Butterfly Valve Model Improvements based on Compressible Flow Testing Benefit Industry AOV Programs

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Abstract

U. S. Nuclear Regulatory Commission issued a Regulatory Issue Summary 2000-03 recommending that all U. S. nuclear power plants take necessary steps to ensure that all safety related power-operated valves will perform their safety related functions under design basis conditions [1][*]. Power operated valves included a large population of quarter-turn air operated valves (AOVs) and hydraulically operated valves (HOVs) for which industry had no validated models. An earlier paper [2] describes a comprehensive development program undertaken by Kalsi Engineering, Inc., to provide validated quarter-turn valve models needed by the industry. *Phase 1* of this program included a large matrix of *incompressible flow tests* performed on various types of butterfly, ball and plug valves used in AOVs [2]. The present paper describes *Phase 2: The compressible flow test* program that is presently being conducted to validate the new quarter turn valve models. The benefits of the new validated models to actual containment purge and vent valves are also included in the paper.

The new models have significantly advanced the state-of-the-art in accurately predicting torque requirements typically for quarter-turn valve design variations and operating conditions encountered in nuclear power plants.

Background and Introduction

As shown in Figure 1, there is a fundamental difference in the actuator output between the typical electric motor operated valves (MOVs) and air-operated valves (AOVs) that directly affects the margin between the actuator output torque capability and valve torque requirements. Since the output from a typical AC powered MOV actuator is constant throughout the stroke, only the peak required torque magnitude, regardless of the stroke position where it occurs, is important to determine margin throughout the stroke. However, the output of a typical quarter-turn AOV actuator varies with stroke; therefore, the models to predict torque requirements for the valve need to have a *position dependent accuracy* to determine margin at each point of the stroke.

There were two significant programs in the industry that included testing and validated models of butterfly valves for safety-related nuclear power plant MOV applications:

- The NRC/INEL Containment Purge and Vent Valve Test Program [3, 4], and
- EPRI's Butterfly Valve Guide [5] and MOV Performance Prediction Methodology (EPRI MOV PPM) Development Program [6].

Under the NRC/INEL program, three butterfly valves were tested with gaseous nitrogen under blowdown conditions. This testing was limited to single offset disc design, because the NRC survey results showed that this design has the dominant population in the U. S. Nuclear power plants. Symmetric disc, double

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[*] The numbers in brackets denote references listed at the end of this paper.

and triple offset disc designs were not included in this test program. Furthermore, the NRC/INEL program did not include testing of two valves in series.

The EPRI MOV PPM included the development of models for symmetric, single and double offset disc designs. The scope of EPRI test programs, however, was limited to incompressible flow testing of symmetric and single-offset disc designs. Furthermore, the EPRI MOV PPM did not have a specific criteria for position dependent accuracy, as long as the peak dynamic torque requirements were bounding, because the objective of the EPRI program was to address *motor-operated valves*. The EPRI MOV PPM model was validated against the NRC/INEL test data and found to provide bounding predictions of torque requirements [6]. However, as shown in Figure 2, the torque signature predicted by the model was found to be unconservative over certain portions of the stroke. Therefore, EPRI cautions the users not to use the torque signature for determining torque requirements especially for compressible flow applications [6, 8].

In addition to these two major industry programs, there has been some limited amount of compressible flow testing done by other investigators [9, 10, 11]. Although those investigators presented some informative insights, they did not provide models or validated test data for butterfly valves used in nuclear power plant applications.

To overcome the above-described limitations of the NRC/INEL program, the EPRI MOV PPM and other industry programs, Kalsi Engineering, Inc., initiated a new quarter-turn valve model development and testing program. Under Phase 1 of the program (which has been completed) incompressible flow tests were performed on a large number of quarter-turn butterfly, ball and plug valves. Phase 2 of this program covers compressible flow tests being performed on various types of butterfly valves, as described below.

Phase 2: Compressible Flow Test Program

Objectives

The objectives of the Phase 2 Test Program are to test quarter-turn valves under compressible flow to quantify the effect of:

1. Geometry of disc shapes commonly used in nuclear power plant applications on aerodynamic torque;
2. The ratio of the differential pressure and upstream pressure on torque predictions; and
3. Upstream elbows on the torque requirements for valves of different geometries.

Test Matrix

The following parameters were varied in the test matrix:

- Disc geometry: symmetric, single offset, double offset
- Disc orientation for non-symmetric disc valves: shaft upstream, shaft downstream
- Upstream sources pressures (psig): 75, 60, 45, 30, 15, 10, 5
- Pressure ratios (P_{up}/P_{down}): Varied from 1.33 to 6 by throttling the downstream throttle valve (DTV)
- Location of upstream elbow: 0D-8D
- Orientation of upstream elbow

All tests were performed on 6-inch scaled models that represent various disc shapes and aspect ratios of actual valves (Figure 6). The approach of using precisely scaled models to predict performance of large valves has been validated as shown in Figures 4 and 5 [5].

Facility Description

The compressible flow blow down test facility (shown in Figure 3) consists of three 3600 ft^3, 425 psig tanks supplied by a 383 cfm Ingersoll Rand reciprocating-pump air compressor, a 60 feet long 6-inch straight pipe test section with an open field at the discharge. A quick-opening 6-inch 300-lb air-operated ball valve, used to initiate or block the flow, is located at the head of the test section. The flow metering section, located immediately downstream of this block valve, and consists of 20 feet (40D) of straight pipe followed by a Dieterich mass flowmeter. The test valve is located 30 feet (60D) downstream of the flowmeter. The downstream throttle valve (DTV), a 6-inch motor-operated butterfly valve, is located 8 feet (16D) downstream of the test valve. The DTV is followed by 2-feet of open-ended discharge pipe.

Figure 7 shows actual air blowdown testing in progress with a fully open butterfly valve downstream of the test valve.

Instrumentation

Table 1 provides the range and uncertainty of the following parameters that were measured:

- *Flow*: The flowmeter is a Dieterich Standard Mass ProBar with a 4-20 mAmp output calibrated to measure a mass flow rate of 120,000 PPH. This flow meter is an annubar with a Rosemount differential pressure (DP) transmitter that measures DP and fluid temperature to instantaneously compute mass flow. The ProBar is also equipped with an electronic module that provides the absolute pressure, DP and temperature at the annubar.

- *Pressures:* Pressures are measured on the supply header (P3), and upstream (P1) and downstream (P2) of the test valve, using 100 psig 4-20 mAmp pressure transmitters. Pressure taps for P1 and P2 are located at 2D upstream and 6D downstream of the test valve per ANSI/ISA-S75.02-1996 Standard [12].

- *Differential Pressure*: Two different approaches are used to obtain DP measurements with a high degree of accuracy. Low DP's (under 36 psi) across the test valve are measured using a Rosemount DP transmitter to minimize measurement uncertainty. DP above 36 psi is calculated by subtracting measured values of P2 from P1.

- *Temperatures*: Thermocouples located at 2D upstream and 6D downstream of the test valve measure the fluid temperatures, T1 (upstream) and T2 (downstream), respectively.

- *Angle*: Disc angle is measured using a 5-inch 4-20 mAmp linear position indicator with a cable attached to it that wraps around a grooved drum installed the valve stem. Valve position is measured continuously from 0-degrees (full closed) to 90-degrees (full open).

