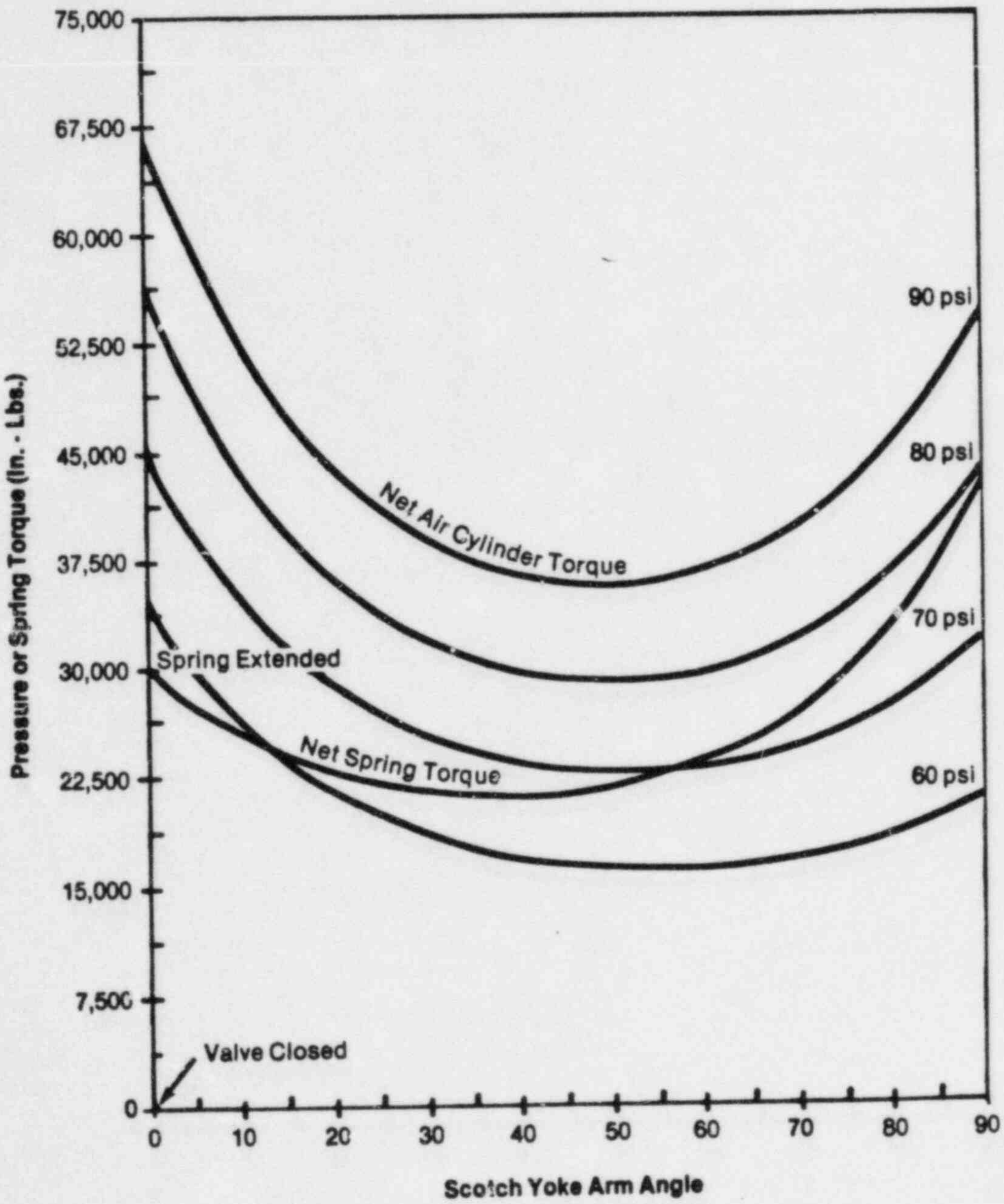


Figure 4