- *Torque*: Stem torque is measured using a 1500 in-lb calibrated torque cell comprising a full Wheatstone bridge that is directly coupled to the valve stem.

Test Procedure

The standard test sequence consists of 24 open and close strokes consisting of 3 static tests, 7 partial stroke dynamic tests, and 14 full stroke dynamic tests, as shown in Table 1. Two opening and closing static tests are run at the start of the test sequence one with the test specimen un-pressurized and the other with the test specimen pressurized to the maximum test pressure. Only the un-pressurized pair of static tests is repeated at the end of the test sequence. In each of these three static tests the differential pressure across the test valve is zero.

Partial stroke tests are performed at each level of upstream test pressure to be tested (7 in the complete test matrix). Each partial stroke test comprises stroking the valve closed from a 10 degrees open position and then reopening it to a 10 degrees open position. This enables an accurate evaluation of bearing torque and bearing coefficient of friction for the pair of full dynamic strokes that follow the partial stroke.

All dynamic strokes are performed using the *discrete resistance test method*, i.e., the upstream and downstream piping resistance is held constant and the test valve opening angle is varied over its entire range from 0° to 90°. Both the baseline and elbow test matrices are conducted using this typical test sequence.

Additional tests were performed to further evaluate the effect of certain specific parameters on aerodynamic torques and torque coefficients over a wider range of choked and unchoked flow conditions. A series of such tests are *discrete disc angle tests*. During each of these tests, the test specimen's opening angle is held constant, and the flow is varied by throttling the downstream valve. Other examples of additional tests that were performed are: (1) where the downstream throttle valve is removed and the test specimen discharges directly to atmosphere, and (2) when the downstream pressure tap is moved further downstream.

Data Processing

- Raw data acquired from the static, dynamic and bearing torque tests are first averaged per degree of disc opening.
- Packing friction is calculated from static tests performed with and without the system pressure.
- The bearing torque is calculated and used to determine a bearing torque coefficient, which in turn is applied to calculate bearing torque at various valve differential pressures. These bearing torques are used with dynamic torque data at each corresponding disc position to determine the opening and closing aerodynamic torque and torque coefficients.
- Non-dimensional aerodynamic torque coefficients, C_t and C_{tc} are calculated by dividing the aerodynamic torque by the cube of the stem diameter and valve differential pressure or absolute pressure upstream of the test valve, respectively.

Quality Assurance

All testing and model development program activities were conducted under a quality assurance program that meets the 10CFR50 Appendix B requirements.

Results and Discussion

The key results from compressible flow tests performed on a non-symmetric disc of a double-offset design are presented in this section. Results are presented in Figures 8 through 12. The results presented are for shaft downstream orientation (flat face forward) only, since this orientation exhibits certain interesting characteristics (e.g., change in the direction of the hydrodynamic torque) that are not present in the shaft upstream tests. During all blowdown tests, the facility was able to provide a relatively constant pressure upstream of the test valve for various test pressure levels covered by the matrix (Figure 8).

The aerodynamic torque results for different upstream pressures from the highest to the lowest range in the matrix are shown in Figure 9. As expected, the hydrodynamic torque decreases as the upstream pressure is decreased. However, this change is not linear at all disc positions, as shown in the non-dimensional torque coefficients derived from these tests. The torque coefficients vary significantly for different pressures, and at different disc positions. This dependence is different for each disc angle and is governed by the disc shape and the ratio of upstream to downstream pressures. In fact, the torque changes direction from self-opening to self-closing as one transitions from highly choked conditions to low upstream pressure unchoked conditions.

A significant objective of the compressible flow test program was to determine the influence of flow resistance downstream of the test valve on the torque requirements. Since containment purge valves are used in pairs, this is especially important for inboard valve torque requirements because they are influenced by the presence of the outboard valve. Figure 11 shows that the magnitude of the downstream resistance from low to medium to high dramatically affects the hydrodynamic torque requirements for an offset disc with a shaft downstream orientation. This can provide a significant relief on torque requirements for the inboard valve.

To provide more detailed insight into the dependence of aerodynamic torque on pressure ratios from highly choked conditions to unchoked low ΔP conditions discrete angle tests were performed. The results from discrete angle tests for two different disc positions (35° and 60°) are shown in Figure 12. The aerodynamic torque decreases non-linearly as a function of $\Delta P/P_{up}$ ratio, and for high disc opening angles; it changes direction from self-opening to self-closing. The phenomenon responsible for such behavior depends upon pressure distributions on the upstream and downstream side of the faces that are affected by the movement of the shock fronts that emanate at the disc edges and move onto the downstream side of the disc.

In summary, the compressible flow tests are providing validated data for non-dimensional aerodynamic torque coefficients and their dependence on various disc shapes, disc orientation, disc angle, downstream resistance, upstream pressure, and $\Delta P/P_{up}$ ratio.

Benefits In Actual Containment Purge & Vent Valve Applications

Figure 13 shows an actual 18" double-offset disc containment purge valve application in which the use of new validated compressible flow butterfly valve models was able to provide a positive margin (where negative margins were previously calculated using earlier methodologies). This eliminated the need for equipment modifications.

Figures 14 and 15 show 6" precisely scaled models of a 48" single-offset design and 18" double-offset design containment purge valves that were manufactured to address plant-specific margin issues. The prediction of validated torque requirements based on precisely scaled models eliminates the conservative factors included in our new generic butterfly valve models to cover variations in the hydrodynamic torque due to minor variations in disc shapes of basic generic category, (e.g., symmetric, single-offset, double-offset).

Conclusion

The Phase 2 Compressible Flow Test Program provides significant improvement in predicting midstroke torque requirements for quarter-turn valves used in critical, safety-related applications in nuclear power plants. The improvements in AOV margins provided by the new models have eliminated the need for equipment modifications in several applications.

Acknowledgements

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Stroke No.	*Stroke Description*	*Downstream Throttle Valve Angle*	*Target Upstream Pressure, psig*	*Stroke Direction*
1	Static Stroke	O	O	O → C → O
2	Static Stroke	O	Max Pressure	O→ C → O
3	Partial Stroke (10º)	Preset Angle	75	C → O → C
4	Dynamic Stroke	Preset Angle	75	O → C
5	Dynamic Stroke	Preset Angle	75	C → O
6	Partial Stroke (10º)	Preset Angle	60	C → O → C
7	Dynamic Stroke	Preset Angle	60	O → C
8	Dynamic Stroke	Preset Angle	60	C → O
9	Partial Stroke (10º)	Preset Angle	45	C → O → C
10	Dynamic Stroke	Preset Angle	45	O → C
11	Dynamic Stroke	Preset Angle	45	C → O
12	Partial Stroke (10º)	Preset Angle	30	C → O → C
13	Dynamic Stroke	Preset Angle	30	O → C
14	Dynamic Stroke	Preset Angle	30	C → O
15	Partial Stroke (10º)	Preset Angle	15	C → O → C
16	Dynamic Stroke	Preset Angle	15	O → C
17	Dynamic Stroke	Preset Angle	15	C → O
18	Partial Stroke (10º)	Preset Angle	10	C → O → C
19	Dynamic Stroke	Preset Angle	10	O → C
20	Dynamic Stroke	Preset Angle	10	C → O
21	Partial Stroke (10º)	Preset Angle	5	C → O → C
22	Dynamic Stroke	Preset Angle	5	O → C
23	Dynamic Stroke	Preset Angle	5	C → O
24	Static Stroke	O	O	O → C→ O

Table 1

Standard Test Sequence for a Selected Preset Angle of the Downstream Throttle Valve. The Complete Sequence of Tests is Repeated with Different Preset Angles to Simulate Different Downstream Resistances.