ACTUATOR DEPRESSURIZATION TRANSIENT

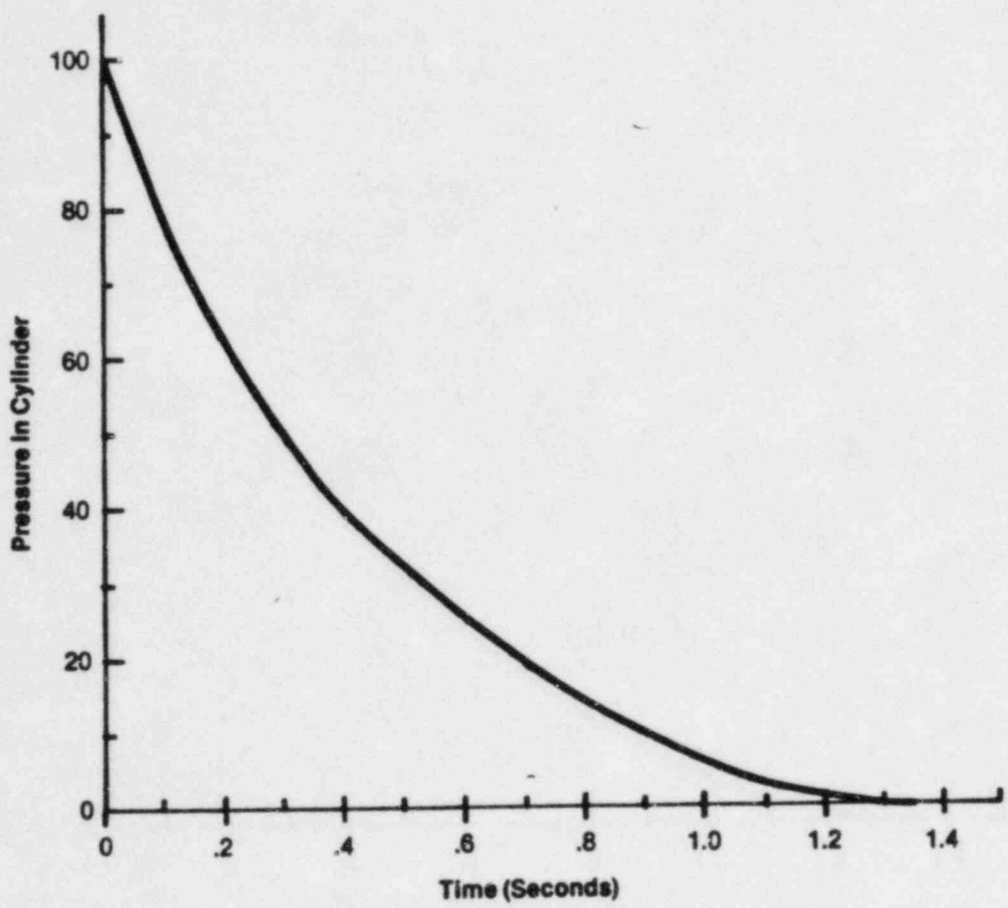
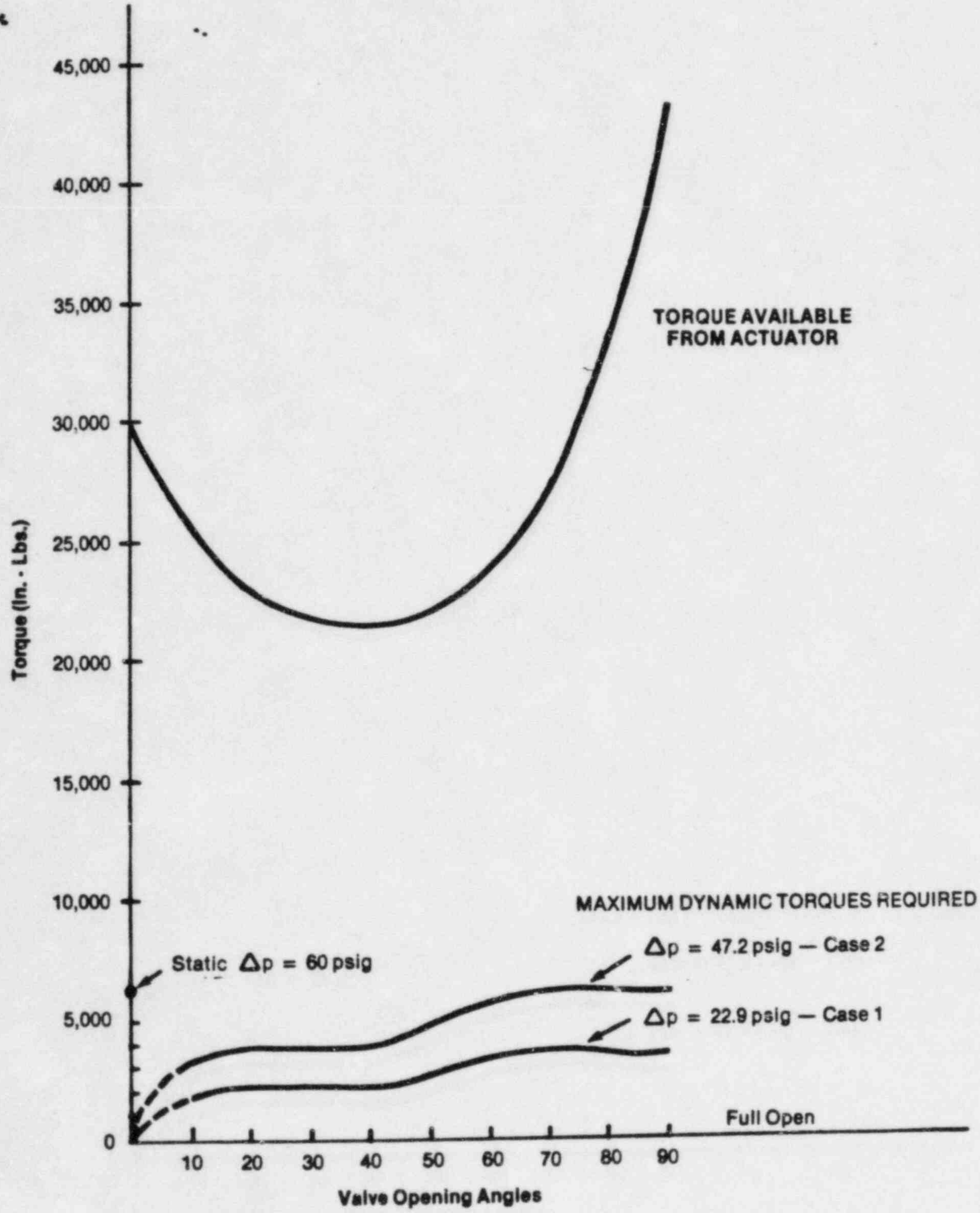