Parameter	***Max. Permissible Measurement Uncertainty***	***Range***
Pressures	± 1% of full scale	0-100 psi
Differential Pressure	± 1% of full scale	0-1000 in of water
Stem Torque	± 1% of full scale	0-1500 in-lb
Valve Travel	± 1% of full scale	0-90 degrees
Fluid Temperature	± 2-degrees	-20°F to 200°F
Flow	± 1% of full scale	0-120,000 lbs/hr

Table 2

Instrumentation Range and Accuracy

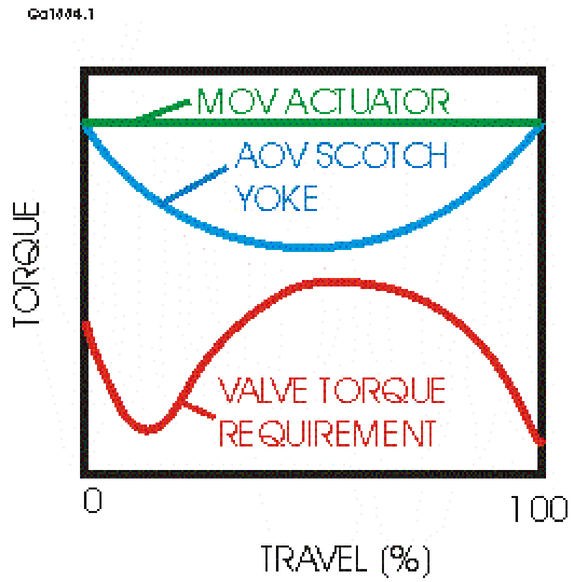


Figure 1

Accurate Position-dependent Torque Predictions are Required for AOVs Because the Actuator Output Varies With Travel.

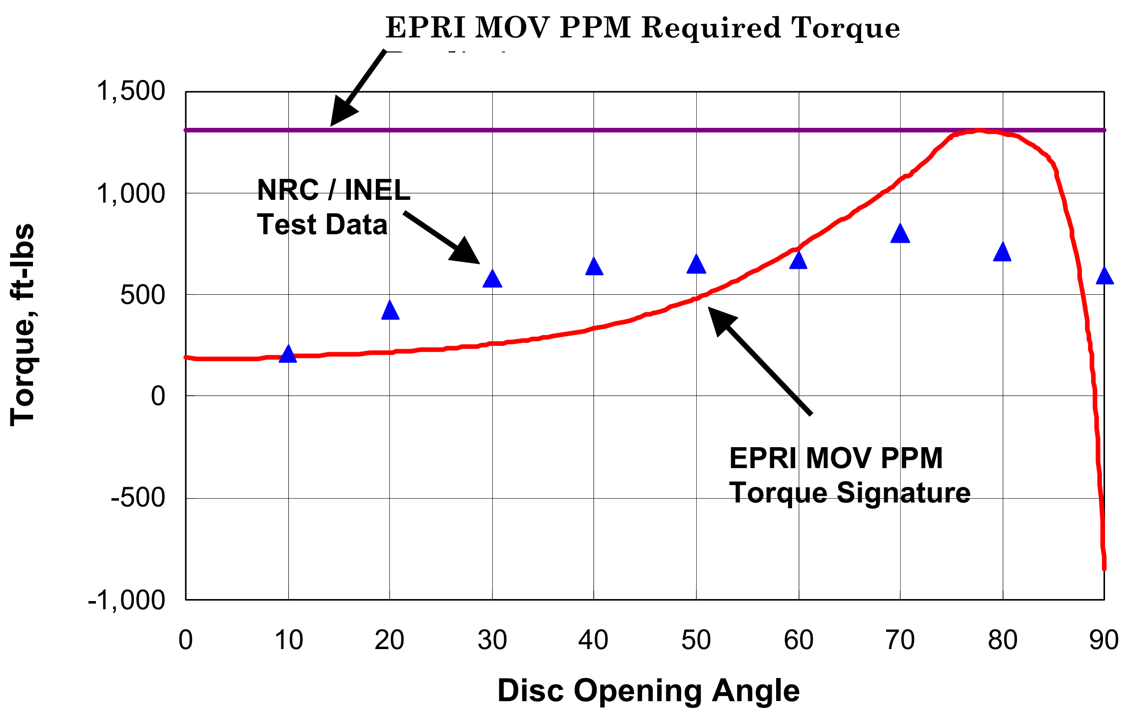


Figure 2

EPRI MOV Required Torque Prediction Bounds Test Data for Compressible Flow. Torque Signature Does Not Bound Test Results And Is Not Intended for Design Basis Calculations [Ref. 5, 6, 8]