Figure 5

COMPARISON OF AVAILABLE ACTUATOR TORQUE TO REQUIRED TORQUE AT VALVE



ATTACHMENT 1

to

PURGE VALVE OPERABILITY REPORT

ISA Transactions, Vol. 8, No. 4, 1969

"Effects of Fluid Compressibility on Torque in Butterfly Valves"

by

Floyd P. Harthun

ISA TRANSACTIONS: 8 (281–288) 1969

Effect of Fluid Compressibility on Torque in Butterfly Valves*

FLOYD P. HARTHUN†
Fisher Governor Company
Marshalltown, Iowa

▶ 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

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

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

$$\Delta P/P_1 \to 0$$

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

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

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.

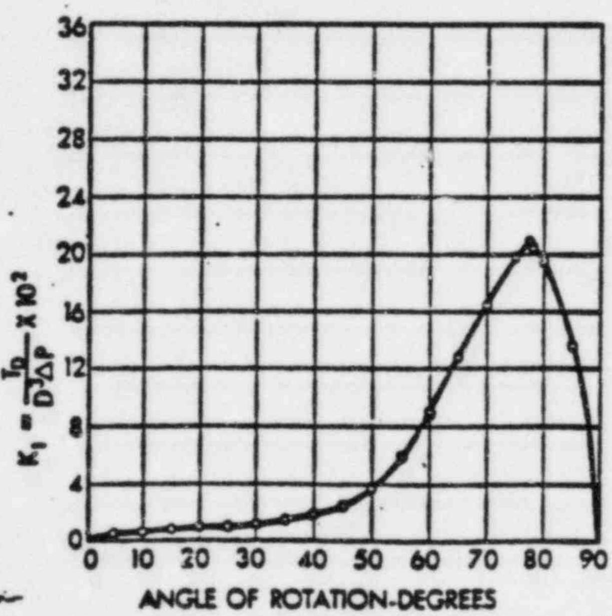


Figure 1. Dimensionless torque coefficient, 4-in. butterfly valve incompressible flow: P_1 = 100 psig, P = 110 psi.

Determined by test

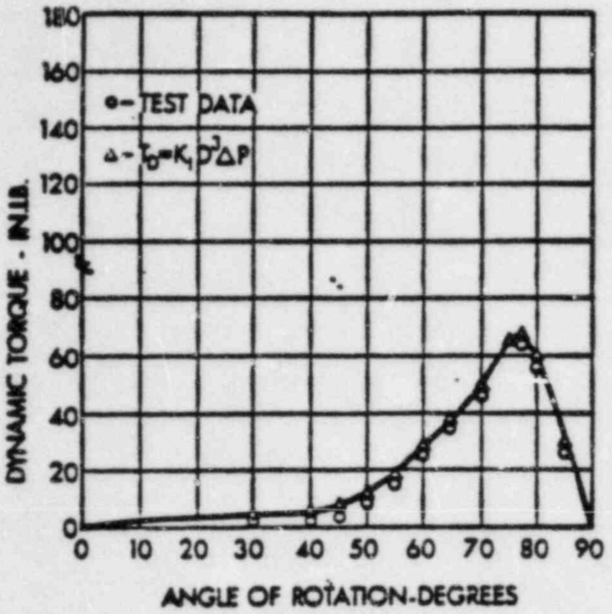


Figure 2. Dynamic torque vs. angle of disc rotation, 4-in. butterfly valve, comparison of experimental results with calculated torque, incompressible flow: P_1 = 100 psig, ΔP = 5 psi.

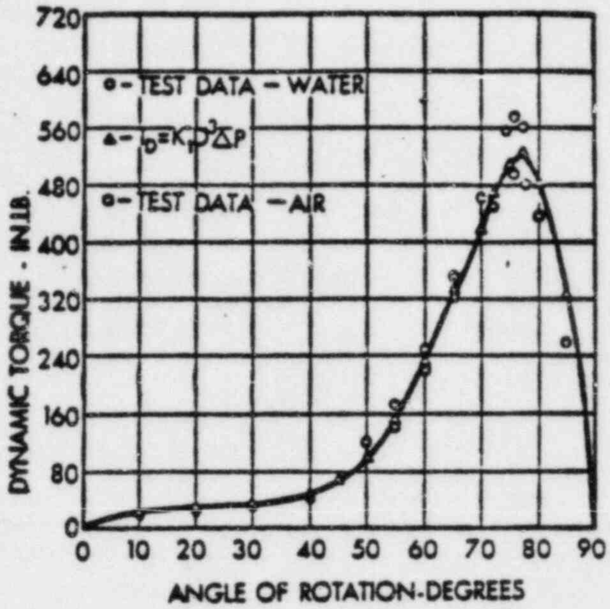


Figure 4. Dynamic torque vs. angle of disc rotation, 8-in. butterfly valve, comparison of experimental results with calculated torque, incompressible flow: P_1 = 100 psig, ΔP = 5 psi.

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.

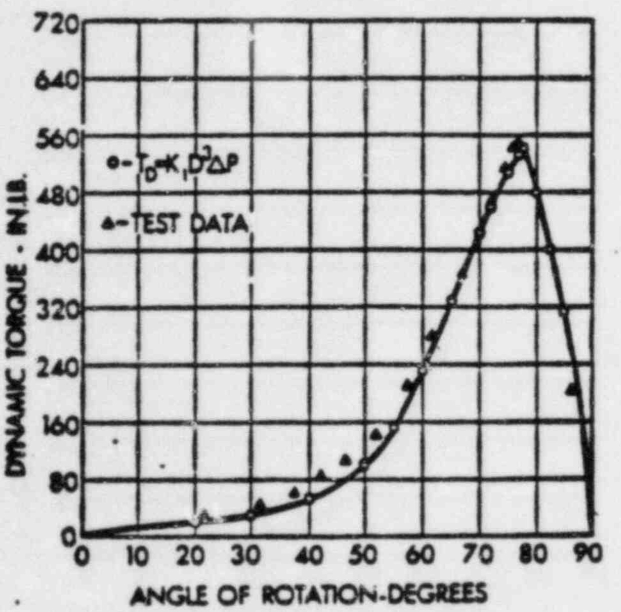


Figure 3. Dynamic torque vs. angle of disc rotation, 8-in. butterfly valve, comparison of experimental results with calculated torque, incompressible flow: P_1 = 100 psig, ΔP = 5 psi.

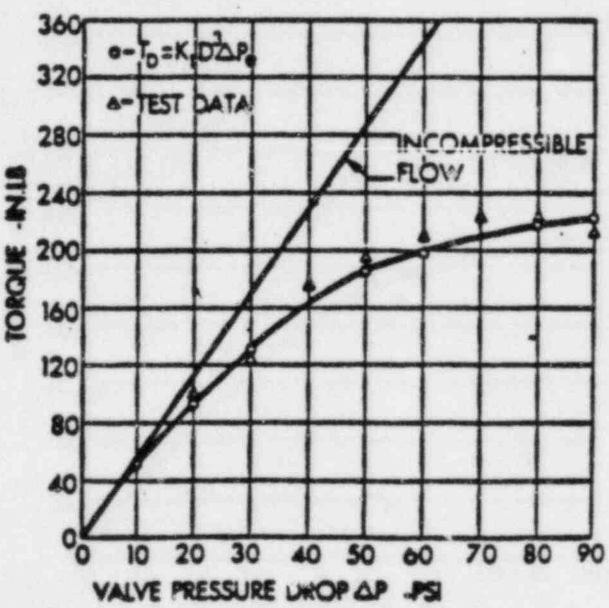


Figure 5. Dynamic torque vs. valve pressure drop, 4-in. butterfly valve, 60° disc rotation, comparison of experimental results with calculated torque, compressible flow: P_1 = 214.4 psia, flowing medium = air.

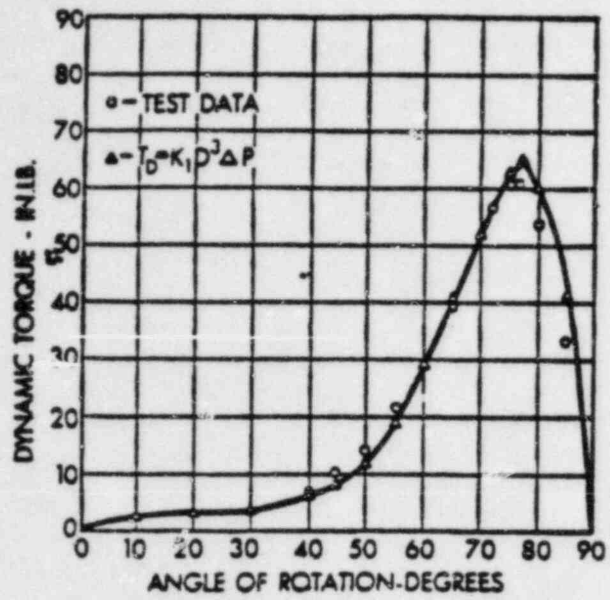


Figure 6. Dynamic torque vs. angle of disc rotation, 4-in. butterfly valve, comparison of experimental results with calculated torque, compressible flow: P_1 = 114.4 psia, ΔP = 5 psi ($\Delta P/P_1$ = 0.0446), flowing medium = air.