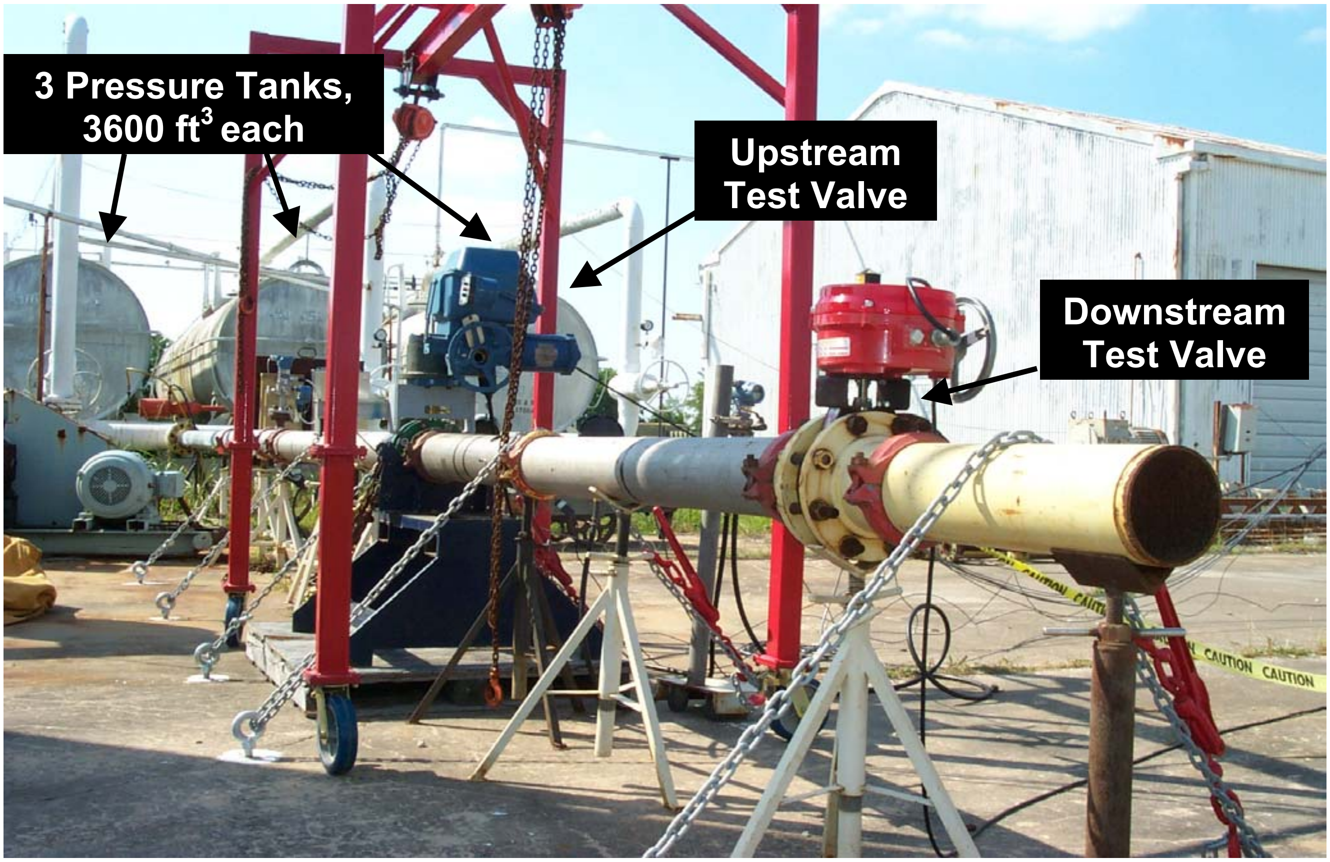
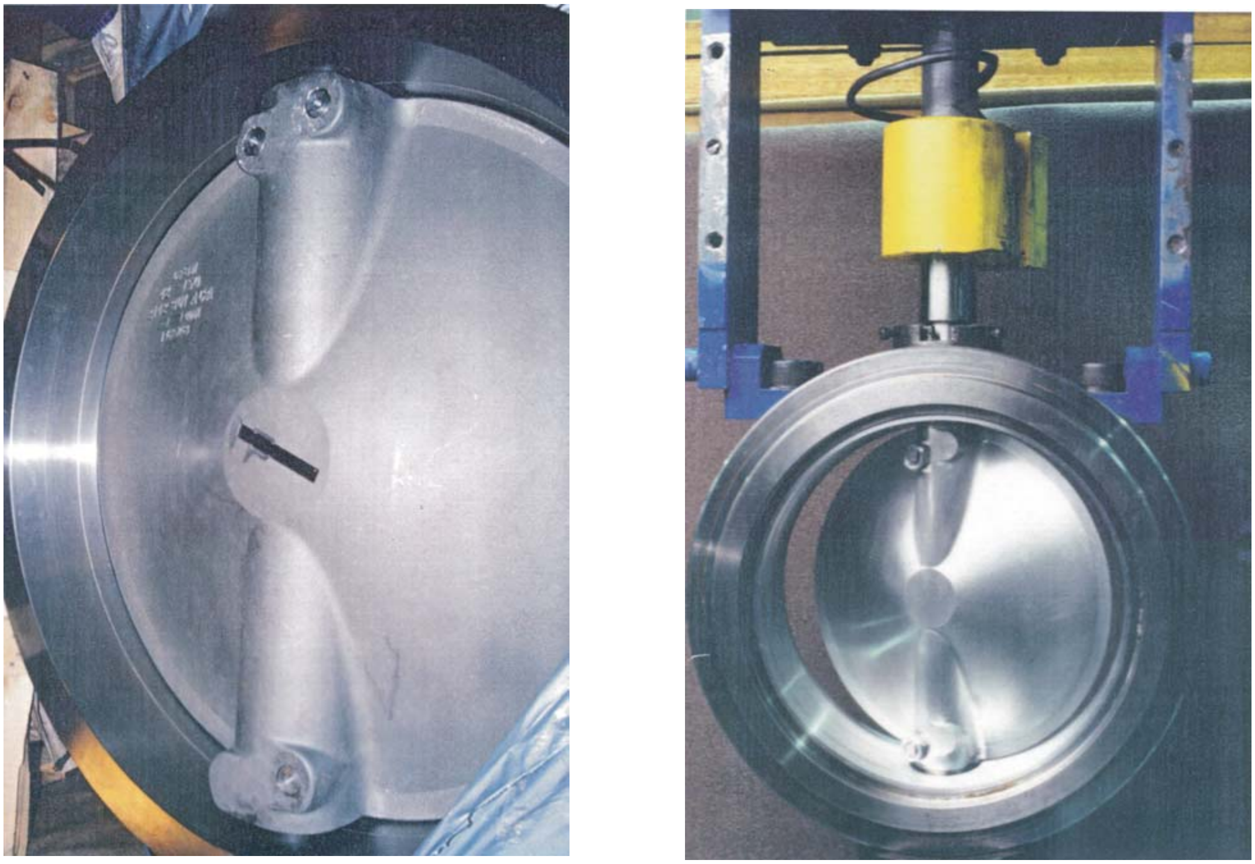


Figure 3

**Compressible Flow Test Facility Showing Test Setup
With Two Butterfly Valves in Series**



42" Valve **6" Model**

Figure 4

A Precisely Scaled 6” Model of a 42” Valve Used to Validate Scaling Methodology

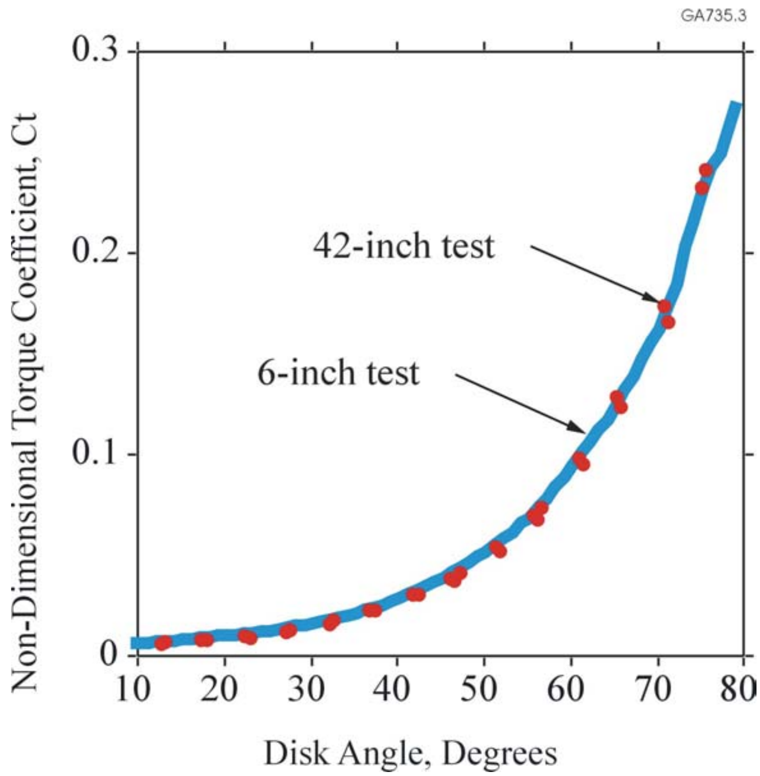


Figure 5

Test Results Validate the Use of Scaled Models to Predict Torque Requirements for Large Valves