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

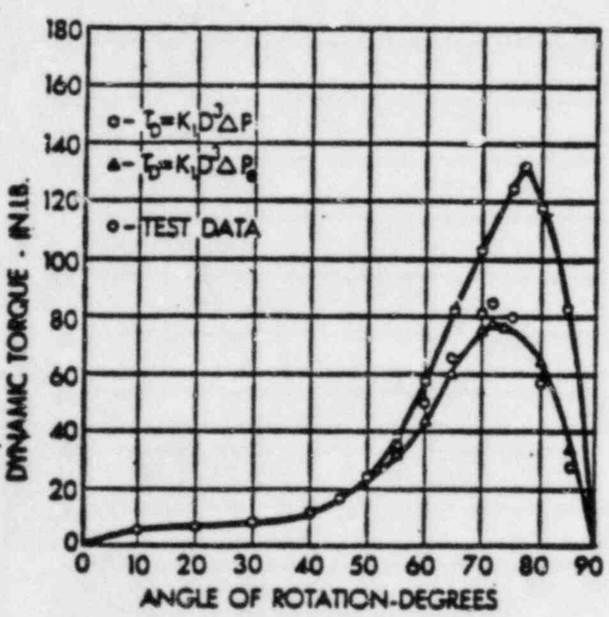


Figure 7. Dynamic torque vs. angle of disc rotation, 4-in. butterfly valve, comparison of experimental results with calculated torque, compressible flow: P_1 = 64.4 psia, ΔP = 10 psi ($\Delta P/P_1$ = 0.115), flowing medium = air.

0.155

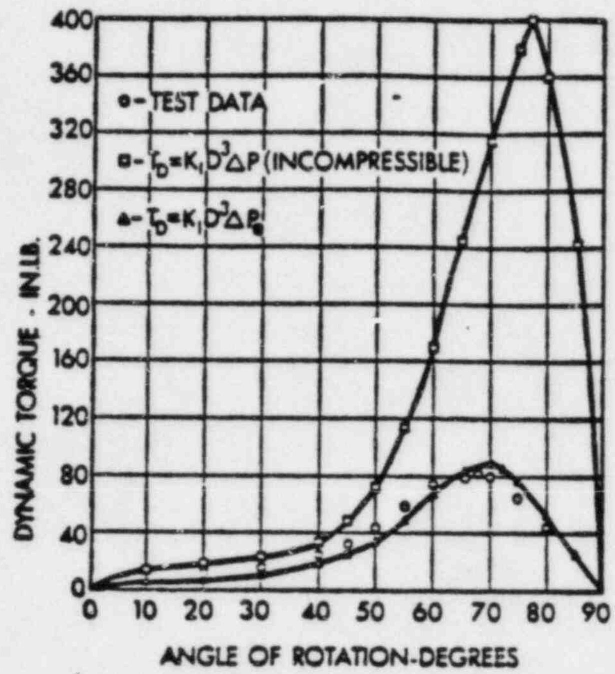


Figure 8. Dynamic torque vs. angle of disc rotation, 4-in. butterfly valve, comparison of test results with calculated torque, compressible flow: P_1 = 64.4 psia, ΔP = 30 psi ($\Delta P/P_1$ = 0.466) (critical flow) flowing medium = air.

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_6 = 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
D C = Nominal valve diameter, in.
F = Force, lb
G = Specific gravity
K_1 = 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_e = 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}