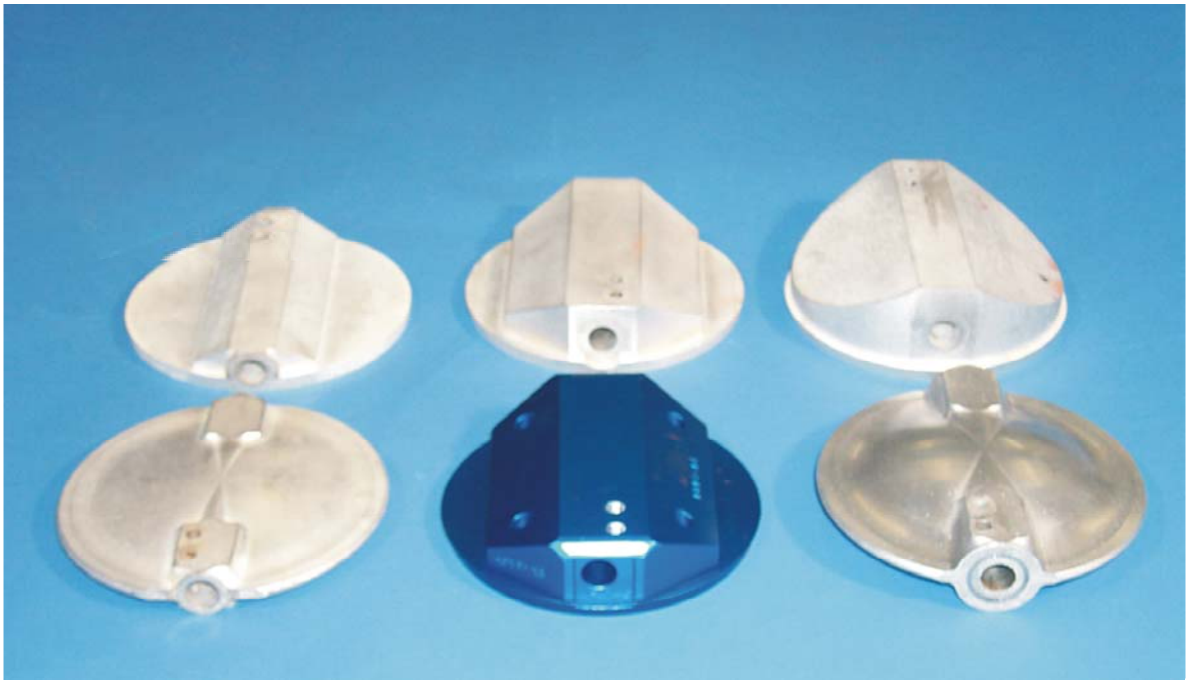


Figure 6

Symmetric, Single-Offset and Double-Offset Disc Geometries Included in the Test Matrix

Figure 7

Air Blowdown Testing in Progress

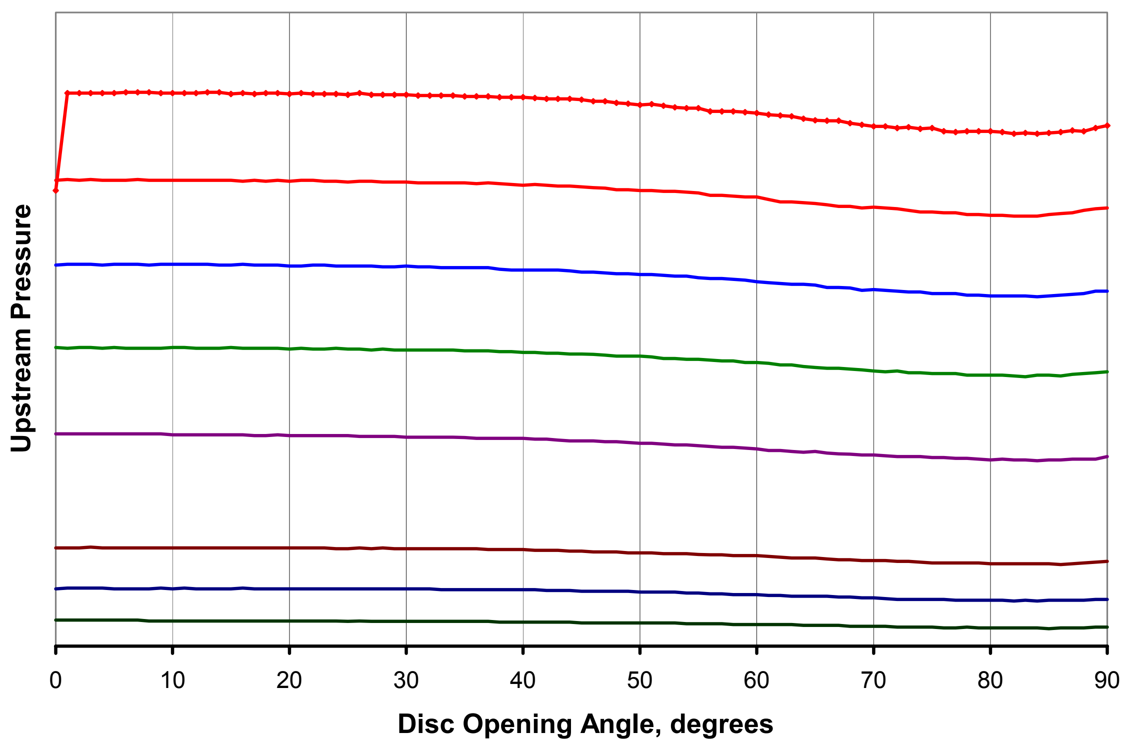


Figure 8

Upstream Test Pressures Were Maintained Relatively Constant for Each Stroke During a Series of Blowdown Tests

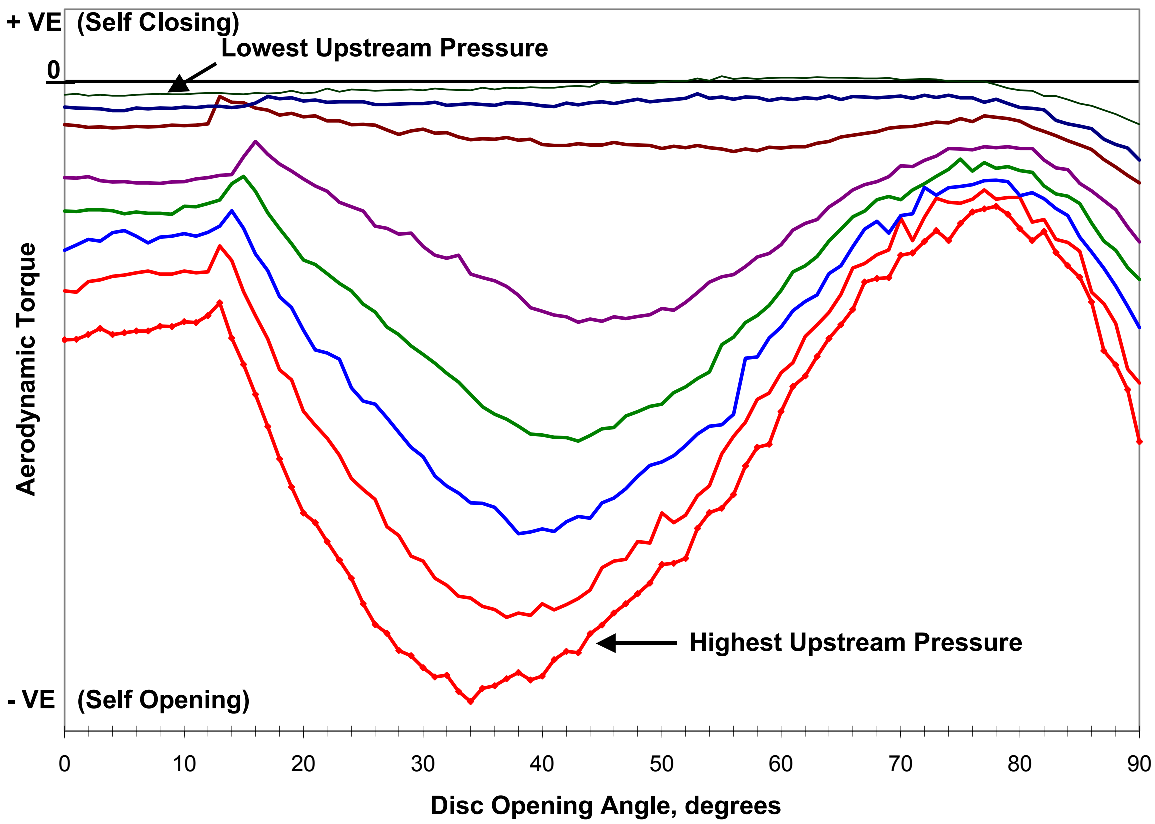


Figure 9

Test Results Showing the Effect of Upstream Pressures and Choking Levels on Dynamic Torque for Shaft Downstream Orientation

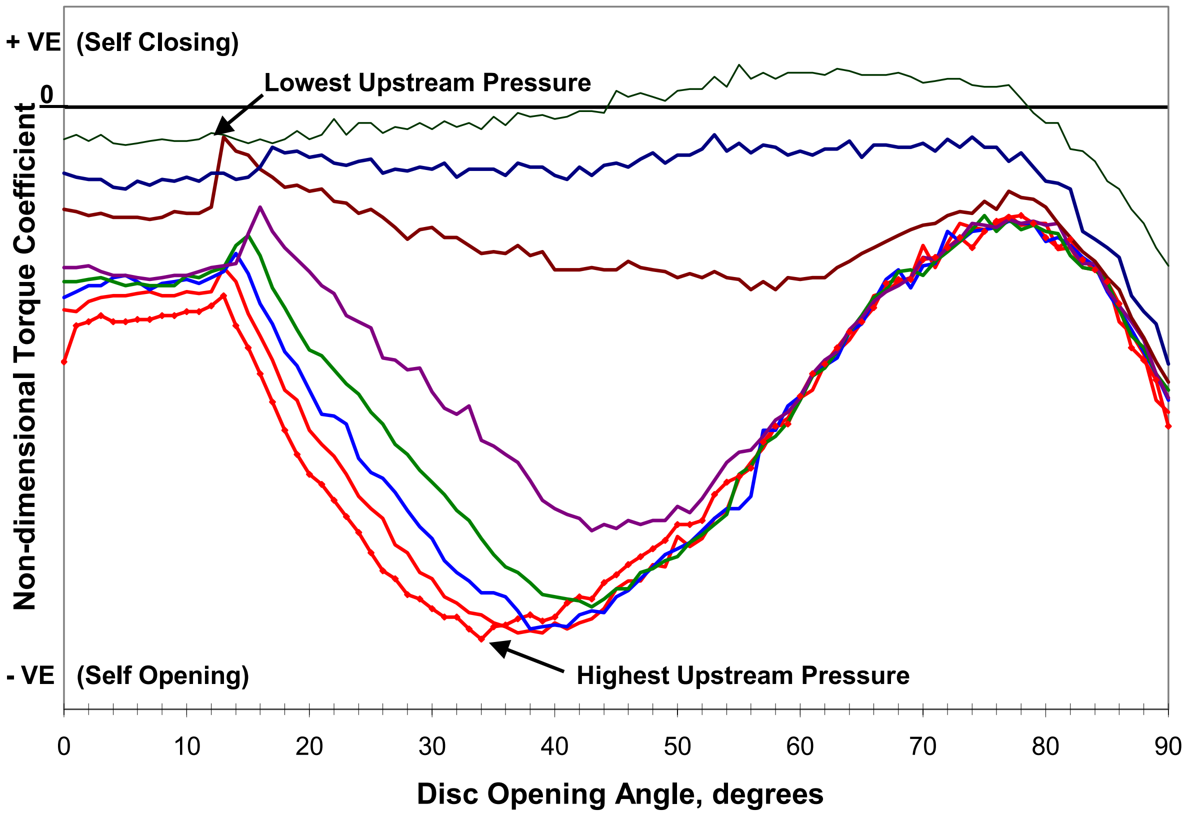


Figure 10

Non-dimensional Aerodynamic Torque Coefficients for Shaft Downstream Exhibit Strong Dependence on Pressure Drop Ratio