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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 2

to

PURGE VALVE OPERABILITY REPORT

<u>Torque - Pressure Drop Relationships used in</u>
<u>Butterfly Valve Dynamic Torque Coefficient Selection</u>

a. Selection Figure 1
b. Selection Figure 2

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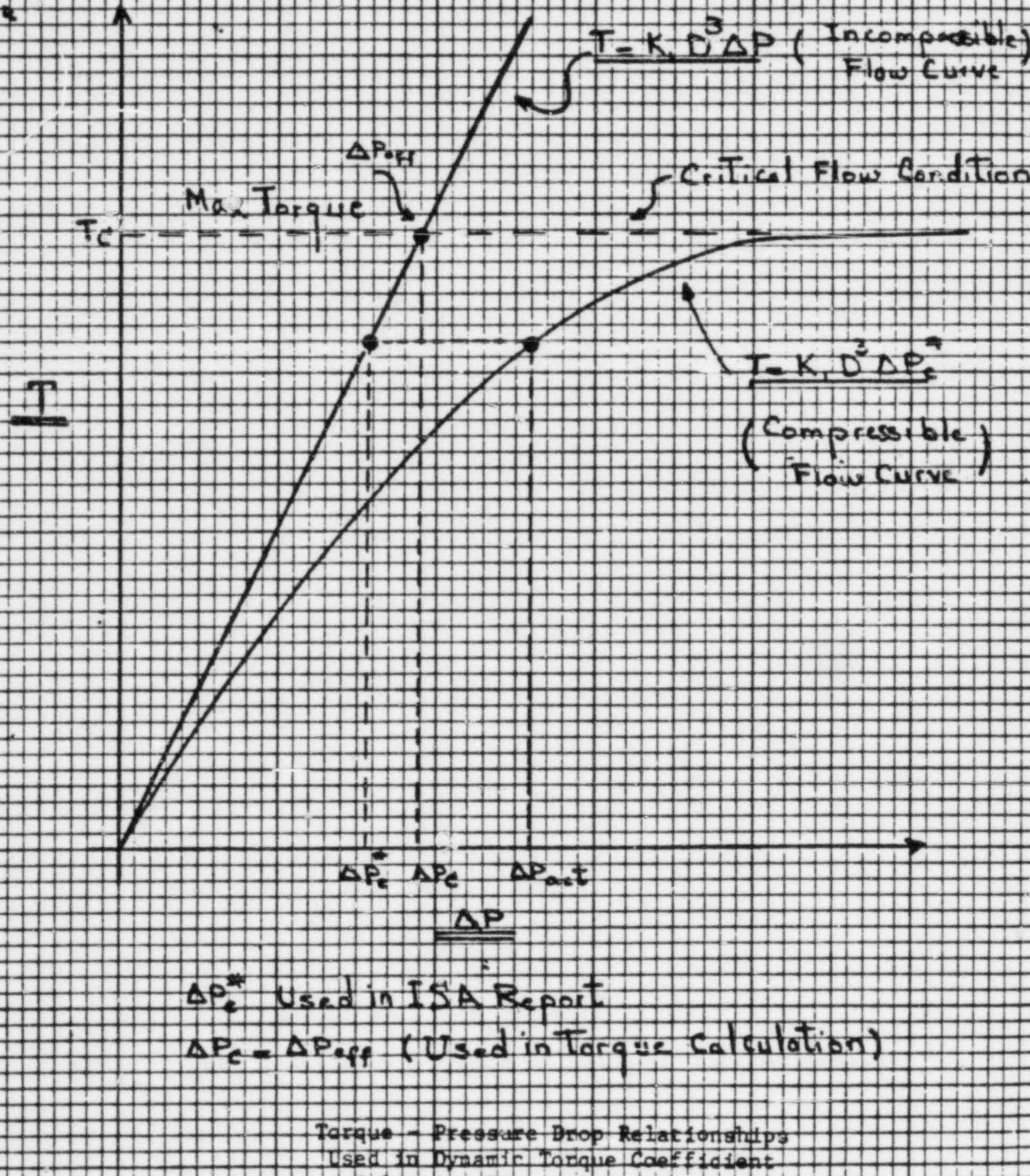
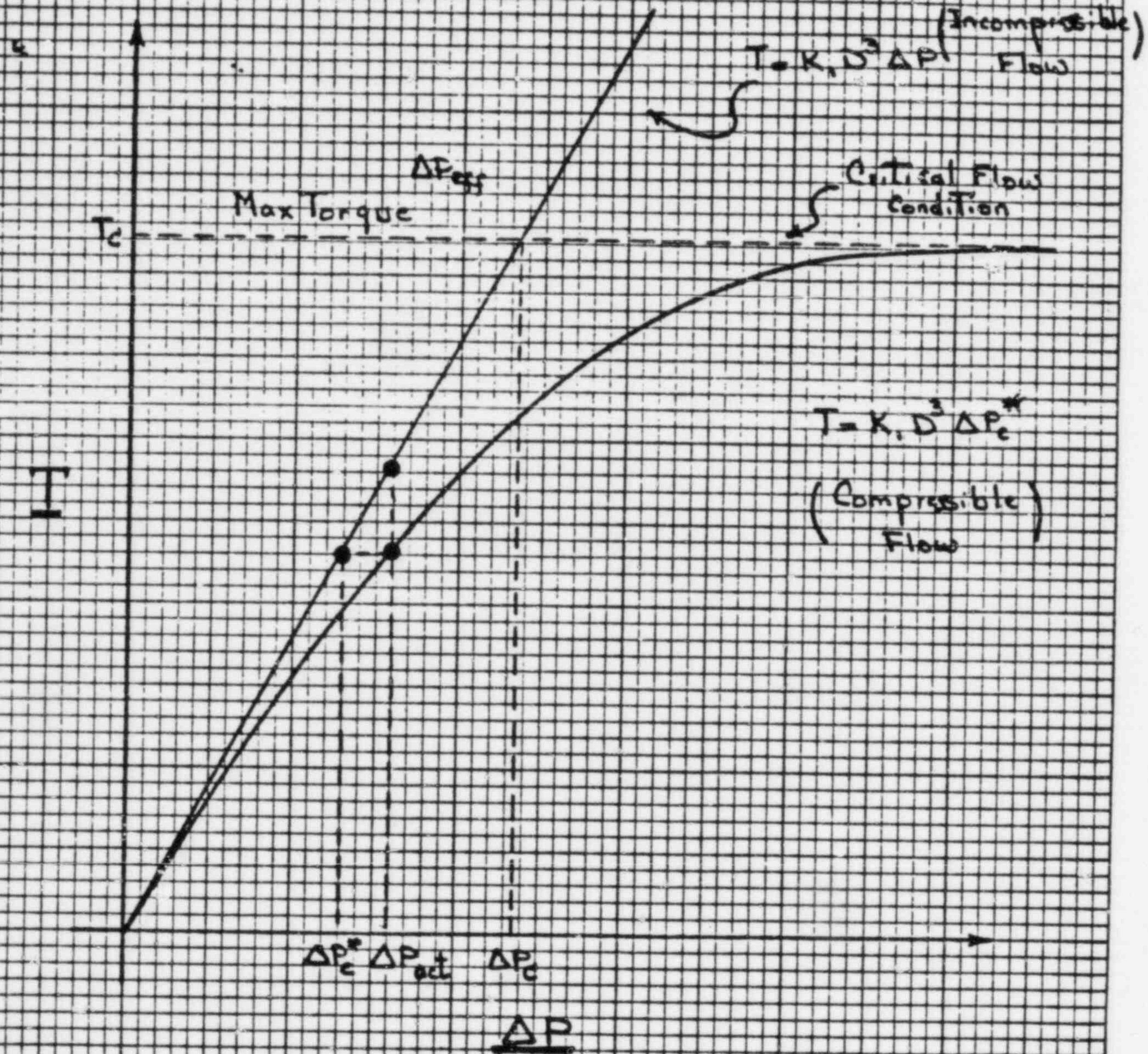


ΔP_c^* Used in ISA Report

$\Delta P_c = \Delta P_{eff}$ (Used in Torque Calculation)

Torque - Pressure Drop Relationships
Used in Dynamic Torque Coefficient
Selection Figure 1



ΔP_c^* Used in I.S.A Report

$\Delta P_{act} < \Delta P_c = \Delta P_{eff}$ (ΔP_{act} Used in Torque Calculation)

Torque - Pressure Drop Relationships
Used in Dynamic Torque Coefficient
Selection Figure 2

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