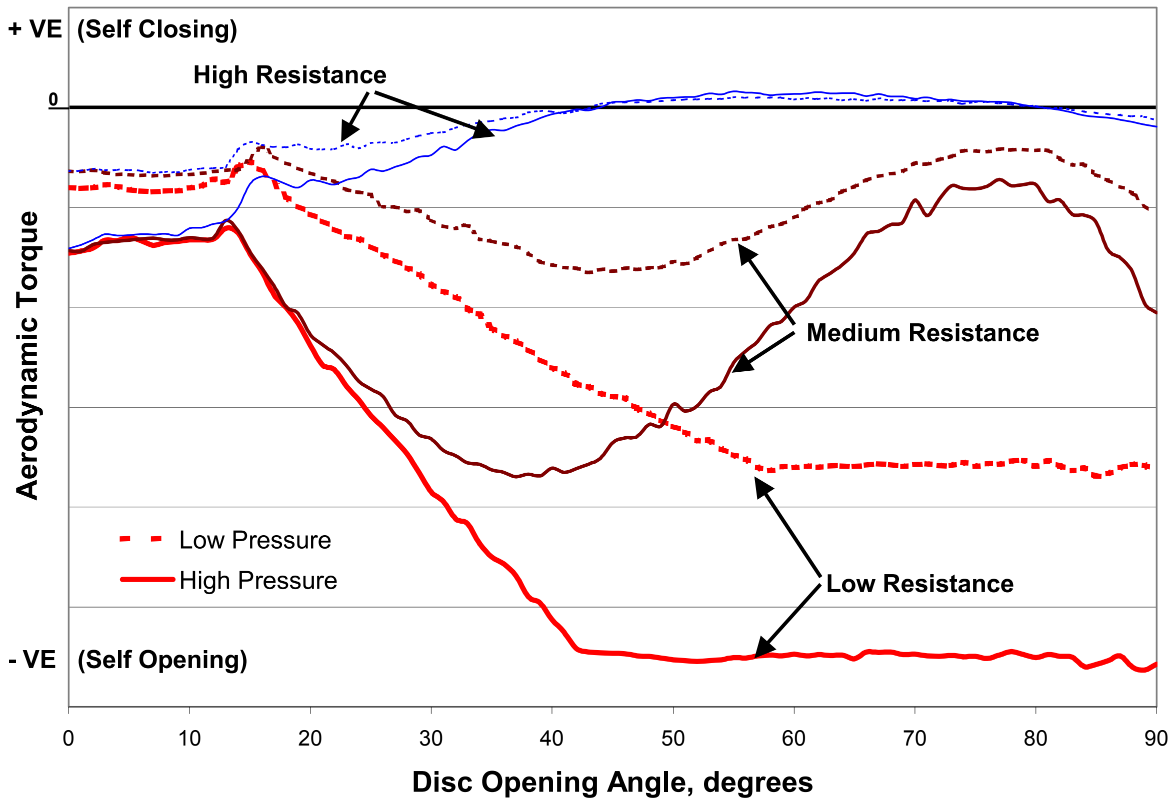


Figure 11

Test Results Showing the Effect of Downstream Resistance on Aerodynamic Torque

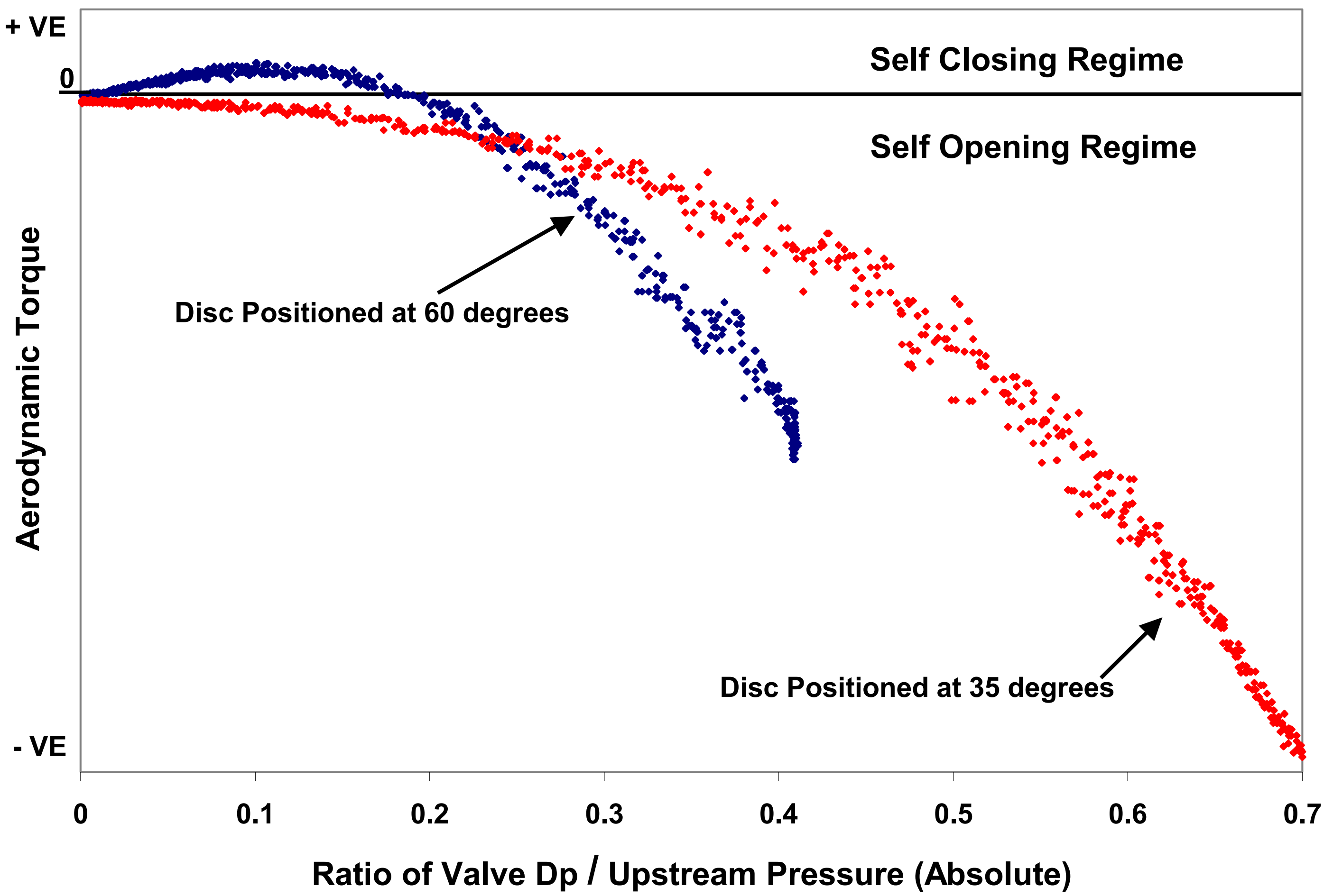


Figure 12

Effect of Valve ΔP / P_{up} Ratio on Aerodynamic Torque for Offset Disc. For High Disc Openings, Torque Changes from Self-Opening to Self-Closing as ΔP / P_{up} is Lowered.

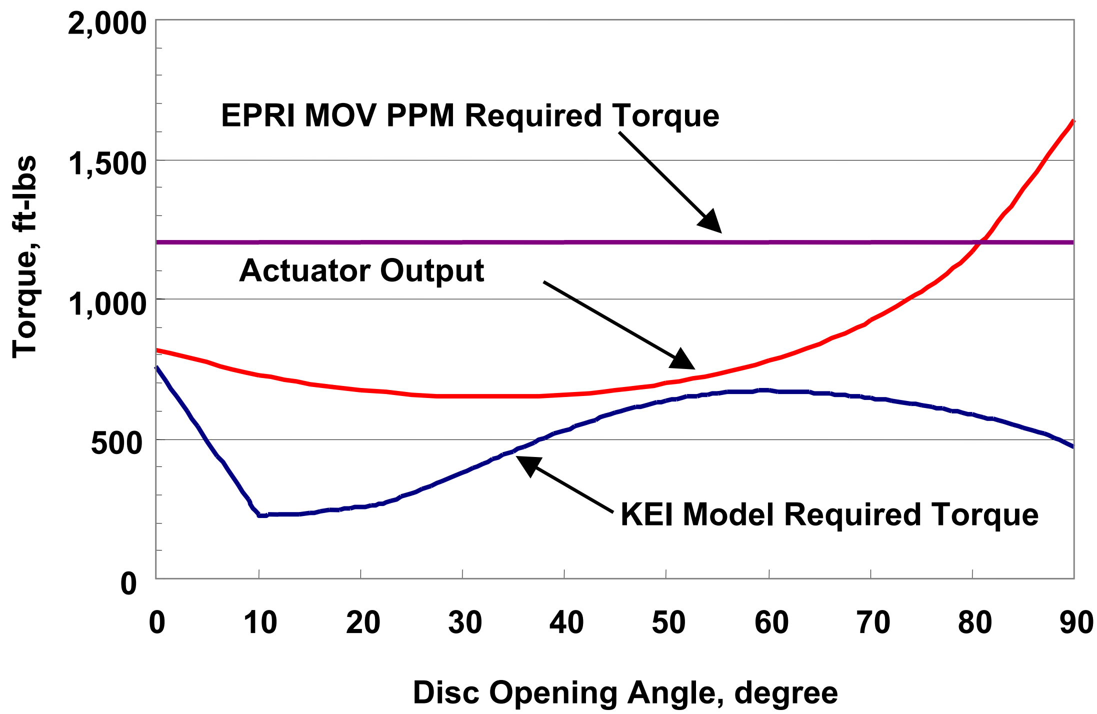


Figure 13

An Actual Containment Purge Valve Application Shows Margin Improvement Achieved by the Use of New Validated Compressible Flow Butterfly Models.

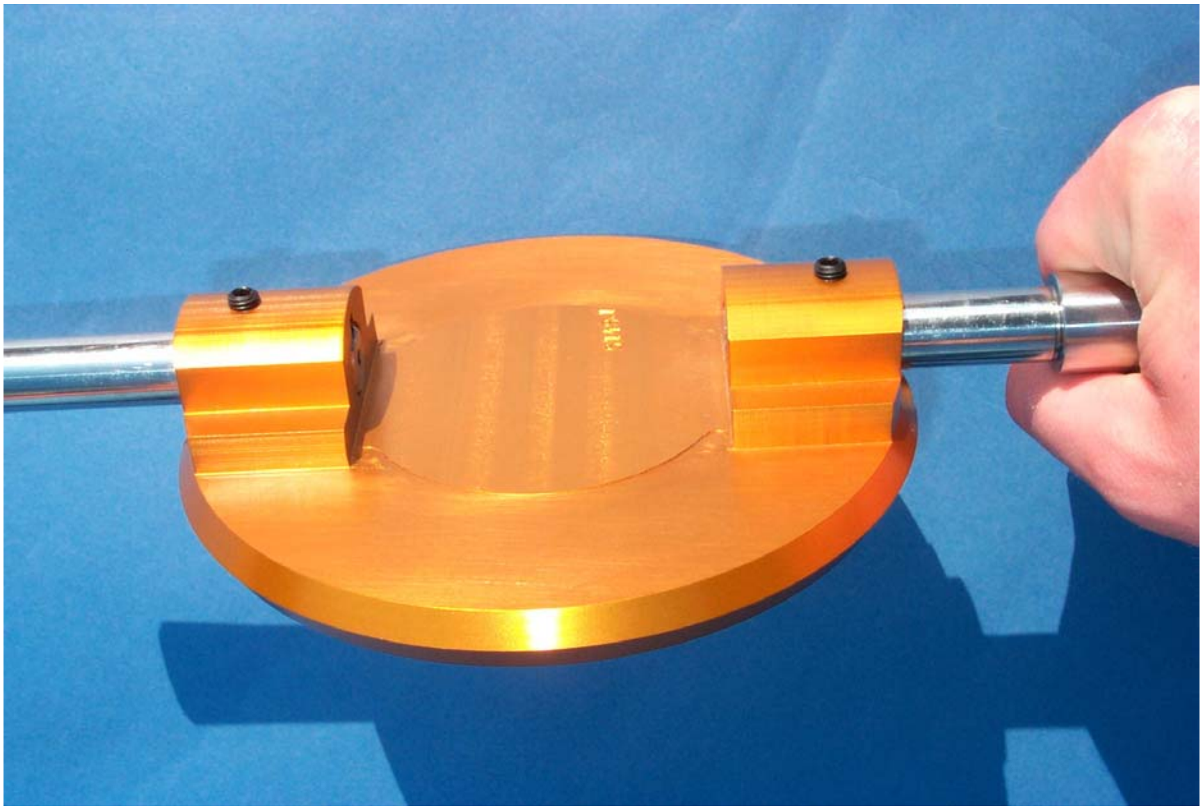


Figure 14

6” Precisely Scaled Model Of a
48” Single-Offset Design Containment Purge Valve

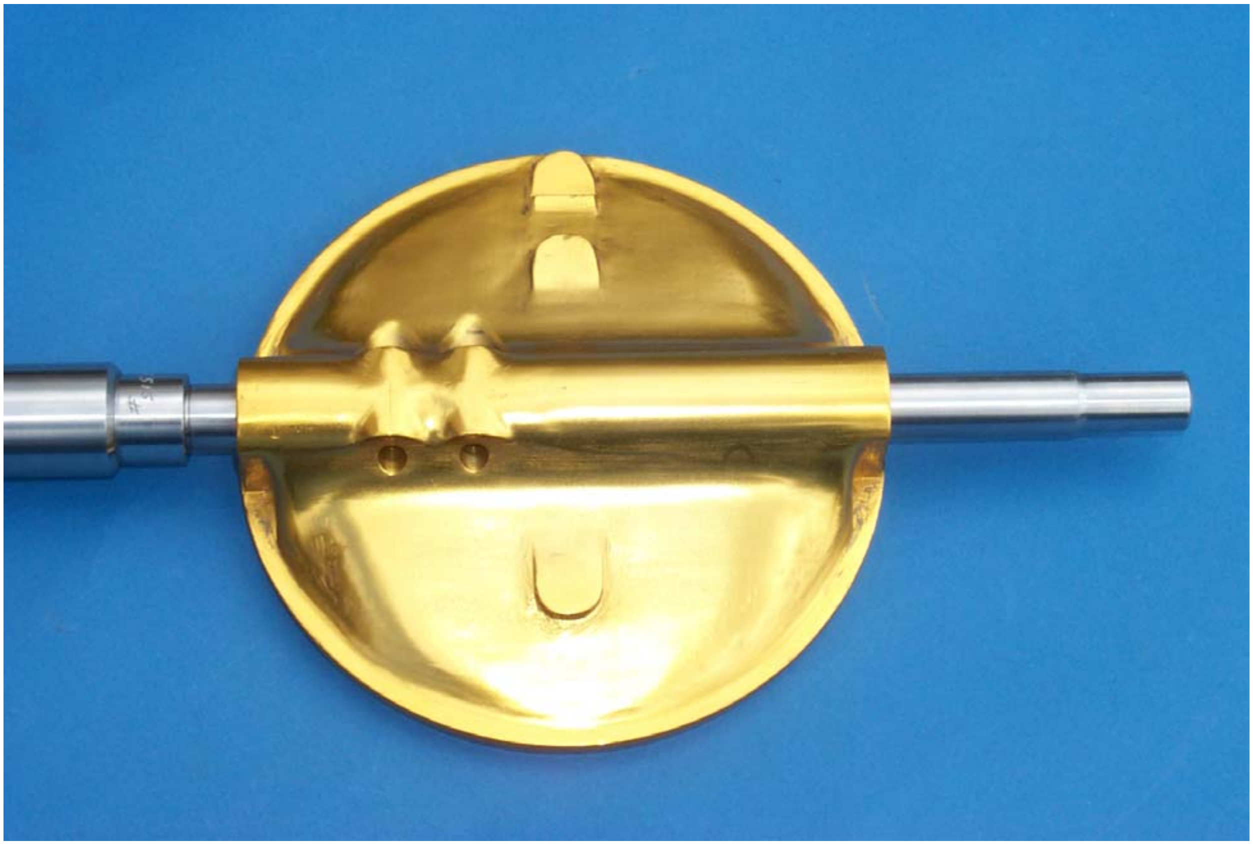


Figure 15

6" Precisely Scaled Model Of a 18" Double-Offset Design Containment Purge Valve